



Electrical Submersible Pumps Manual

Design, Operations,
and Maintenance

Second Edition

Gabor Takacs



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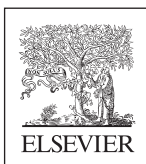
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Preface to the First Edition

Electrical submersible pumping (ESP) is the only kind of artificial lifting for which the original time of invention is known exactly and can be attributed to one man, **Armais Arutunoff**. He made his first experiments in the Baku oilfields at the Caspian Sea in the late 1910s, and was later the founder of the company “**Russian Electrical Dynamo of Arutunoff**” whose acronym **REDA** is still very well known all over the world. Arutunoff’s (who alone received about 90 patents related to submersible equipment) pioneering work started an industry that today lifts $\sim 10\%$ of the world’s crude oil production.

From their early days on, ESP units have excelled in lifting much greater liquid rates than most of the other types of artificial lift and have found their best use in high-rate onshore and offshore applications. Continuous technological development in the last almost 100 years has enormously modified the application ranges for ESP equipment. High gas production, quickly changing liquid production rates, viscous crudes, etc., conditions once very detrimental to ESP operations, are now easily handled by present-day units. All these end up in the indispensability of ESP equipment in the petroleum industry not only today but in the foreseeable future as well.

I wrote this book with the needs of a petroleum engineering graduate student in mind and with the objective of covering all aspects of up-to-date theories and practices in ESP technology. While working on the manuscript, I used parts of it in industrial short courses and I always considered the feedback from participants when improving the material. This way, I believe, the target audience of the book is even broader and includes practicing engineers as well. Throughout the text, worked examples help readers understand basic principles as well as design and analysis procedures.

This book, along with its two predecessors (*Modern Sucker-Rod Pumping*, PennWell, 1993 and *Gas Lift Manual*, PennWell, 2005), concludes my coverage of the three most important artificial lift methods: sucker-rod pumping, gas lifting, and submersible pumping. Because these are the very technologies used on the majority of artificially lifted oil wells, anyone studying these books will have readily available a complete and up-to-date knowledge base encompassing the major artificial lift technologies. I sincerely hope that readers will appreciate the advantages of a uniform approach and treatment of the different topics coming from a single author.

My work on this project spanned several years and I started writing the book while heading the Petroleum Engineering Department of the Miskolc University, Hungary. Final chapters, however, were written in Abu Dhabi where, from August 2007 on, I assumed the position of acting Director of the Petroleum Engineering Program at The Petroleum Institute. My sincere thanks are due to the management of both institutions for their continuous support.

Last but not least the everlasting patience and understanding of my wife Bea and a loving family is earnestly appreciated.

Gabor Takacs
Abu Dhabi, May 2008

Preface to the Second Edition

Since the publication of the first edition of this book in 2009 many important developments have occurred in the electrical submersible pumping (ESP) industry. Most of them are evolutionary, but there are at least three revolutionary technological changes that may shape the future. These are the use of powder metallurgy in manufacturing submersible pump stages, the application of permanent magnet synchronous electric motors, and the rise of a new pump technology: the V-pump. The new stage manufacturing process produces more complex pump stages (which are more and more frequently designed by CFD methods) and permits to reach very high pump efficiencies never seen before. Such stages are inherently more balanced than their casted counterparts and can be used at exceptionally high speeds. At high speeds, pump stages develop greatly increased hydraulic heads, so less stages are required and the length and weight of submersible pumps decrease significantly.

The application of permanent magnet motors in other industries is quite widespread: the ESP industry has just started using them. Their advantages are numerous, but the most important is the lower power requirement and the longer service life, as compared with present-day induction motors. In the present scenario of low oil prices the proven 10%–30% reduction in production costs is so desirable that the number of ESP installations using permanent magnet motors grows exponentially.

The totally new pump type called V-Pump excels in conditions detrimental for centrifugal pumps: high viscosity fluids, abrasive solids, or gas problems. Driven by a usual ESP motor, this pump can work under such adverse conditions where no submersible centrifugal pump could operate; it surely will expand the application ranges of ESP operations in the near future.

One does not need ESP (this time extra-sensory perception) to predict that these revolutionary developments will soon become part of the daily routine in ESP operations.

This completely revised and expanded second edition of *Electrical Submersible Pumps Manual* describes in detail the new developments just mentioned along with all the significant advancements in other areas. As before, my objective was to provide the reader a clear assessment of state-of-the-art ESP technology, which is a very important part of the artificial lift discipline. For this reason I surveyed all available sources for new ideas, equipment, and procedures and condensed my findings in this book for the benefit of fellow engineers. The

close to 300 references cited in the text prove that all significant contributions to the advancement of our industry were detected and properly considered.

While researching for the book, I received tremendous amounts of help from too many individuals to name; their contributions are gratefully appreciated.

Gabor Takacs, PhD
Budapest, May 2017

Chapter 1

Introduction

1.1 ARTIFICIAL LIFTING

Usually, oil wells in the early stages of their lives flow naturally to the surface and are called **flowing wells**. Flowing production means that the pressure at the well bottom is sufficient to overcome the sum of pressure losses occurring along the flow path to the separator. When this criterion is not met, natural flow ends and the well **dies**. The two main reasons for a well's death are as follows: (1) their flowing bottom-hole pressure drops below the total pressure losses in the well, or (2) pressure losses in the well become greater than the bottom-hole pressure needed for moving the wellstream to the surface. The first case occurs because of the removal of fluids from the underground reservoir; the second case involves an increasing flow resistance in the well. This can be caused by (1) an increase in the density of the flowing fluid as a result of decreased gas production or (2) various mechanical problems such as a small tubing size and downhole restrictions.

Artificial lifting methods are used to produce fluids from wells already dead or to increase the production rate from flowing wells; several lifting mechanisms are available to choose from. The importance of artificial lifting is clearly seen from the total number of installations: according to one estimate there are ~2 million oil wells worldwide of which about 50% are placed on some kind of artificial lift [1].

One widely used type of artificial lift method uses a **pump** set below the liquid level in the well to increase the pressure so as to overcome the pressure losses occurring along the flow path. Other lifting methods use **compressed gas**, injected from the surface into the well tubing to help lifting of well fluids to the surface. Although all artificial lift methods can be distinguished based on the previous basic mechanisms, the customary classification is somewhat different as discussed below.

1.1.1 Gas Lifting

All versions of gas lift use high-pressure natural gas injected into the wellstream at some downhole point. In **continuous flow** gas lift, a steady rate of gas is injected into the well tubing aerating the liquid and thus reducing the

pressure losses occurring along the flow path. Because of the reduction of flow resistance, the well's original bottom-hole pressure becomes sufficient to move the gas/liquid mixture to the surface and the well starts to flow again. Therefore, continuous flow gas lifting can be considered as the continuation of **flowing production**.

In **intermittent gas lift**, gas is injected periodically into the tubing string whenever a sufficient length of liquid has accumulated at the well bottom. A relatively high volume of gas injected below the liquid column pushes that column to the surface as a slug. Gas injection is then interrupted until a new liquid slug of the proper column length builds up again. Production of well liquids, therefore, is done by **cycles**. The **plunger-assisted** version of intermittent gas lift uses a special free plunger traveling into the well tubing to separate the upward-moving liquid slug from the gas below it. These versions of gas lift physically displace the accumulated liquids from the well, a mechanism totally different from that of continuous flow gas lifting.

1.1.2 Pumping

Pumping involves the use of a **downhole pump** to increase the pressure in the well to overcome the sum of flowing pressure losses. It can be further classified using several different criteria, e.g., the operational principle of the pump used. However, the generally accepted classification is based on the way the downhole pump is driven and distinguishes between rod and rodless pumping.

Rod pumping methods utilize a **string** of rods connecting the downhole pump to the surface driving mechanism, which, depending on the type of pump used, makes an oscillating or rotating movement. The first kinds of pumps to be applied in water and oil wells were of the **positive displacement** type requiring an alternating vertical movement to operate. The dominant and oldest type of rod pumping is **walking-beam pumping**, or simply called sucker-rod pumping. It uses a positive displacement plunger pump, and its most well-known surface feature is a pivoted walking beam.

The need for producing deeper wells with increased liquid volumes necessitated the evolution of **long-stroke** sucker-rod pumping. Several different units were developed with the common features of using the same pumps and rod strings as in the case of beam-type units, but with substantially longer pump stroke lengths. The desired long strokes did not permit the use of a walking beam, and completely different surface driving mechanisms had to be invented. The basic types in this class are distinguished according to the type of surface drive used: pneumatic drive, hydraulic drive, or mechanical drive long-stroke pumping.

A newly emerged rod pumping system uses a **progressing cavity pump** that requires the rod string to be rotated for its operation. This pump, like the plunger pumps used in other types of rod pumping systems, also works on the principle of positive displacement but does not contain any valves.

Rodless pumping methods, as the name implies, do not have a rod string to operate the downhole pump from the surface. Accordingly, other means (besides mechanical) are used to drive the downhole pump, such as electric or hydraulic. A variety of pump types are utilized with rodless pumping including centrifugal, positive displacement, or hydraulic pumps. **Electric submersible pumping** (ESP) utilizes a submerged electrical motor driving a multistage centrifugal pump. Power is supplied to the motor by an electric cable run from the surface. Such units are ideally suited to produce high liquid volumes.

The other lifting systems in the rodless category all employ a high-pressure power fluid that is pumped down the hole. **Hydraulic pumping** was the first method developed; such units have a positive displacement pump driven by a hydraulic engine, contained in one downhole unit. The engine or motor provides an alternating movement necessary to operate the pump section. The hydraulic **turbine-driven** pumping unit consists of a multistage turbine and a multistage centrifugal pump section connected in series. The turbine is supplied with power fluid from the surface and drives the centrifugal pump at high rotational speeds, which lifts well fluids to the surface.

Jet pumping, although it is a hydraulically driven method of fluid lifting, completely differs from the rodless pumping principles discussed so far. Its downhole equipment converts the energy of a high-velocity jet stream into useful work to lift well fluids. The downhole unit of a jet pump installation is the only oil-well pumping equipment known today containing no moving parts.

1.1.3 Comparison of Lift Methods

Although there are some other types of artificial lift known, their importance is negligible compared with those just mentioned. Thus, there is a multitude of choices available to an engineer when selecting the type of lift to be used. Although the use of many of those lifting mechanisms may be restricted or even ruled out by actual field conditions such as well depth, production rates desired, and fluid properties, usually more than one lift system turns out to be technically feasible. It is then the production engineer's responsibility to select the type of lift that provides the **most profitable** way of producing the desired liquid volume from the given well(s). After a decision is made concerning the lifting method to be applied, a complete design of the installation for initial and future conditions should follow.

1.1.3.1 Lifting Capacities

To provide a preliminary comparison of the available artificial lift methods, Fig. 1.1 is presented where approximate maximum liquid production rates of the different installations are given [2] in the function of lifting depth. The figure shows three lifting mechanisms capable of producing exceptionally high liquid rates: gas lifting, ESP, and jet pumping. As seen, gas lifting

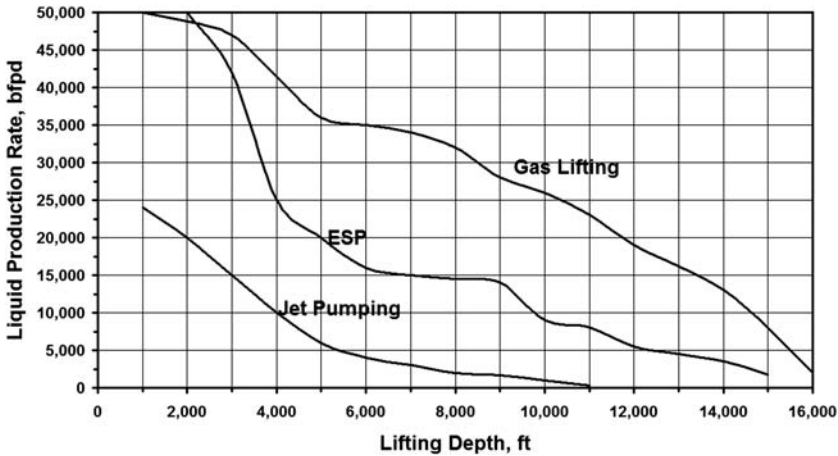


FIGURE 1.1 Maximum liquid production rates versus lifting depth for various high-rate artificial lift methods.

(continuous flow) can produce the greatest amounts of liquid from any depth. In all cases, lifting depth has a profound importance on the liquid volume lifted, with well rates rapidly decreasing in deeper wells.

Fig. 1.2, on the other hand, includes artificial lift methods of moderate liquid production capacity: hydraulic pumping, progressive cavity pumping, rod pumping, and plunger lifting. In most cases, lifting depth negatively affects the lifting capacity of the artificial lifting methods. It can be noted that sucker-rod and progressing cavity pumps produce very similar rates from the lifting

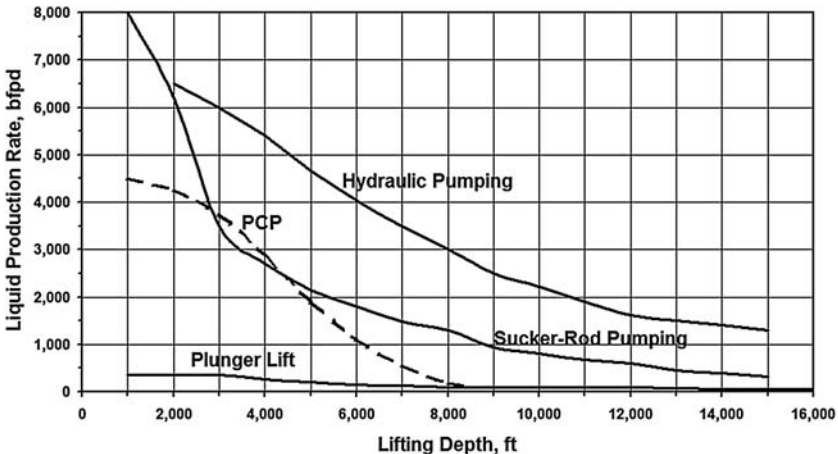


FIGURE 1.2 Maximum liquid production rates versus lifting depth for various artificial lift methods of moderate capacity.

depth range of 3,000 to 6,000 ft; this fact combined with the much lower investment and production costs of PCP installations explains the great popularity of PCP applications over rod pumping applications in recent years.

1.1.3.2 System Efficiencies

Energy efficiencies of present-day artificial lift methods are very different as shown in Fig. 1.3. The overall efficiency of an artificial lift installation is found from the total energy required to operate the system and the hydraulic power spent on lifting the fluids to the surface. The efficiency is the product of the individual efficiencies of the system's components. The decisive part of the overall efficiency is due to the effectiveness of the lifting mechanism, e.g., the energy efficiency of the pump used, but power losses in the well and on the surface can also have a great impact on the final figure. The basic prerequisite for high total energy efficiency, therefore, is the application of a highly efficient lifting mechanism.

The most energy-efficient device available for artificial lifting is the progressing cavity pump that can be more than 70% efficient in converting mechanical energy to hydraulic work. Because the use of progressing cavity pumps in oil wells requires relatively simple surface and downhole installations with low levels of energy losses in system components, PCP systems are the most efficient among the artificial lift methods. No wonder that, wherever well conditions fall in their application ranges, the number of PCP installations is growing very fast.

Next in the line, as shown in Fig. 1.3, are sucker-rod pumping and ESP installations with maximum system efficiencies of about 60%. Although sucker-rod and electric submersible pumps alone can have quite high energy efficiencies, both lifting methods are plagued by high downhole losses in their power transmission system. In addition to these losses, free gas entering the

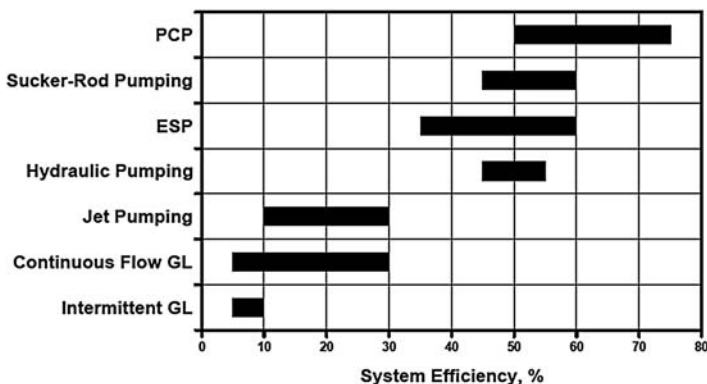


FIGURE 1.3 Energy efficiencies of artificial lift methods.

pumps dramatically reduces their hydraulic output and consequently the overall system efficiency.

Hydraulic pumping installations utilizing positive displacement pumps usually have power efficiencies around 50%. Jet pumping and continuous flow gas lifting are relatively less-efficient artificial lift methods with maximums around 30%. Intermittent gas lift has the lowest energy efficiency among the available lift methods.

1.1.3.3 Further Considerations

The selection of the proper lifting method for a given well or field requires more than comparing the production capabilities and efficiencies of the possible systems. Fluid properties, field conditions, operating and investment cost estimates, and possible production problems are all to be considered before a final decision is arrived at. [Table 1.1](#) provides help for a preliminary selection of possible candidates and for eliminating those methods not suitable for actual conditions.

1.2 SHORT HISTORY OF ELECTRICAL SUBMERSIBLE PUMPING APPLICATIONS

Unlike most of the other artificial lift methods such as gas lifting or sucker-rod pumping, for which invention cannot be attributed to any person or any definite time, ESP was invented and developed by a Russian named **Armais Arutunoff** in the late 1910s [3]. In 1911, he started the company “**Russian Electrical Dynamo of Arutunoff**” (its acronym **REDA** still being known all over the world) and developed the first electric motor that could be operated submersed in an oil well. To get funding for the development of his ideas, **Mr. Arutunoff** first immigrated to Germany in 1919, and then finally settled in the United States in 1923. The **US Patent** he received on the electrical submersible pump [4] was issued in 1926 and covered the principal features of this new artificial lift method. The first ESP installation was successfully operated in the **El Dorado** field in Kansas in 1926. **Mr. Arutunoff** moved to Bartlesville, Oklahoma, in 1928 where he started the Bart Manufacturing Co., later reorganized as **REDA Pump Co.** in 1930.

The first ESP units were driven by three-phase two-pole electric induction motors of $5\frac{3}{8}$ " or $7\frac{7}{8}$ " OD. The biggest motor was about 20 ft long and developed 105 HP. Directly above the motor a “**seal unit**” was attached, the main task of which was to prevent the leakage of well fluids into the motor. On top of the seal unit, a multistage centrifugal pump lifted well fluids to the surface. The complete ESP unit (motor, seal, and pump) was run into the well on the bottom of the tubing string, electricity being supplied from the surface to the motor by a special three-conductor cable. Even today, these are the main components of ESP installations. After more than 80 years of operation, the **REDA Company** established by **Mr. Arutunoff**, who alone received 90

TABLE 1.1 Main Features of Artificial Lift Installations

AL Method	SR Pumping	Gas Lifting	ESP	PCP	Hydraulic Pumping	Jet Pumping	Plunger Lift
Maximum operating depth, ft	16,000	18,000	15,000	12,000	17,000	15,000	19,000
Maximum operating rate, bpd	6,000	50,000	60,000	6,000	8,000	20,000	400
Maximum operating temperature, °F	550	450	400	250	550	550	550
Gas handling	Fair to Good	Excellent	Fair	Good	Fair	Good	Excellent
Corrosion handling	Good to Excellent	Good to Excellent	Good	Fair	Good	Excellent	Excellent
Solids handling	Fair to Good	Good	Fair	Excellent	Fair	Good	Fair
Fluid gravity, API	>8°	>15°	>10°	<40°	>8°	>8°	>15°
Offshore application	Limited	Excellent	Excellent	Limited	Good	Excellent	N/A

ESP, electric submersible pumping; *PCP*, progressing cavity pumping; *SR*, sucker rod.

Modified after Lea JF. Artificial lift selection. SPE petroleum engineering handbook, vol. IV: Society of Petroleum Engineers; 2007 [Chapter 10].

patents related to submersible equipment, is still one of the leading suppliers of ESP equipment to the world's petroleum industry. Most of the other US manufacturing firms have roots related to the original company [5].

From its conception on, ESP units have excelled in lifting much greater liquid rates than most of the other types of artificial lift and found their best use in high-rate onshore and offshore applications. It is believed that today ~10% of the world's oil supply is produced with submersible pumping installations.

During its long history, ESP equipment underwent a continuous improvement. The first breakthrough came in the early 1950s when “**seal units**” (aka **protectors**) with mechanical seals on their shafts considerably increased ESP run lives because they provided a much better protection against leakage of well fluids into the motor. Production of gassy wells was always a problem and the use of simple gravitational (reverse-flow) gas separators did not solve the problem completely until the first **rotary gas separator** [6] was introduced in the early 1970s. Although the other components of the ESP unit have also evolved, the next revolutionary moment came when the first variable speed ESP unit was installed in August 1977 [7]. The **variable speed drive** (VSD) changes the frequency of the electric current driving the ESP motor and thus considerably modifies the head performance of the submersible pump. By properly setting the driving frequency, a very basic limitation of ESP units can be eliminated, and the lifting capacity of the submersible pump can easily be modified to match the inflow performance of the well. Without a VSD unit, in wells with unknown liquid production capacities the ESP unit must be exchanged with a unit better fitting the inflow to the well, which usually involves a costly workover operation.

Running and pulling of conventional ESP units involves the use of heavy workover units because the tubing string should be moved in or out of the well. Reduction of the high workover costs can be accomplished if the ESP unit is run on a wire rope of the right mechanical strength. **Cable-suspended** units, first appearing in the oil field in the late 1970s, became very popular for their advantageous features, especially in the offshore environment [8]. Similar advantages can be reached with **coiled tubing** (CT)-conveyed ESP units, first installed in Alaskan fields in 1992 [9].

1.3 BASIC FEATURES OF ELECTRIC SUBMERSIBLE PUMPING INSTALLATIONS

1.3.1 Applications

The first big-scale success for ESP installations occurred in the late 1920s when the Oklahoma City field was converted from beam pumping to submersible pumping. The ESP units could lift oil volumes of up to 1,000 bpd, an amount two to three times greater than beam pumping units were able to

produce [3]. Early applications also showed their advantages in waterflood operations where increasing the liquid rates could greatly raise oil production.

Today, main applications include onshore waterflood operations (both liquid production and water injection), offshore production, and all other cases where electricity is available and large volumes must be lifted. The usual range of liquid rates, in the typical installation depth range of 1,000–10,000 ft, is between 20,000 and 200 bpd, heavily decreasing with well depth (see Fig. 1.1). Extreme depth and liquid rate limits of present-day ESP units are around 15,000 ft and 30,000 bpd, respectively. One recent case study [10] reported a sustained liquid rate of 31,800 bpd from installations utilizing 2,000 HP tandem motors from wells with 5¹/₂" tubing and 9⁵/₈" casing strings.

Special applications outside the petroleum industry, such as production from **water supply wells**, can reach much greater liquid rates; one major manufacturer offers submersible pumps capable of 64,000 bpd maximum production.

1.3.2 Advantages and Limitations

General **advantages** of using ESP units can be summed up as follows, based on [11–13]:

- Ideally suited to produce **high** to **extremely high** liquid volumes from medium depths. Maximum rate is around 30,000 bpd from 1,000 ft.
- Energy **efficiency** is relatively high (around 50%) for systems producing over 1,000 bpd.
- Can be used in **deviated wells** if set in a straight section.
- Requires **low maintenance**, provided the installation is properly designed and operated.
- Can be used in **urban locations** because surface equipment requires minimal space.
- Well suited to the **offshore** environment because of the low space requirements.
- Corrosion and scale treatments are relatively easy to perform.

General **disadvantages** are listed below:

- A reliable source of **electric power** of relatively high voltage must be available.
- The **flexibility** of ESP systems running on a constant electrical frequency is very low because the centrifugal pump's liquid producing capacity practically cannot be changed. Proper installation design based on accurate well inflow data and matching the unit's capacity to well deliverability is crucial, otherwise costly workover operations are required to run a new unit in the well. The use of **VSDs** can eliminate most of these problems but at an extra cost.

- **Free gas** present at suction conditions deteriorates the submersible pump's efficiency and can even totally prevent liquid production. The use of gas separators or gas handlers is required if more than 5% of free gas enters the pump.
- Sand or **abrasive** materials in well fluids increase equipment wear. Special abrasion-resistant materials are available but increase capital costs.
- Repair of ESP equipment in oil-field conditions is difficult; faulty equipment must be sent to the manufacturer's repair shop.
- High well **temperature** is a limiting factor, standard equipment is limited to about 250°F, and use of special materials increases the temperature limit to 400°F.
- Production of **high-viscosity** oils increases power requirements and reduces lift.
- Running and pulling costs are high because of the need for heavy workover rigs. Cable-suspended or CT-deployed ESP units reduce workover costs.

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Chapter 2

Review of Fundamentals

2.1 INTRODUCTION

People familiar with the electrical submersible pumping (ESP) industry realize that this kind of artificial lift involves the application of **several** engineering disciplines: electrical, mechanical, and hydraulic. A basic background in these fields is necessary because all are needed to design, operate, or analyze ESP installations. Some of the important applications of these disciplines are listed here:

- Electrical engineering
 - Voltage drop in cables
 - Operation of electric motors and transformers
- Mechanical engineering
 - Operation of high-speed shafts
 - Mechanical seals
 - Radial and axial bearings
- Hydraulics
 - Flow in well tubing
 - Operation of centrifugal pumps

This chapter presents a short summary of background theories of the most important topics usually met in designing and analyzing ESP installations and covers well inflow performance, basic hydraulics, electrical fundamentals, and the basics of systems (Nodal) analysis.

2.2 WELL INFLOW PERFORMANCE

2.2.1 Introduction

The proper design of any artificial lift system requires an accurate knowledge of the fluid rates that can be produced from the reservoir through the given well. Present and also future production rates are needed to accomplish the following basic tasks of production engineering:

- selection of the right **type** of lift,
- detailed **design** of production equipment, and
- estimation of future well performance.

The production engineer, therefore, must have a clear understanding of the effects governing **fluid inflow** into a well. Lack of information may lead to overdesign of production equipment or, in contrary, equipment limitations may restrict attainable liquid rates. Both of these conditions have an undesirable impact on the economy of artificial lifting and can be the cause of improper decisions as well.

A well and a productive formation are interconnected at the **sandface**, which is the cylindrical surface where the reservoir is opened. As long as the well is kept shut in, sandface pressure equals reservoir pressure and thus no inflow occurs to the well. It is easy to see that, in analogy to flow in surface pipes, fluids in the reservoir flow only between points having different pressures. Thus, a well starts to produce when the pressure at its sandface is decreased below reservoir pressure. Fluid particles in the vicinity of the well then move in the direction of pressure decrease and, after an initial period, **a stabilized rate** develops. This rate is controlled mainly by the pressure prevailing at the sandface, but it is also affected by a multitude of parameters such as reservoir properties (rock permeability, pay thickness, etc.), fluid properties (viscosity, density, etc.), and well completion effects (perforations, well damage). These latter parameters being constant for a given well, at least for a considerable length of time, the only means of controlling production rates is the control of **bottom-hole pressures**. The proper description of well behavior, therefore, requires that the relationship between bottom-hole pressures and the corresponding production rates be established. The resulting function is called the well's **inflow performance relationship** (IPR) and is usually obtained by running **well tests**.

2.2.2 The Productivity Index Concept

The simplest approach to describe the inflow performance of oil wells is the use of the **productivity index** (PI) concept. It was developed using the following simplifying assumptions:

- Flow is **radial** around the well.
- A single-phase, **incompressible** liquid is flowing.
- Permeability distribution in the formation is **homogeneous**.
- The formation is fully saturated with the given liquid.

For the previous conditions, **Darcy's equation** can be solved to find the production rate of the well:

$$q = \frac{0.00708 k h}{\mu B \ln\left(\frac{r_e}{r_w}\right)} (p_R - p_{wf}) \quad (2.1)$$

where:

q = liquid rate, STB/d
 k = effective permeability, mD
 h = pay thickness, ft
 μ = liquid viscosity, cP
 B = liquid volume factor, bbl/STB
 r_e = drainage radius of well, ft
 r_w = radius of wellbore, ft
 p_R = reservoir pressure, psi
 p_{wf} = flowing bottom-hole pressure, psi

Most parameters on the right-hand side of the above equation are constant, which permits collecting them into a single coefficient called PI :

$$q = PI (p_R - p_{wf}) \quad (2.2)$$

where:

q = liquid rate, STB/d
 PI = productivity index, STB/d/psi
 p_R = reservoir pressure, psi
 p_{wf} = flowing bottom-hole pressure, psi

This equation states that liquid inflow into a well is directly proportional to **pressure drawdown**. It plots as a straight line on a pressure versus liquid flow rate diagram, as shown in Fig. 2.1. The endpoints of the PI line are the **average reservoir pressure**, p_R , at a flow rate of zero and the maximum **potential rate** at a bottom-hole flowing pressure of zero. This maximum rate is the well's **absolute open flow potential (AOFP)** and represents the flow rate that would occur if flowing bottom-hole pressure ($FBHP$) could be reduced to zero. In practice, it is not possible to achieve this rate, and the $AOFP$ is only used to compare the deliverability of different wells.

The use of the PI concept is quite straightforward. If the average reservoir pressure and the PI are known, Eq. (2.2) gives the flow rate for any $FBHP$. The well's PI can either be calculated from reservoir parameters or be measured by taking flow rates at various $FBHP$ s.

Example 2.1

A well was tested at $p_{wf} = 1,500$ psi pressure and it produced $q = 1,000$ bpd of oil. Shut-in bottom-hole pressure was $p_{ws} = 2,500$ psi. What is the well's PI and what is the oil production rate at $p_{wf} = 1,000$ psi?

Solution

Solving Eq. (2.2) for PI and substituting the test data:

$$PI = q / (p_R - p_{wf}) = 1,000 / (2,500 - 1,500) = 1 \text{ bopd/psi.}$$

The rate at 1,000 psi is found from Eq. (2.2):

$$q = PI(p_{ws} - p_{wf}) = 1(2,500 - 1,000) = 1,500 \text{ bopd.}$$

2.2.3 Inflow Performance Relationships

2.2.3.1 Introduction

In many wells on artificial lift, bottom-hole pressures below **bubble-point pressure** are experienced. Thus, there is a **free gas** phase present in the reservoir near the wellbore, and the assumptions that were used to develop the PI equation are no longer valid. This effect was observed by noting that the PI was not a constant as suggested by Eq. (2.2). Test data from such wells indicate a downward curving line, instead of the straight line shown in Fig. 2.1.

The main cause of a curved shape of inflow performance is the **liberation** of solution gas due to the decreased pressure in the vicinity of the wellbore. This effect creates an increasing gas saturation profile toward the well and simultaneously decreases the effective permeability to liquid. Liquid rate is accordingly decreased in comparison with single-phase conditions and the well produces less liquid than indicated by a straight-line PI curve. Therefore, the constant PI concept cannot be used for wells producing below the bubble-point pressure. Such wells are characterized by their IPR curve, which is discussed in the following section.

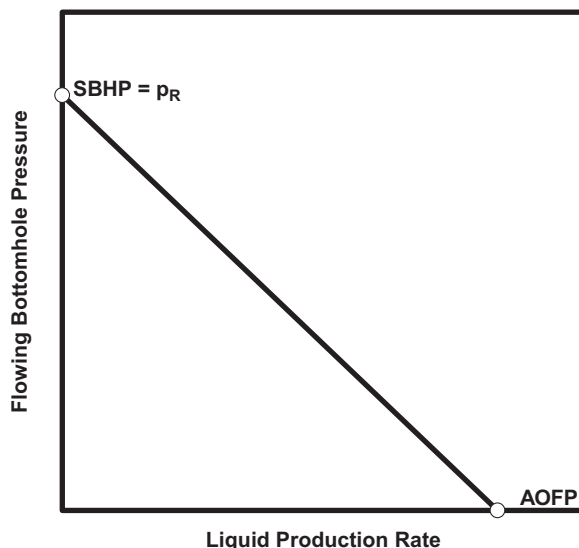


FIGURE 2.1 Well inflow performance with the constant productivity index concept. *AOFIP*, absolute open flow potential; *SBHP*, static bottom-hole pressure.

2.2.3.2 Vogel's Inflow Performance Relationship Correlation

Vogel used a numerical reservoir simulator to study the inflow performance of wells depleting solution gas drive reservoirs. He considered cases below bubble-point pressure and varied pressure drawdowns, fluid, and rock properties. After running several combinations on the computer, Vogel found that all the calculated IPR curves exhibited the same general shape [1]. This shape is best approximated by a dimensionless equation given as follows and is graphically depicted in Fig. 2.2.

$$\frac{q}{q_{\max}} = 1 - 0.2 \frac{p_{wf}}{p_R} - 0.8 \left(\frac{p_{wf}}{p_R} \right)^2 \quad (2.3)$$

where:

q = production rate at bottom-hole pressure p_{wf} , STB/d

$q_{\max}$ = maximum production rate, STB/d

p_R = average reservoir pressure, psi

p_{wf} = flowing bottom-hole pressure, psi

Although Vogel's method was originally developed for solution gas drive reservoirs, the use of his equation is generally accepted for other drive mechanisms as well. It was found to give reliable results for almost any well with a bottom-hole pressure below the bubble point of the crude.

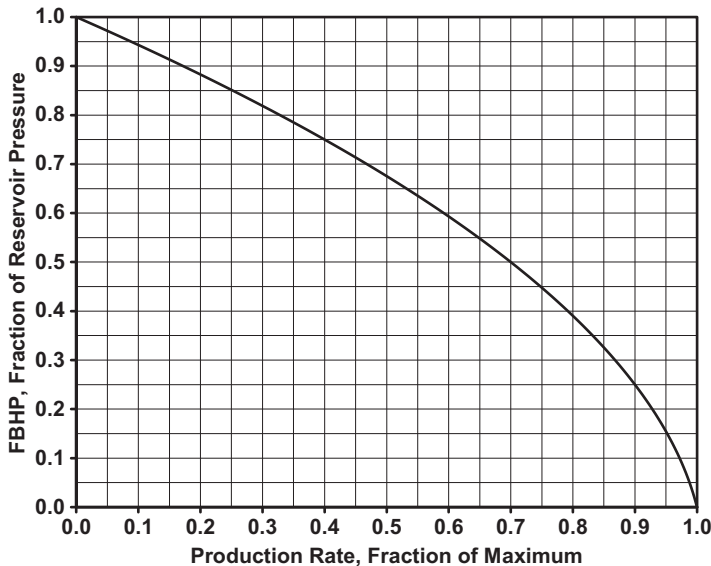


FIGURE 2.2 Vogel's [1] dimensionless inflow performance curve. *FBHP*, flowing bottom-hole pressure.

To use Vogel’s method, reservoir pressure needs to be known along with a single stabilized rate and the corresponding *FBHP*. With these data, it is possible to construct the well’s IPR curve by the procedure discussed in the following example problem.

Example 2.2

Using data of the previous example find the well’s maximum flow rate (*AOFP*) and construct its IPR curve, by assuming multiphase flow in the reservoir.

Solution
Substituting the test data into Eq. (2.3):

$$1,000/q_{\max} = 1 - 0.2(1,500/2,500) - 0.8(1,500/2,500)^2 = 0.592.$$

From the above equation the *AOFP* of the well:

$$q_{\max} = 1,000/0.592 = 1,689 \text{ bopd.}$$

Now find one point on the IPR curve, where $p_{wf} = 2,000$ psi using Fig. 2.2. $p_{wf}/p_R = 2,000/2,500 = 0.8$, and from Fig. 2.2:

$$q/q_{\max} = 0.32, \text{ and } q = 1,689 \cdot 0.32 = 540 \text{ bopd.}$$

The remaining points of the IPR curve are evaluated similarly.

Fig. 2.3 shows the calculated IPR curve along with a straight line valid for $PI = 1$, as found in Example 2.1. Calculated parameters for the two cases are listed as follows:

	Maximum Rate	Rate at 1,000 psi
Vogel	1,689	1,330
Constant <i>PI</i>	2,500	1,500

Comparison of the preceding results indicates that considerable errors can occur if the constant *PI* method is used for conditions below the bubble-point pressure.

2.2.3.3 The Composite Inflow Performance Relationship Curve

The **Vogel** correlation, as discussed in the previous section, can be applied (1) if the well’s *FBHPs* are below the bubble-point pressure and (2) if only oil is produced. The **composite IPR curve** introduced by **Brown [2]** eliminates these restrictions and provides a way to describe the well’s inflow performance in a broad range of conditions.

It should be clear that inflow conditions at pressures greater than the bubble-point pressure are described by the constant *PI* principle discussed previously. According to **Darcy’s Law**, however, the same constant *PI* should control the inflow conditions of a well producing water only. Thus, in wells with a water cut of 100%, inflow for any conditions is described by the *PI* alone.

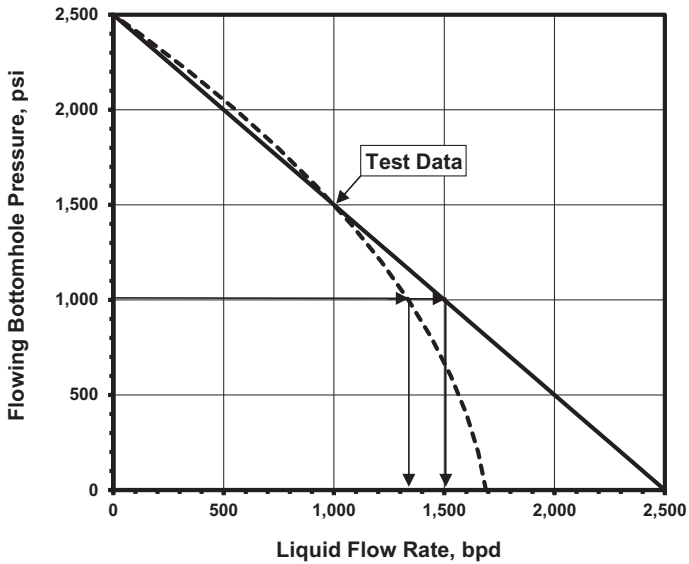


FIGURE 2.3 Comparison of inflow performance relationship curves for Example 2.2.

Wells producing liquids with a water cut of less than 100% and with pressures lower than the bubble-point pressure should have IPR curves somewhere between the curves valid for pure oil (the Vogel correlation) and the ones valid for pure water production. The schematic description of such **composite IPR curves** is presented in Fig. 2.4.

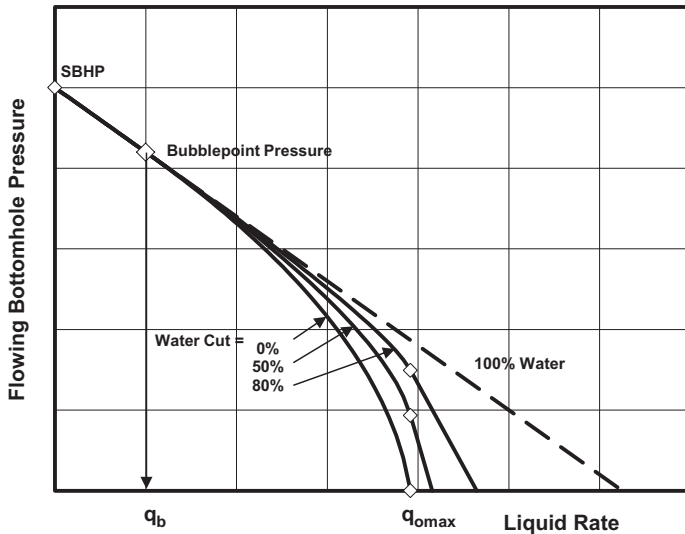


FIGURE 2.4 Schematic depiction of composite inflow performance relationship curves. *SBHP*, static bottom-hole pressure.

The composite curves shown in the figure exhibit three distinct intervals:

1. Well inflow at pressures greater than the bubble-point pressure is along a straight line having a slope equal to the PI .
2. For liquid rates less than the maximum oil rate q_{omax} , the slope of the curve is composed of the slope for the Vogel curve and the constant PI curve.
3. When the well produces liquid rates greater than its maximum oil rate q_{omax} , a straight line is used because the production is mostly water.

In the following, the calculation of $FBHP$ s for given liquid rates is described while using the **composite IPR curve** principle. This enables one to construct the well's IPR curve and to find $FBHP$ s for any rate or to find the liquid rates belonging to any $FBHP$ value. The construction of the IPR curve is done differently in each of the intervals introduced previously.

For water cuts of 100% or for liquid rates less than the rate valid at the bubble-point pressure the following formula can be used:

$$FBHP = SBHP - \frac{q_l}{PI} \quad (2.4)$$

$FBHP$ s for water cuts less than 100% and liquid rates less than the well's maximum oil rate are found from:

$$FBHP = f_w \left(SBHP - \frac{q_l}{PI} \right) + 0.125 f_o p_b \left(-1 + \sqrt{81 - 80 \frac{q_l - q_b}{q_{\text{omax}} - q_b}} \right) \quad (2.5)$$

Finally, for water cuts less than 100% and liquid rates greater than the well's maximum oil rate, $FBHP$ is calculated from the expression:

$$FBHP = f_w \left(SBHP - \frac{q_{\text{omax}}}{PI} \right) - (q_l - q_{\text{omax}}) \text{slope} \quad (2.6)$$

where:

$FBHP$ = flowing bottom-hole pressure, psi

$SBHP$ = static bottom-hole pressure, psi

PI = productivity index, bpd/psi

q_l = liquid flow rate, bpd

f_w = water cut, —

f_o = produced oil fraction, —

p_b = bubble-point pressure, psi

The previous formulas include the following unknown parameters:

PI , the slope of the straight portion of the IPR curve

q_b , the liquid rate at the bubble-point pressure

q_{omax} , the well's maximum oil rate

slope , the slope of the IPR curve at liquid rates greater than q_{omax}

These parameters are evaluated from production well tests providing liquid rates at different $FBHP$ s. Their determination, based on a single well test,

depends on the relation of the actual *FBHP* and the well's *SBHP* to the bubble-point pressure and is accomplished as detailed in the following.

Case 1: *SBHP* and Test Pressures above the bubble-point pressure.

The well's *PI* is easily found from Eq. (2.2) as:

$$PI = \frac{q_{\text{test}}}{SBHP - FBHP_{\text{test}}} \quad (2.7)$$

The liquid rate valid at a bottom-hole pressure equal to the bubble-point pressure is calculated as given here:

$$q_b = PI(SBHP - p_b) \quad (2.8)$$

The maximum oil rate, q_{omax} , when producing 100% oil is found from:

$$q_{\text{omax}} = q_b + \frac{PI p_b}{1.8} \quad (2.9)$$

The slope of the composite curve's linear section is evaluated from:

$$\text{slope} = \frac{f_w \frac{0.001 q_{\text{omax}}}{PI} + 0.125 f_o p_b \left(-1 + \sqrt{81 - \frac{80(0.999 q_{\text{omax}} - q_b)}{q_{\text{omax}} - q_b}} \right)}{0.001 q_{\text{omax}}} \quad (2.10)$$

Finally, the well's maximum liquid production rate, q_{lmax} , is found from:

$$q_{\text{lmax}} = q_{\text{omax}} + f_w \frac{SBHP - \frac{q_{\text{omax}}}{PI}}{\text{slope}} \quad (2.11)$$

With the previous parameters known, *FBHPs* for any rate are evaluated from Eqs. (2.4)–(2.6), resulting in the points of the **composite IPR curve**.

Case 2: *SBHP* above and Test Pressure below the bubble-point pressure.

Since the point belonging to the production test does not fall on the linear portion of the IPR Curve, the *PI* can only be determined by a trial-and-error procedure. The iterative determination of the proper *PI* value is facilitated by assuming the following initial value:

$$PI_{\text{initial}} = \frac{q_{\text{test}}}{f_o \left\{ SBHP - p_b + \frac{p_b \left[1 - 0.2 \frac{FBHP_{\text{test}}}{p_b} - 0.8 \left(\frac{FBHP_{\text{test}}}{p_b} \right)^2 \right]}{1.8} \right\} + f_w (SBHP - FBHP_{\text{test}})} \quad (2.12)$$

Based on this *PI*, the calculation sequence detailed for **Case 1** is followed and first the parameters q_b , q_{omax} , and *slope* are determined. Based on those, the flowing bottom-hole pressure belonging to the test rate q_{test} is found. If the calculated value differs from the measured *FBHP*, a new iteration step is required. Usually, only a few iterations are needed to arrive at the proper *PI*

value. Based on the final value, the well's **composite IPR curve** is evaluated from Eqs. (2.4)–(2.6).

Case 3: *SBHP* and Test Pressures below the bubble-point pressure.

Again, iteration is required to arrive at the proper *PI* value and the use of the following initial value is advised:

$$PI_{\text{initial}} = \frac{q_{\text{test}}}{f_o \frac{SBHP \left[1 - 0.2 \frac{FBHP_{\text{test}}}{SBHP} - 0.8 \left(\frac{FBHP_{\text{test}}}{SBHP} \right)^2 \right]}{1.8} + f_w (SBHP - FBHP_{\text{test}})} \quad (2.13)$$

The calculation steps for the construction of the **composite IPR curve** are identical to those detailed for **Case 2** with the following modification: in the relevant formulas the substitutions $p_b = SBHP$ and $q_b = 0$ must be used.

Example 2.3

Calculate the points of the composite IPR curve for a well that produces a liquid rate of 150 bpd at a flowing bottom-hole pressure of 2,200 psi and has a static *BHP* of 2,500 psi. The produced oil's bubble-point pressure at bottom-hole conditions is 2,100 psi and the producing water cut equals 60% ($f_w = 0.6$).

Solution

Since both the *SBHP* and the $FBHP_{\text{test}}$ values are greater than the bubble-point pressure, calculations follow those described for **Case 1**.

The *PI* of the well is found from Eq. (2.7), and using the measured test data we get:

$$PI = 150 / (2,500 - 2,200) = 0.5 \text{ bpd/psi.}$$

The liquid rate at the bubble-point pressure is found from Eq. (2.8):

$$q_b = 0.5(2,500 - 2,100) = 200 \text{ bpd.}$$

The maximum oil rate, as found from Eq. (2.9), equals:

$$q_{\text{omax}} = 200 + 0.5 \cdot 2,100 / 1.8 = 783 \text{ bpd.}$$

The slope of the linear portion of the curve is evaluated from Eq. (2.10):

$$\begin{aligned} \text{slope} = & \left\{ 0.6 \cdot 0.001 \cdot 783 / 0.5 + 0.125 \cdot 0.4 \cdot 2100 \right. \\ & \left. \left[-1 + (81 - 80(0.999 \cdot 783 - 200) / (783 - 200))^{0.5} \right] \right\} \\ & / (0.001 \cdot 783) = 8.22. \end{aligned}$$

After these preliminary calculations, points on the three sections of the composite IPR curve can be evaluated. In the following, detailed calculations for only one point in each interval are presented.

Above the bubble point, take the liquid rate of 100 bpd and find the corresponding *FBHP* from Eq. (2.4) as follows:

$$FBHP = 2,500 - 100/0.5 = 2,300 \text{ psi.}$$

In the next interval, take a rate of 600 bpd to find the *FBHP* required from Eq. (2.5):

$$FBHP = 0.6(2,500 - 600/0.5) + 0.125 \cdot 0.4 \cdot 2,100$$

$$\left\{ -1 + [81 - 80(600 - 200)/(783 - 200)]^{0.5} \right\} = 1,212 \text{ psi.}$$

Finally, take a liquid rate $q_l = 820$ bpd, greater than the maximum oil rate, and find the corresponding *FBHP* from Eq. (2.6):

$$FBHP = 0.6(2,500 - 783/0.5) - (820 - 783)8.22 = 256.2 \text{ psi.}$$

2.3 HYDRAULIC FUNDAMENTALS

2.3.1 Tubing Flow Calculations

Because the fluid discharged by the electrical submersible pump enters the tubing string, the discharge pressure of the pump should overcome the sum of pressure losses occurring along the flow path at the actual fluid production rate. The components of the necessary discharge pressure are the following:

1. The **wellhead pressure** valid at the given liquid production rate. It is found from the separator pressure and the pressure losses along the flow line.
2. The **net hydrostatic pressure** acting on the pump. Because there is always a liquid column present in the annulus above the pump, the net pressure is calculated from the true vertical depth of the **dynamic liquid level** and the pressure gradient of the produced fluid.
3. The **frictional pressure drop** occurring in the tubing string valid at the given flow rate. When calculating the frictional losses, the measured depth of the pump is to be used in the **Darcy–Weisbach equation** along with the friction factor found from the **Moody diagram**.

In the ESP industry, it is customary to calculate the above components in **head** units where head is measured in length units (ft) and is expressed from pressure as follows:

$$H = \frac{2.31 p}{\gamma_l} \quad (2.14)$$

where:

H = head, ft

p = pressure, psi

γ_l = specific gravity of the produced liquid, –

For estimating the frictional head losses in well tubing, most of the ESP manufacturers and other sources [3–5] use the **Hazen–Williams formula** reproduced below:

$$\Delta h_{fr} = 0.2083 \left(\frac{100}{C} \right)^{1.85} \frac{q^{1.85}}{ID^{4.86}} \quad (2.15)$$

where:

Δh_{fr} = frictional head loss, ft/100 ft

C = pipe quality number, —

q = flow rate, gpm

ID = pipe inside diameter, in

The above formula is valid for water flow and can be used for calculating head losses in new or used pipes by substituting the pipe quality numbers of $C = 120$ or $C = 94$, respectively.

Fig. A.1 in Appendix A contains a chart based on the **Hazen–Williams formula**, which can be used to estimate frictional head losses occurring in selected API tubing and casing sizes. Bold lines refer to **new** piping; normal lines refer to **used** pipes.

Example 2.4

Calculate the head to be developed by a centrifugal pump set in a vertical well at 6,000 ft, producing an oil of specific gravity = 0.85, at a rate of 2,000 bpd. The liquid level in the annulus is at the pump, tubing head pressure is 400 psi, and tubing size is $2\frac{7}{8}$ in.

Solution

The total head to be developed by the pump should overcome the individual heads corresponding to (1) the wellhead pressure, (2) the net hydrostatic pressure in the tubing string, and (3) the frictional losses in the tubing string.

The head to overcome the wellhead pressure is found from Eq. (2.14):

$$H_1 = 2.31 \, 400 / 0.85 = 1,087 \text{ ft.}$$

The hydrostatic pressure of the 6,000-ft oil column, expressed in head equals $H_2 = 6,000$ ft.

The estimated frictional head loss is found from Fig. A.1 in Appendix A as:

$$\Delta h_{fr} = 36 \text{ ft} / 1,000 \text{ ft.}$$

Using the above value, the total head loss in the tubing is calculated as:

$$H_3 = (36 \, 6,000) / 1,000 = 216 \text{ ft.}$$

The pump's total head, therefore, should equal the sum of the above components:

$$H = 1,087 + 6,000 + 216 = 7,303 \text{ ft.}$$

2.3.2 Electrical Submersible Pumps

Pumps used in the petroleum industry for service on the surface or in a well can be classified into two broad groups: (1) **displacement** (or positive displacement) and (2) **dynamic** pumps [6]. Pumps in sucker-rod pumping or in PCP (progressing cavity pump) installations belong to the first group, whereas the submersible pumps of ESP installations work on the dynamic principle. Within ESP pumps, driven by electric motors, the kinetic energy of the fluid is increased first to be then partly converted to pressure energy that moves the fluid through the pump.

Submersible pumps in ESP service operate with their shafts in the vertical position and are centrifugal pumps with the following main features:

- There are **multistage pumps** with several tens or hundreds of stages connected in series.
- Their impellers are of the **closed** vane design.
- They have a single suction side.
- They are self-priming.
- They have radial or mixed flow configurations.

The technical features listed here are described in the following sections.

2.3.2.1 Operational Basics of Centrifugal Pumps

A single-stage centrifugal pump is a simple machine driven by a prime mover providing a rotary motion and consists of two basic components: (1) the **rotor**, a rotating set of vanes, and (2) the **stator**, the stationary part containing the casing of the rotor, as well as the bearings and seals required for proper operation. ESP pumps use fully enclosed curved **vaness** in their **impellers**, and their stationary parts are called **diffusers**.

Fig. 2.5 depicts one stage of a common multistage centrifugal pump (so-called mixed flow pump) in ESP service. Liquid from the previous stage enters the impeller in an **axial** direction at a relatively low velocity and, because of the high rotational speed of the impeller's vanes, attains a high velocity at the impeller's discharge. Thus, the torque applied by the prime mover to the pump is converted to **kinetic energy** by the vanes. The high-velocity liquid stream discharged at an angle from the horizontal then enters the vane channels of the **diffuser** (the pump's stationary part) where conversion of kinetic energy to pressure energy takes place. The liquid leaving the pump stage at the discharge of the diffuser, therefore, is at a higher pressure level than it was at

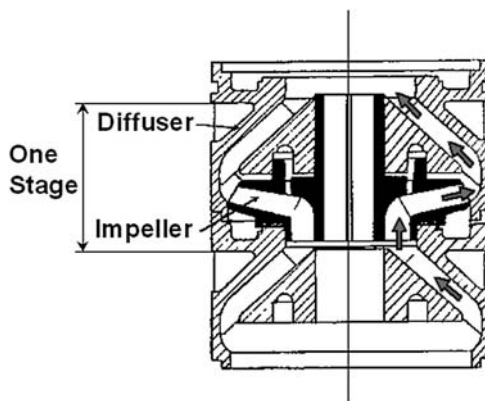


FIGURE 2.5 One stage of a mixed flow centrifugal pump.

the inlet to the impeller, the operation of the stage having increased the flowing pressure. Because the discharge of any stage is led to the intake of the next stage, the process repeats and the pressure of the liquid pumped is accordingly increased.

The energy conversions inside the centrifugal pump described above are governed by the **General Energy Equation** describing the **conservation of energy** between two points in a flowing fluid. This states that the change in the fluid's energy content is equal to the work done on the fluid. The fluid flowing in the pump may have three forms of energy: **potential**, **kinetic**, and **pressure** energy, of which the change in the potential energy due to elevation change is neglected because of the negligible vertical distances in the pump stage. Thus, the sum of pressure and kinetic energies must be **constant** and the energy input from the prime mover is finally converted to an increase in fluid pressure.

Centrifugal pumps can be classified according to the direction of the impeller's discharge and can belong to the **radial**, **axial**, or **mixed flow** groups. In ESP service, however, only radial flow and mixed flow pumps are used; they are illustrated in Fig. 2.6. Radial flow is used in smaller capacity pumps producing less than 3,000 bpd liquid, with mixed flow pumps used at higher rates.

As discussed previously, the liquid's energy content is increased by the rotary action of the impeller's vanes. A radial discharge (aka **pancake**) impeller with partly removed top shroud is shown in Fig. 2.7, where the arrangement of the **vanes** can be seen. The operation of the **fully enclosed** vanes in an impeller of the radial discharge type is illustrated in Fig. 2.8. Liquid from the axial intake enters the impeller and is rotated by the vanes contained in the impeller, and it leaves the impeller with a greatly increased velocity. The figure shows an impeller with **backward curved vanes**, mostly used in ESP pumps, but radial and forward curved vanes are also possible. It should be noted that the cross-sectional area available for liquid flow between

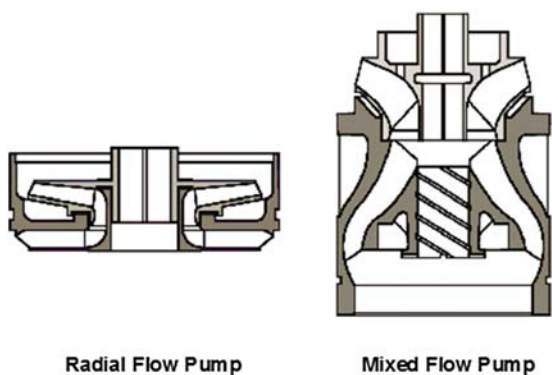


FIGURE 2.6 Radial and mixed flow pump stages.



FIGURE 2.7 Impeller with top shroud partly removed.

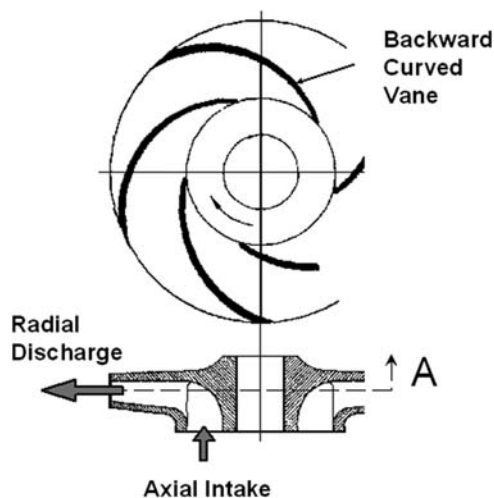


FIGURE 2.8 Liquid flow path and the vanes of an impeller.

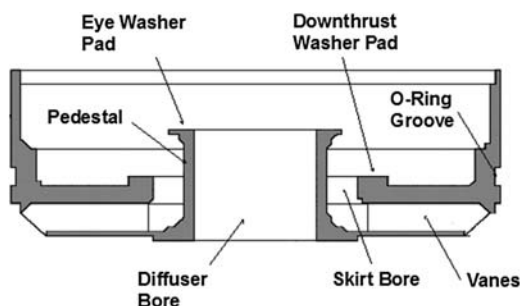


FIGURE 2.9 Parts of a radial flow diffuser.

two successive vanes increases progressively as the liquid particles are moved toward the discharge. This indicates that conversion of kinetic energy to pressure energy first occurs in the impeller, to be completed in the diffuser of the pump stage.

The structure of the **diffuser** of a radial flow (aka **pancake**) pump stage is depicted in Fig. 2.9. The diffuser **bore** accepts from below the **hub** of the impeller, and the next upper stage's impeller is fitted in the skirt bore. The diffuser, like the impeller, also contains **vanes** and the channels between the vanes are designed in such a way that conversion of kinetic energy to pressure takes place in them because of their increasing cross-sectional area in the direction of fluid flow. Diffusers are contained within the **pump housing** and the required number of stages is reached by stacking the right number of diffusers and impellers on top of each other.

The construction and nomenclature of radial flow **impellers** is given in Fig. 2.10. The pump shaft runs in the bore of the **hub** and turns the impeller with the help of **keys** fitted into the key-way of the impeller. In case of **floating impellers**, the impeller is free to move on the shaft in the axial direction but is axially fixed to the shaft in pumps with the so-called **fixed impellers**. The pump's vanes are situated between the two disks (the top and bottom **shrouds**) of the impeller. Upthrust and downthrust **washers** (usually made of laminated plastic material) are provided on the top and the bottom contact surfaces to the

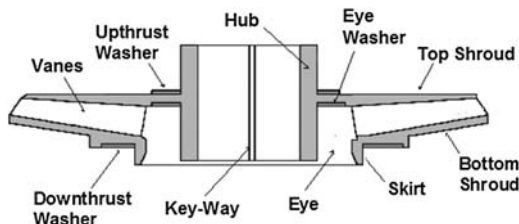


FIGURE 2.10 Parts of a radial flow impeller.

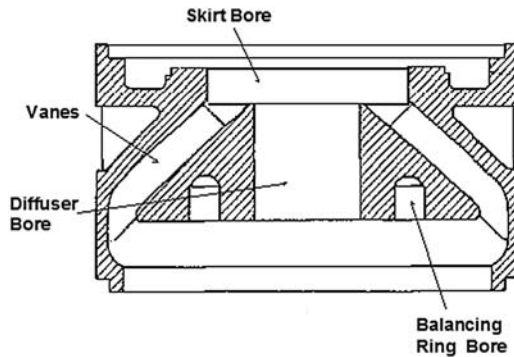


FIGURE 2.11 Nomenclature of mixed flow diffuser parts.

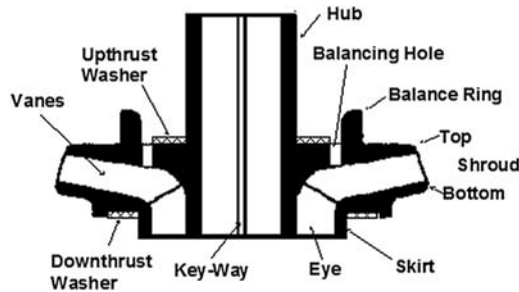


FIGURE 2.12 Nomenclature of mixed flow impeller parts.

diffuser to ensure low frictional forces between the moving impeller and the stationary diffuser.

The construction of mixed flow pump stages is somewhat different from that of radial pumps; typical diffusers and impellers are shown in Figs. 2.11 and 2.12. Quite frequently, balancing mechanisms are provided to decrease the axial forces developing in the pump stage, which include **balance rings** and **balancing holes**. Note that the balancing holes connect the inlet and the discharge sides of the impeller causing a definite drop in pump efficiency and a decline of pressure increase per stage, but at the same time, they effectively reduce the magnitude of axial forces. The importance of axial force reduction will be discussed in a later section.

2.3.2.2 Specific Speed

To properly compare centrifugal pumps of different designs, the concept of **specific speed**, N_s , was introduced. It is defined as the rotational speed (RPM) required to produce a liquid rate of 1 gallon per minute against 1 ft of head with an impeller geometrically similar the investigated one. The formula to

calculate specific speed is the following, where all parameters are taken at the **best efficiency point** (BEP) of the pump:

$$N_s = \frac{N\sqrt{Q}}{H^{0.75}} \quad (2.16)$$

where:

N_s = pump specific speed, RPM

N = pump speed, RPM

Q = pumping rate, gpm

H = head developed by one stage, ft

Specific speed has no practical importance and is only used to compare different centrifugal pump designs. Its value characteristically changes with the type of the pump: radial discharge pumps usually fall in the range $500 < N_s < 1,800$, and mixed flow discharge pumps can reach a maximum around $N_s = 4,500$.

Impeller design and specific speed are in close correlation. Purely **radial** impellers have low N_s values and develop their head mostly because of the conversion of centrifugal force. **Mixed flow** impellers with higher specific speeds operate only partly on the centrifugal force, and more and more of the total head is developed by the lifting action of the impellers. Purely **axial impellers** have the highest specific speeds, and practically all their head is generated by the lifting action of the “propellers.”

2.3.2.3 Pump Performance

As it was shown in the previous sections, a centrifugal pump transforms the mechanical energy (input from the motor driving it) into the kinetic energy of the liquid transported. Because kinetic energy is proportional to the term ρv^2 (density times velocity squared), a centrifugal pump running at a given speed and thus having a constant discharge velocity from its impeller transmits different amounts of energy to liquids of different **density**. Therefore, after converting kinetic energy into pressure, the pressure increase in one stage or in a complete pump **increases** with the density of the liquid being pumped. Putting it another way, the pressure increase developed by the pump divided by liquid density, called **head**, is a constant for a given pump and a given liquid rate. This is the reason why in dealing with centrifugal pumps, head is used instead of pressure in all calculations involving the performance and use of centrifugal pumps. Although the pressure increase developed by any pump depends on the **density** of the liquid being pumped, the head developed is **constant** for any liquid.

The **head developed** by an impeller under ideal operating conditions (neglecting frictional and other losses) is calculated from the increase in centrifugal force acting on the liquid contained between two successive vanes, by assuming an infinite number of vanes [7]. The result of the above derivation

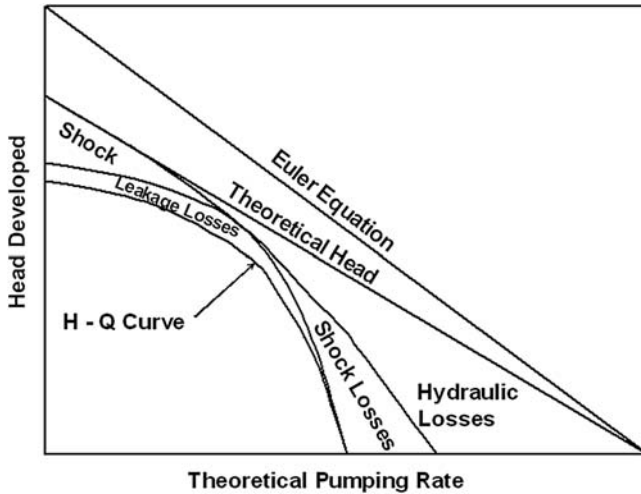


FIGURE 2.13 Derivation of a pump's H-Q curve.

is the famous **Euler equation**, shown in function of the liquid rate across the pump as a straight line in Fig. 2.13. However, in real centrifugal pumps the number of vanes is limited and the liquid contained between two vanes develops a circulating flow which, in turn, modifies the velocity distribution and the theoretical head developed, as shown in the figure.

The actual head developed by the pump is always less than the theoretical one, the difference being consumed by different losses discussed below:

- **Hydraulic losses** progressively increase with liquid rate and are due to fluid friction in the impeller, as well as diffusion losses in the impeller channel.
- **Shock losses** are negligible at the BEP of the pump stage but increase at lower and higher liquid rates. They occur at the entrance and the exit of the impeller and are caused by sudden changes in the direction of flow.
- **Leakage losses** represent the loss of liquid rate through the clearances between the rotating and stationary parts of the pump stage (at the impeller eye, through balancing holes, etc.). They reduce the produced head at any liquid rate because the actual flow rate across the impeller is always greater than the output from the stage. Leakage losses diminish with increased liquid rates, as shown in Fig. 2.13.

The performance of real centrifugal pump stages, after considering all of the above described head losses, is represented by the **H-Q curve** schematically presented in Fig. 2.13. As seen, the head (H) developed by the stage starts from the **shut-off head** valid at zero flow rate and continuously decreases with the liquid rate (Q) across the pump but is independent of the density of the liquid pumped.

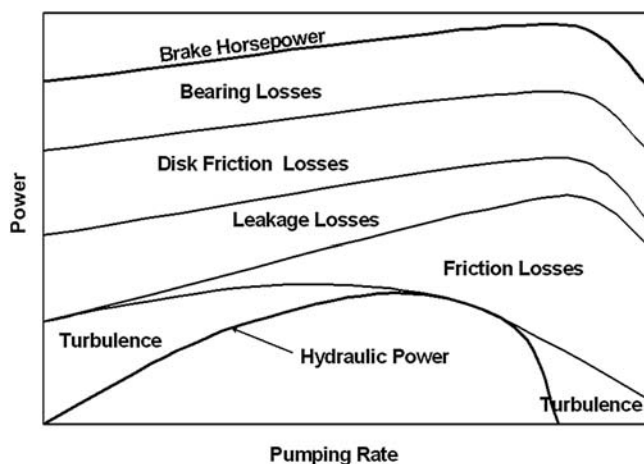


FIGURE 2.14 Power conditions in a centrifugal pump stage.

The power conditions in a centrifugal pump are schematically depicted in Fig. 2.14. The power required to drive the pump is represented by the **brake horsepower**, which must overcome the energy needed to pump the given liquid rate plus all the energy losses occurring in the pump. The **useful hydraulic power** is proportional to the product of the head, the liquid capacity, and the liquid density and can be calculated from the pump's **H–Q curve** at any liquid rate. It is zero at two points: at a liquid rate of zero and at the pumping rate where the head is zero. Turbulent losses are considerable at low and high rates but diminish toward the BEP of the pump. Frictional energy losses in the impeller channels proportionally increase with the liquid rate, and liquid leakage in the pump stage consumes an almost constant energy. Disk friction and bearing losses do not change much with pumping rate and cover the energy requirements for rotating the impeller in the liquid and the mechanical friction losses in the pump's bearings and stuffing boxes, respectively.

The centrifugal pump's energy efficiency can be derived from the **brake horsepower** required to drive the pump and the hydraulic power spent on liquid transfer, by dividing the useful (hydraulic) power by the brake horsepower. If plotted in function of the liquid rate, the **pump efficiency** curve follows the shape of the hydraulic power, as shown in the schematic diagram in Fig. 2.15. This figure depicts the pump performance curves usually supplied by manufacturers of submersible equipment.

Centrifugal pumps are **tested** by running them at a constant speed while varying the pumping rate by **throttling** the flow at the pump discharge. During testing, the following parameters are measured with the proper accuracy: liquid flow rate, suction and discharge pressures, and the brake horsepower required to drive the pump. Pump efficiency is calculated from these data and

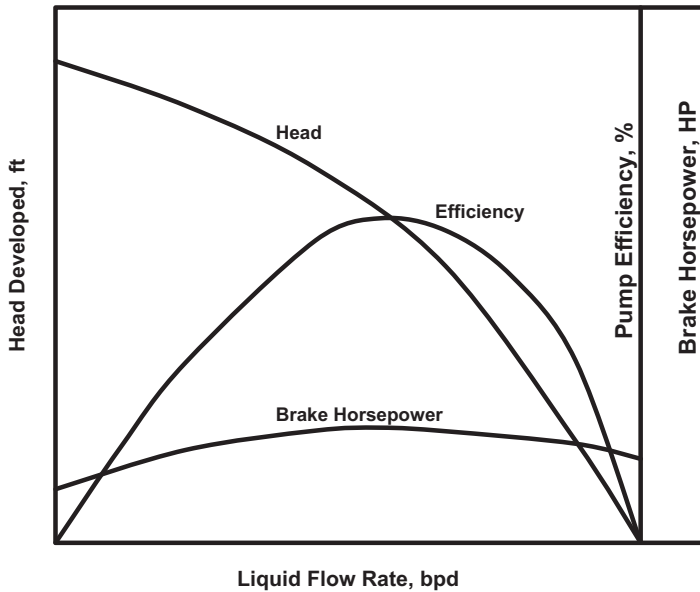


FIGURE 2.15 Schematic pump performance curves.

represents the effectiveness of the pump to convert **mechanical** energy input to the pump into **hydraulic** power used to move the fluid. Based on measurements at different flow rates, the following three parameters should be plotted versus pumping rate:

1. the **head** developed by the pump,
2. the **brake horsepower** required to drive the pump when pumping water, and
3. the pump's power **efficiency**.

As recommended by the **API RP 11S2** [8], performance tests must be based on pumping freshwater (specific gravity = 1.0) at a temperature of 60°F and at a rotational speed of $\text{RPM} = 3,500$ or $\text{RPM} = 2,917$, for 60 and 50 Hz service, respectively. If actual tests are performed with other fluids or speeds, the results must be corrected to standard conditions. Note that head and efficiency do not change with liquid **density** but brake horsepower does. Although tests may be performed on any number of stages, performance curves must represent the operation of one or a specified number of pump stages.

2.3.2.4 Cavitation

When pumping liquids, the pressure at any place inside the pump must not be allowed to fall below the **saturated vapor pressure** of the liquid valid at the operating temperature because cavitation occurs. This involves the forming of small vapor bubbles, which, when taken by the flowing liquid to places at

higher pressure, will suddenly collapse. Collapse of vapor bubbles causes extremely great and sudden increases in pressure and results in mechanical destruction due to **erosion** of the metal parts.

Cavitation can occur in the ESP pump impeller's eye where a great increase in velocity takes place. This involves (according to **Bernoulli's law**) a sudden decrease in flowing pressure and if this pressure falls below the vapor pressure valid at flowing temperature, small vapor bubbles in the liquid stream may form. The harmful consequences of cavitation significantly affect the performance of the centrifugal pump; pump capacity as well as pump efficiency is drastically reduced.

The minimum pressure required at suction conditions to prevent the occurrence of cavitation is known as the **net positive suction head (NPSH)**, and is a measured parameter of each centrifugal pump. Actual NPSH for submersible pumps is the difference between the absolute casing pressure available at the first-stage impeller and the vapor pressure. To prevent cavitation, the actual NPSH must be greater or equal to the minimum NPSH value for the given pump. Although a big problem for centrifugal pumps in general, submersible centrifugal pumps under oilfield conditions usually do not experience cavitation because an adequately great suction pressure is always available due to the presence of a sufficient length of liquid column above the pump intake.

2.3.2.5 Axial Thrust Forces

During the pumping action, various unbalanced forces arise on the centrifugal pump's impeller and these forces are directly transmitted to the pump shaft. The radial components of these forces are taken up by the housing of the ESP pump and do not significantly affect the proper operation. Axial force components, however, are much more detrimental and, if not taken up by **thrust bearings** of the proper capacity, can induce the axial movement of impellers and the pump shaft as well, resulting in mechanical **damage** of the pump stages. This is the reason why in all centrifugal pumps, especially in the submersible versions with several tens or hundreds of stages, provisions must be provided to minimize and compensate axial forces.

Axial forces in ESP pumps can be classified into two groups: static and dynamic forces. **Static forces** arise because of the weight of the pump parts in the produced fluid and act downward:

- the weight of the **impellers** and
- the weight of the pump **shaft**.

Dynamic forces are the results of the pumping action and are related to the flow of the produced liquid through the pump's stages. They take the following forms:

- Forces resulting from the suction and discharge pressures acting on the two shrouds of the impeller. The net of these forces always acts downward.

- The net **inertial force** due to the changes in flow direction inside the pump stage. Because the velocity of the produced fluid inside the impeller is much higher at the discharge side than at the suction, this force always points upward.
- The axial load due to the pump discharge pressure acting on the cross-sectional area of the pump shaft, acting downward.

The dynamic axial forces arising in a conventional mixed flow impeller are illustrated in Fig. 2.16.

As shown, the hydraulic forces acting on the different surfaces of the top and bottom shrouds are greater toward the periphery of the impeller because of the rotation of the fluid. Forces exerted by the discharge pressure are partially **balanced** because they oppose each other. Suction pressure, at the other hand, gives rise to an upward-pointing force. As stated before, the sum of these forces gives a downward pointing net force.

In addition to this, an **inertial force** arises because of the axial change in the momentum of the flowing fluid between the discharge and suction conditions. This upward-pointing component increases the net axial forces.

The sum of the previous forces gives the dynamic components of the impeller's **axial thrust**, which, depending on the impeller design and the flow rate, can be zero (**balanced** state) and can point upward (**upthrust** condition) or downward (**downthrust** condition).

For the proper operation of a centrifugal pump, the reduction of axial thrust in pump stages is of great importance and can be accomplished in different ways. A common solution uses a **balancing ring** and **balancing holes** drilled

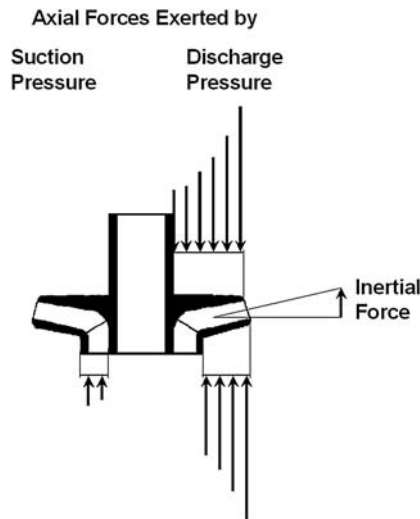


FIGURE 2.16 Distribution of axial forces in a mixed flow impeller.

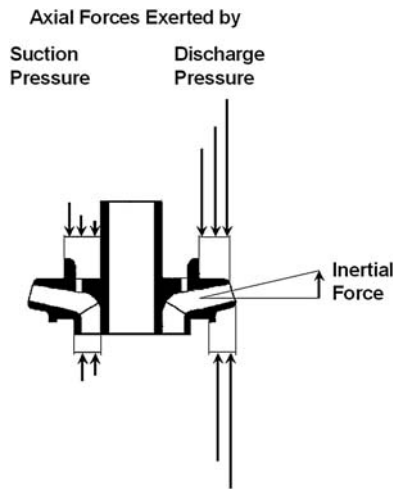


FIGURE 2.17 The effect of balancing holes on axial forces.

in the top shroud of the impeller. These modifications reduce the volumetric efficiency of the stage, but at the same time they greatly reduce the axial forces acting on the impeller, as shown in Fig. 2.17.

Submersible pump impellers are designed to be balanced near the BEP of the stage. To ensure ideal conditions, ESP pumps must be operated at the liquid rate belonging to that point. Because in real applications this is almost impossible to accomplish, there is always an **axial thrust** acting on the pump shaft that must be taken care of.

The schematic variation of axial thrust with the pumping rate in a typical submersible pump stage is given in Fig. 2.18, according to Coltharp [9]. Thrust is at a maximum at a flow rate of zero (**shut-off condition**) and points downward and then continuously decreases for increased liquid rates. After having reached the balanced condition, the thrust changes its sign and enters the **upthrust** region, continuously increasing at greater flow rates. The **operating range** of the pump is selected by the pump designer to ensure the effective compensation of actual forces by properly designed thrust bearings situated in the pump and in the protector section.

2.3.2.6 Affinity Laws

Most centrifugal pumps, either in surface or in downhole service, operate at a constant **speed** because they are usually driven by electric motors connected to a network supplying an electric current with a **constant frequency**. However, their performance at different speeds has long been investigated and the basic rules of variable speed operation have been established [10]. According to the

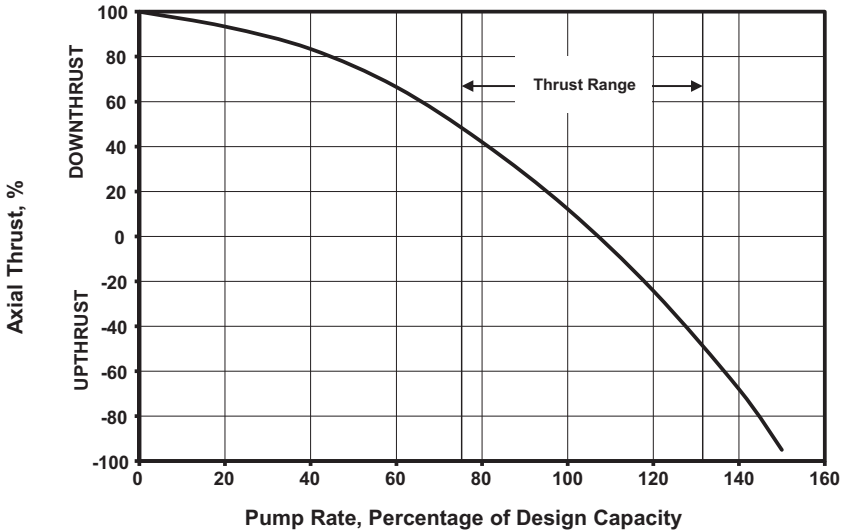


FIGURE 2.18 Variation of axial thrust with the flow rate [9].

affinity laws, the following relationships exist between the actual speed of the centrifugal pump and its most important performance parameters:

- The flow rate of a pump changes directly **proportional** to its operating speed.
- The head developed by the pump changes proportionally to the **square** of the speed.
- The brake horsepower required to drive the pump changes proportionally to the **cube** of the speed.
- The efficiency of the pump does not change with speed.

The above rules can be expressed mathematically as follows:

$$Q_2 = Q_1 \left(\frac{N_2}{N_1} \right) \quad (2.17)$$

$$H_2 = H_1 \left(\frac{N_2}{N_1} \right)^2 \quad (2.18)$$

$$BHP_2 = BHP_1 \left(\frac{N_2}{N_1} \right)^3 \quad (2.19)$$

where:

N_2, N_1 = pumping speeds, RPM

Q_2, Q_1 = pumping rates at N_2 and N_1 , bpd

H_2, H_1 = developed heads at N_2 and N_1 , ft

BHP_2, BHP_1 = required brake horsepowers at N_2 and N_1 , HP

The **affinity laws**, as given above, are written up using the pumping speed, N , but speed can be substituted by the electrical **frequency**, f , supplied to the motor that drives the pump. This is justified by the fact that the speed of induction motors used in ESP service is directly **proportional** to the frequency of the electric power (see Eq. 2.28). Therefore, the above formulas are often used with the electric frequency in place of the pumping speed.

To illustrate the effects of speed changes on the head performance of a centrifugal pump, Fig. 2.19 is presented where **H–Q curves** for different electrical frequencies are plotted for an example pump. As seen, the head developed by the ESP pump increases for increasing, and decreases for decreasing, frequencies. The chart, sometimes called a “**tornado curve**,” includes curves that connect the points having the same pump efficiency values on the different head curves. Curves connecting the BEPs as well as the minimum and maximum limits of the recommended operating rates for the different frequencies are also plotted. Note that pump efficiencies are constant on these curves demonstrating the previous statement that pump efficiency does not change with pump speed.

It can be observed that higher frequencies (pumping speeds) move the BEPs to greater liquid flow rates; at the same time the recommended operating liquid rate range (the difference between the maximum and minimum rate limits) of the pump increases. This behavior allows a simple adjustment of the operating range of any given pump in wells of unknown or rapidly changing inflow performance if pumps driven by **variable speed drives**

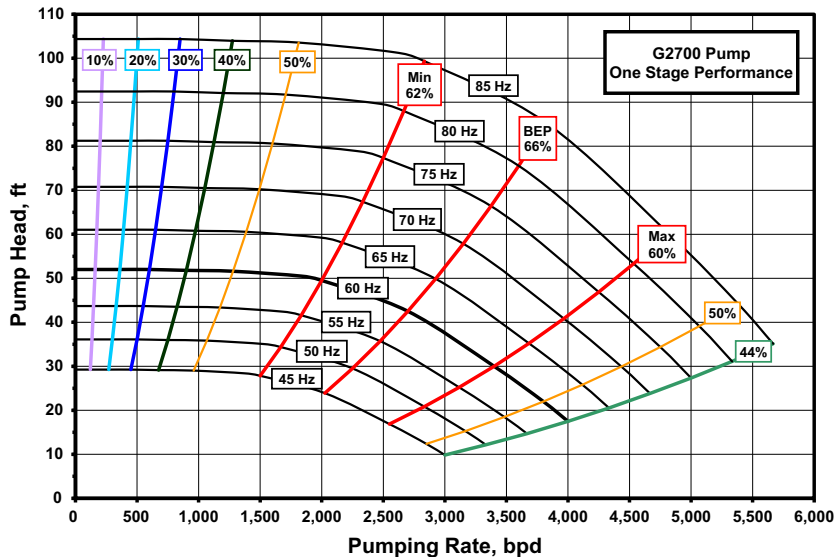


FIGURE 2.19 H–Q curves of a pump at different speeds.

(VSDs) are used. Utilization of VSD units in such cases eliminates the need to exchange the pump to one for which operating range covers the well's required liquid rate.

Example 2.5

The BEP of one stage of a centrifugal pump is at a rate of 1,200 bpd, where the head is 52 ft, and the required brake horsepower is 1 HP. The pump operates at a speed of 3,500 RPM (60 Hz service); find its parameters at 50 Hz for a speed of 2,917 RPM.

Solution

The new rate at the new speed is found from Eq. (2.17):

$$Q_2 = 1,200(2,917/3,500) = 1,000 \text{ bpd.}$$

The new head and brake horsepower are calculated from Eqs. (2.18) and (2.19), respectively:

$$H_2 = 52(2,917/3,500)^2 = 36 \text{ ft.}$$

$$BHP_2 = 1(2,917/3,500)^3 = 0.6 \text{ HP.}$$

Using electrical frequency instead of pumping speed in the **affinity law** formulas, the calculations are as follows:

The new rate at the new frequency is found as:

$$Q_2 = 1,200(50/60) = 1,000 \text{ bpd.}$$

The new head and brake horsepower are calculated:

$$H_2 = 52(50/60)^2 = 36 \text{ ft.}$$

$$BHP_2 = 32(50/60)^3 = 0.6 \text{ HP.}$$

As seen, the results are identical.

The **affinity laws** permit the creation of pump performance curves at any rotational speed from curves known at a given speed. When converting the **head** performance curve (the most important parameter of the centrifugal pump) from one pumping speed to another, combination of Eqs. (2.17) and (18) results in the following formula:

$$H_2 = H_1 \left(\frac{Q_2}{Q_1} \right)^2 \quad (2.20)$$

Because the values H_1 and Q_1 are constant, the formula can be modified to:

$$H_2 = k Q_2^2 \quad (2.21)$$

where:

H_2 = developed head at N_2 , ft

Q_2 = pumping rate at N_2 , bpd

k = constant found from Eq. (2.20)

The plot of this function in the H versus Q coordinate system is a second-order **parabola** started from the origin and running through the H_1 – Q_1 point valid for the original speed. Many such curves can be constructed, and they are called “**affine parabolas**”; they connect the corresponding operating points on the head performance curves that belong to the different speeds. The “tornado curve” presented in Fig. 2.19 contains these curves, which, in addition to the just described feature, represent constant pump efficiencies.

If the operation of centrifugal pumps with different impeller sizes is compared at different pumping speeds, the **affinity** laws are modified and include the impeller diameters as well. As before, instead of the pumping speeds also the electrical frequencies can be used in the formulas given here.

$$Q_2 = Q_1 \left(\frac{N_2}{N_1} \right) \left(\frac{d_2}{d_1} \right) \quad (2.22)$$

$$H_2 = H_1 \left(\frac{N_2}{N_1} \right)^2 \left(\frac{d_2}{d_1} \right)^2 \quad (2.23)$$

$$BHP_2 = BHP_1 \left(\frac{N_2}{N_1} \right)^3 \left(\frac{d_2}{d_1} \right)^3 \quad (2.24)$$

where:

N_2, N_1 = pumping speeds, RPM

Q_2, Q_1 = pumping rates at N_2 and N_1 , bpd

H_2, H_1 = developed heads at N_2 and N_1 , ft

d_2, d_1 = impeller diameters, in

BHP_2, BHP_1 = required brake horsepowers at N_2 and N_1 , HP

2.4 ELECTRICAL FUNDAMENTALS

2.4.1 Alternating Current

Most electric motors used to drive ESP installations are AC (alternating current) motors, i.e., they rely on a supply of alternating electric voltage. Alternating voltage or AC reverses its direction from positive to negative and alternates between constant positive and negative values. In contrast to direct current, AC permits the use of **transformers** to change its voltage to any desired level. This feature is exploited in long-distance **transmission** of electric power when the voltage of the generated electricity is raised to very high values, thus effectively reducing the size of the required transmission wires and the power losses occurring. At the consumer end, another transformer is used to reduce the high voltage to the desired value before it is used.

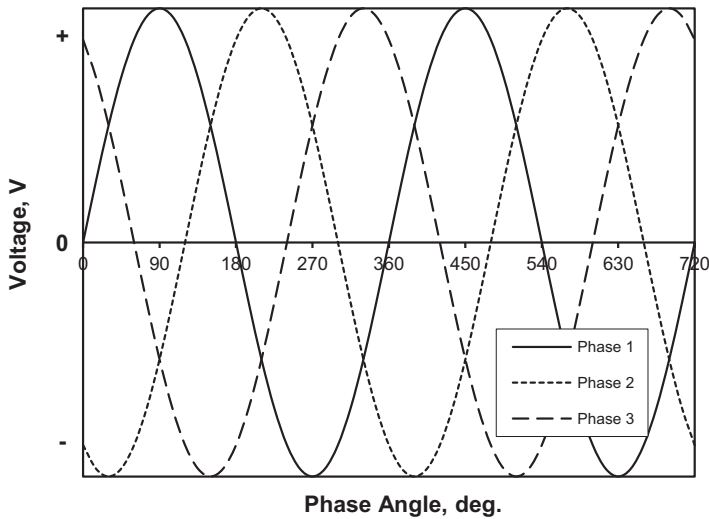


FIGURE 2.20 Change of line voltages in three-phase alternating current (AC) circuits.

Motors of greater power capacity are usually driven by **three-phase** electric supplies, each phase having its alternating voltage and current pattern. Voltage and current in each phase change according to **sine curves**, as shown in Fig. 2.20, and are shifted from each other by one-third of the cycle time. In one full cycle (360 degrees in the figure) the direction of the current changes two times; the number of cycles per one second is called the **frequency** of the electricity, with usual values of 60 or 50 Hz. The change in voltage, for example, with time can be expressed with the following formula:

$$V = V_{\max} \sin(2\pi ft) \quad (2.25)$$

where:

V = voltage, Volts

$V_{\max}$ = maximum value or amplitude, Volts

f = frequency, Hz

t = time, s

The flow of an AC in a wire or any other equipment involves **energy losses** dissipated as **heat**, independent of the direction of the current. To calculate these losses, the **effective current** is introduced, which represents the heating effect of the current and is calculated as the **root mean square** (rms) value defined as follows:

$$I_{\text{eff}} = \sqrt{\frac{1}{T} \int_0^T I^2 dt} = \frac{I_{\max}}{\sqrt{2}} = 0.707 I_{\max} \quad (2.26)$$

where:

I_{eff} = effective current, Amps

I_{max} = maximum current, Amps

I = current, Amps

T = cycle time, s

The effective current and voltage are indicated by the usual measuring devices. Because voltage and current change rapidly with time, instead of their actual values, the rms values are usually given to describe their magnitude. For example, 50 A of AC means that the rms value of the actual AC is $I_{\text{eff}} = 50$ A.

2.4.2 Alternating Current Circuits, Alternating Current Power

Electric circuits connected to AC power supplies typically contain different kinds of components, which, according to their electric behavior, can be classified as **resistors**, **inductors**, and **capacitors**.

If an AC flows through a pure **resistor**, the voltage drop across it is in the direction of the current; thus voltage and current are in phase. The magnitude of the voltage drop is determined from the well-known **Ohm's law**:

$$U = R I \quad (2.27)$$

where:

U = voltage, V

R = resistance, Ohms

I = current, A

In the case of transformers and motors, **coils** of wire are placed around iron cores to generate magnetic fields. These coils do not behave as pure resistances because they resist the flow of current and are, therefore, called **inductors**. If the current increases, the system stores energy in the magnetic field, which is released when the current starts to decrease. Thus, magnetic effects react on the original current with the end result that current and voltage are not in **phase**. In an electric circuit containing only inductors, the current is **lagging** the voltage by a phase angle of exactly $\pi/2$ radians (90 degrees). To describe the opposing of the flow of AC by an inductor, the **inductive impedance**, X_L , is used with the units of Ohm.

Capacitors, i.e., conductive materials (plates) separated by insulating materials, behave differently. They take energy from the system to charge their plates and then return the stored energy when the electric charge is removed. Capacitors oppose the change in voltage and cause the current to **lead** the voltage by $\pi/2$ radians (90 degrees). They oppose the AC because of their **capacitive impedance** denoted by X_C .

Because most electric circuits contain all three kinds of components (resistors, inductors, and capacitors), it is easy to see that voltage and current are **never** in phase in an AC circuit. Theoretically, the **phase angle** between them may vary in the range from $-\pi/2$ to $\pi/2$, but in practice the current usually **lags** the voltage and the phase angle is in the range of 0 to $\pi/2$.

In AC circuits containing resistive, inductive, and capacitive components the opposition to a sinusoidal AC is characterized by the total electrical impedance of the circuit. The concept of electrical **impedance** generalizes **Ohm's law** to AC circuits. It is calculated from the following formula and has a unit of Ohms:

$$Z = \sqrt{R^2 + (X_L - X_C)^2} \quad (2.28)$$

where:

Z = total impedance, Ohms

R = resistance, Ohms

X_L = inductive impedance, Ohms

X_C = capacitive impedance, Ohms

The phase angle between the voltage and the current in an AC circuit has a great impact on the calculation of **electric power**. Consider the case shown in Fig. 2.21 where the current lags the voltage by an angle of φ . The **line current** (as measured by an ammeter) can be broken into two components: one in the direction of the voltage (real current) and one at right angles to it (magnetizing current). Because electric power is defined as voltage multiplied by current, one can calculate two kinds of powers depending on the current component used: (1) **apparent power** found from the line current and (2) **real power** from the real current.

For three-phase systems with identical currents in each phase and equal voltages between any pairs of wires, **apparent power** (often designated as **KVA** and measured in thousands of volt amperes) is calculated from the formula below:

$$KVA = \sqrt{3} \frac{U_{\text{line}} I_{\text{line}}}{1,000} = 1.732 \cdot 10^{-3} U_{\text{line}} I_{\text{line}} \quad (2.29)$$

where:

KVA = apparent electric power, 1,000 V A

U_{line} = line voltage, V

I_{line} = line current, A

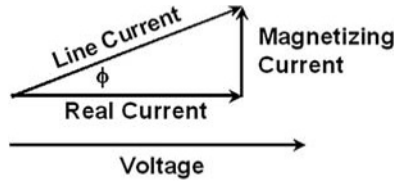


FIGURE 2.21 Illustration of a lagging alternating current (AC).

The **real electric power** (often designated as **KW** and measured in kW units) is found by considering that the real current equals $I_{\text{real}} = I_{\text{line}} \cos \varphi$ and:

$$KW = \sqrt{3} \frac{U_{\text{line}} I_{\text{line}}}{1,000} \cos \varphi = 1.732 \cdot 10^{-3} U_{\text{line}} I_{\text{line}} \cos \varphi \quad (2.30)$$

where:

KW = real electric power, kW

U_{line} = line voltage, V

I_{line} = line current, A

The term $\cos \varphi$ is called the **power factor** (often designated as **PF**) and is, by definition, the **ratio** of the real power to the apparent power, real power being equal to the power measured by a wattmeter and apparent power being calculated from ammeter and voltmeter readings. The actual value of power factor is a measure of the **power efficiency** of the AC circuit: the greater it is, the more efficient is the power consumption in the system. Its maximum value is at a phase angle of zero and equals $\cos(0) = 1$, when voltage and current are in phase.

2.4.3 Transformers

A transformer is a device that changes the voltage in an AC circuit and works on the principle of the electromagnetic **induction**. It consists of an iron **core** of different possible shapes and two **coils** of insulated metal wires. One of the coils (called **primary**) is connected to an AC power source and the other one (called **secondary**) is connected to the circuit to be powered. The voltage generated by electromagnetic induction in the secondary coil relates to the primary voltage, as the number of turns in the secondary coil relates to those in the primary coil. This ratio is called the **turn ratio** of the transformer and the secondary voltage is calculated according to the formula given below:

$$U_s = U_p \frac{N_s}{N_p} \quad (2.31)$$

where:

U_s, U_p = secondary and primary voltage, V

N_s, N_p = number of turns in the secondary and primary coils, —

The **frequencies** of the electric currents in both coils are identical, but the currents will be different. Assuming no energy losses occur during the voltage transformation (practically true in real transformers), the input and output electric powers should be equated as $U_p I_p = U_s I_s$ and the following formula is found for the relationship of currents:

$$I_s = I_p \frac{N_p}{N_s} \quad (2.32)$$

where:

I_s, I_p = secondary and primary currents, Amperes

N_s, N_p = number of turns in the secondary and primary coils, —

Transformers can be used to increase (**step up**) or to decrease (**step down**) the primary voltage and are grouped accordingly (see Fig. 2.22). Transformers used in ESP installations can be three **single-phase** transformers or one **three-phase** transformer, both filled up with dielectric oil to facilitate cooling and insulation. In **autotransformers** (Fig. 2.23), in contrast to regular ones, the

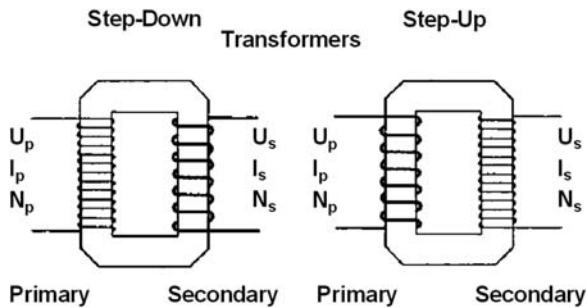


FIGURE 2.22 Schematics of step-up and step-down transformers.

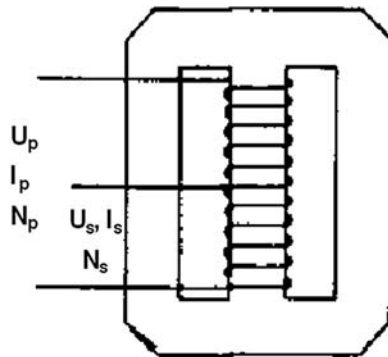


FIGURE 2.23 Schematic diagram of an autotransformer.

primary and secondary circuits are not insulated from each other and such transformers contain one coil only. The coil is connected to the power source and part of it is connected to the driven (secondary) circuit.

Example 2.6

An ESP installation requires a voltage of 1,500 V, and a current of 35 A on the surface, where the available voltage is 380 V. What should be the necessary transformer's turn ratio and what is the current load on the electric network?

Solution

The turn ratio from Eq. (2.31):

$$N_s/N_p = 1,500/380 = 3.95.$$

The current in the primary coil is found from Eq. (2.32) as:

$$I_p = 35 \cdot 3.95 = 138 \text{ A}.$$

2.4.4 Electric Motors

The originally and still universally used electric motors in ESP service are three-phase, two-pole, asynchronous AC **induction** motors; the great majority of installations today are powered by such motors. An emerging motor type that was introduced to the artificial lift field in the early 2000s utilizes permanent magnets in the rotor and is called **permanent magnet motor** (PMM). The following sections give a basic overview of these two motor types.

2.4.4.1 Induction Motors

Three-phase, two-pole, squirrel-cage AC **induction** motors used in ESP service belong to the most common type of electric motors. These motors work on the principle of the electromagnetic **induction**, which states that an electric current is induced in any conductor moving in a magnetic field. The magnetic field is generated in the **stator**, the standing part of the motor containing one coil for each phase. This field rotates with the changes in direction of the AC because the electromagnets change their magnetic poles twice for every cycle of the AC.

The rotational speed of the magnetic field is the motor's **synchronous speed**, which depends on the frequency of the AC and the number of poles (usually two) the stator has:

$$N_{\text{synch}} = \frac{120f}{p} \quad (2.33)$$

where:

N_{synch} = synchronous speed, RPM

f = frequency of the AC power, Hz

p = number of poles in the stator, —

Inside the stator, attached to the shaft of the motor, is the “squirrel-cage” type **rotor** isolated from the stator by a narrow **air gap**. The rotor consists of copper bars short-circuited by two copper “**end rings**”; it is not connected to the power source. The rotating magnetic field maintained by the stator windings induces the flow of an AC in the rotor; thus the rotor becomes a set of **electromagnets**. The poles of the rotor’s magnetic field are attracted and repelled by the unlike and like poles, respectively, of the stator, and the rotor maintains the continuous rotation of the motor shaft.

Because the operation of the motor depends on electromagnetic induction requiring a relative movement between the stator’s (primary) magnetic field and the rotor’s wires, it must be clear that the rotor must always revolve at a **slower** speed than the motor’s synchronous speed. That is why these motors are also called **asynchronous motors**. The speed difference is called the “**slip**” of the motor. It is sometimes defined as a fraction of the synchronous speed:

$$slip = \frac{N_{synch} - N}{N_{synch}} \quad (2.34)$$

where:

$slip$ = motor slip, —

N_{synch} = synchronous speed, RPM

N = motor speed, RPM

Typical ESP motors behave very similarly to **NEMA B** motors and have a slip of 100–150 RPM at fully loaded conditions. Their synchronous speed at 60 Hz operation is, substituting $p = 2$ in Eq. (2.33) for the usual motors with two poles, 3,600 RPM. Therefore, actual speeds of 3,500 RPM or less can be expected at fully loaded conditions [11].

As seen from Eq. (2.33), the speed of the ESP motor depends mainly on the **frequency** of the AC power. Therefore, if driving of the submersible pump at different speeds is required, the change in the driving power’s frequency is the most feasible solution, often used in ESP installations with VSDs.

2.4.4.2 Permanent Magnet Motors

The PMMs used in the ESP industry are **brushless AC synchronous** motors. Their main difference from induction motors is the **rotor** that, instead of the squirrel cage construction, contains permanent **magnets**. The magnets manufactured from **rare-earth** metals (the strongest made from neodymium iron boron (NdFeB)) using powder sintering technology generate an extremely strong, static **magnetic flux** required for motor operation. They provide a **constant** magnetic field in the motor’s air gap that is not subject to failure except for demagnetization by overheating. Because sintered NdFeB magnets

have a Curie temperature of about 500°F (above which magnetic properties of the metal are lost), they can safely be used in ESP motors.

The **stator** of a PMM, on the other hand, is similar to that of an induction motor and contains the **coils** connected to the three-phase AC power source. Just like in an induction motor, a rotating magnetic field is created in the air gap by the three-phase current. This magnetic field interacts with the permanent magnets of the rotor and creates the rotation of the motor shaft. Due to the high magnetic flux created by the permanent magnets, PMMs are capable of generating two to four times the power of an equivalently sized induction motor.

The working principle of PMMs is illustrated for the case of a **two-pole** motor operated under scalar control in Fig. 2.24.

The stator windings (magnet wire) for the three phases U, V, and W are represented by small circles in the stator core, the direction of current flow is also indicated. Four different positions of the rotor (carrying the permanent magnets) are shown to illustrate the rotation of the motor shaft. In the first diagram, only two phases (V and W) are energized from the power source, phase U is not. The strong magnetic flux of the North pole of the permanent magnet and the magnetic field created by the inflowing currents in phases V and W interact and the resultant force is indicated by a small arrow. At the same time, the South pole's magnetic field and the fields created by the outflowing phase V and W currents create a force of opposite direction. These forces result in a torque acting on the motor shaft, which turns in the counterclockwise direction.

As the rotor turns, the next phase (phase V) is switched off and the events just described repeat and similar forces act on the rotor, keeping it in rotation. Switching on and off of the phases continues successively (as seen in the figure) and the rotor is turned by the permanent magnets attached to it at a speed corresponding to the frequency of the AC power. It is clearly seen that PMMs are **synchronous** motors. It is obvious from the basic description of their operating principle that PMMs necessitate the use of special motor **controllers**. These provide switching on and off of the three phases based on the actual position of the rotor and utilize sophisticated power technology solutions in the VSDs that drive PMMs. The solution described in Fig. 2.24 is a **sensorless** control and utilizes the voltage induced by the rotor in the coil of

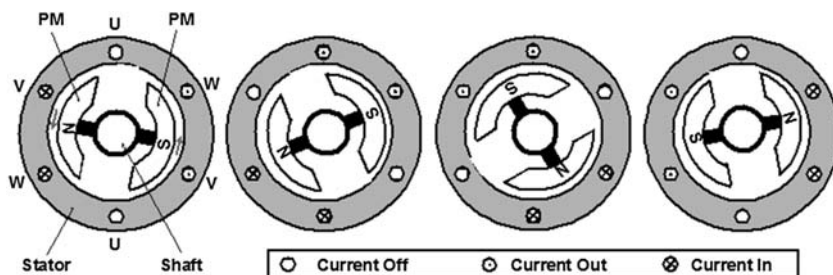


FIGURE 2.24 Operating principle of a permanent magnet brushless motor.

the phase not connected to the power supply. The rotor and the idle coil create a generator, whose output voltage (back electromotive force) precisely indicates the position of the rotor. The VSD unit controls the switching on and off of the phases to the motor's stator according to changes in rotor position. Sophisticated control circuits are used to ensure that the magnetic poles of the permanent magnet and the stator's rotating flux are always at optimum positions to each other so that the torque on the motor shaft is maximized.

PMMs in the ESP industry normally have two or four **poles**; their synchronous speed is found from Eq. (2.33) as a function of the electrical frequency. For 60 Hz operation, their speed is 3,600 RPM for a two-pole and 1,800 RPM for a four-pole motor. Motor speed increases with increased frequencies, for example, at 200 Hz the motors turn at a speed of 12,000 RPM for a two-pole and 6,000 RPM for a four-pole motor. Their relative advantages over induction motors will be detailed in Chapter 3.

2.4.5 Electric Cable

The three-phase electric power is led from the surface to the electric motor through the three wires of the ESP cable. The individual wires of the cable can be considered as long, thin **conductors**, the electric resistance of which is directly proportional to their length and inversely proportional to their cross section, as described by the equation:

$$R = \rho \frac{L}{A} \quad (2.35)$$

where:

- R = electrical resistance, Ohms
- ρ = resistivity of the material, Ohm ft
- L = length of the conductor, ft
- A = cross-sectional area, ft²

The voltage drop along a cable is found from **Ohm's law** (see Eq. 2.27), based on the current flowing through it. When calculating voltage drops, one should consider the temperature of the cable because the resistance of most metals increases with an increase in temperature. The following formula is generally used to find the cable's resistance at any temperature:

$$R_T = \frac{L_c r}{1,000} [1 + 0.00214(T - 77)] \quad (2.36)$$

where:

- R_T = cable resistance at temperature T , Ohms
- L_c = cable length, ft
- r = specific resistance of conductor at 77°F, Ohm/1,000 ft
- T = cable temperature, °F

Example 2.7

Find the total resistance of a 6,000-ft long ESP cable at an average cable temperature of 300°F, if its specific resistance at 77°F is 0.16 Ohms/1,000 ft.

Solution

Using Eq. (2.36) we get:

$$R_T = 6,000 \, 0.16 / 1,000 [1 + 0.00214(300 - 77)] = 0.96 \, 1.477 = 1.42 \, \text{Ohm}.$$

2.5 BASICS OF SYSTEMS ANALYSIS

Systems analysis of producing oil and gas wells (often called *Nodal Analysis*) is one of the latest additions to the petroleum engineer's arsenal of design and analysis tools. The methodology and calculation procedures developed in the last decades are based on the recognition that the underground reservoir, the producing well, and the surface liquid and gas handling equipment constitute a complex, **interrelated** system. Accordingly, any process in any element of the system entails changes that occur not only in the given part but also in the system as a whole. This section introduces the basic principles of systems analysis as adapted to the description of producing well behavior.

2.5.1 Introduction

Petroleum fluids found in an underground reservoir move through a complex system to reach their destinations on the surface. This system is called the **production system** and comprises the following main components: the **reservoir**, the producing **well**, the surface **flow line**, and the **separator**. Some of these can further be divided into smaller elements, for example, the well, besides the tubing string, may contain safety and/or gas lift valves, as well as other components. The production system is thus a system of **interconnected** and **interacting** elements that all have their own specific performance relationships, but each, in turn, also depends on and influences the other elements. To produce fluids from the well, all components of the system must work together. Thus, the solution of any fluid production problem requires that the production system be treated as a complete entity.

The outlines of this principle were first given by **Gilbert** [12], the father of production engineering, in the 1950s. He described the interaction of the reservoir, the well, and the wellhead choke and proposed a system-oriented solution for determining the production rate of a flowing well. The practical use of Gilbert's ideas was limited, mainly due to the limitations of the methods available in his time for modeling the performance of the system's elements. During the last decades, however, research into the behavior of the

individual hydraulic elements of oil and gas wells has been very intensive. Because of this progress, there exist today several different theories, calculation, and design procedures that reliably model the performance of each element of a production system. Good examples for this are the numerous correlations available for calculating pressure traverses in vertical and horizontal pipes.

The wide selection of available calculation models and the advent of computers, which eased the burden of the necessary calculations, led to the reappearance of Gilbert's ideas in the early 1980s [13,14]. The new contributions not only aim at the **numerical simulation** of the production system's hydraulic behavior but also enable the **optimization** of the system to produce the desired flow rate most economically.

The systems analysis methods and procedures mentioned above were named “**Nodal Analysis**” by **K.E. Brown**, and the term has generally been accepted. A full treatment of Nodal Analysis principles has been given by **Beggs** [15]. The application of this theory to flowing and artificially lifted oil wells can have immediate practical and economical advantages.

2.5.2 The Production System

The **production system** of any oil or gas well comprises part of the reservoir, the system transporting well fluids to the surface, and the surface separation equipment. These components have their own performance relationships describing their **behavior** under different flow conditions. The formation is characterized by the laws of flow in porous media, whereas in most of the other components single-phase or multiphase flow in pipes takes place. Accordingly, a proper description of the total system's behavior cannot be achieved without a thorough investigation of each component's performance. The different calculation models developed for the characterization of the system's components, therefore, provide a firm foundation for systems analysis.

Consider a simple flowing oil well with only some of the components described above. The schematic drawing of this simplified case is shown in Fig. 2.25. Fluids in the formation flow to the well from as far as the boundary of the drainage area, 1. After entering the well through the sandface, vertical flow in the tubing starts at the well bottom, 2. In the tubing string vertical or inclined tubing flow takes place up to the wellhead, 3. A surface choke bean is installed at 4, which is also the intake point to the flow line. Horizontal or inclined flow in the flow line leads into the separator, 5.

The points in the flowing well's production system designated by the numbers in Fig. 2.25 constitute the “**nodes**” of the systems analysis theory. They separate the different components of the system: the formation, the tubing string, the flow line, etc. Between any two node points, flowing pressure

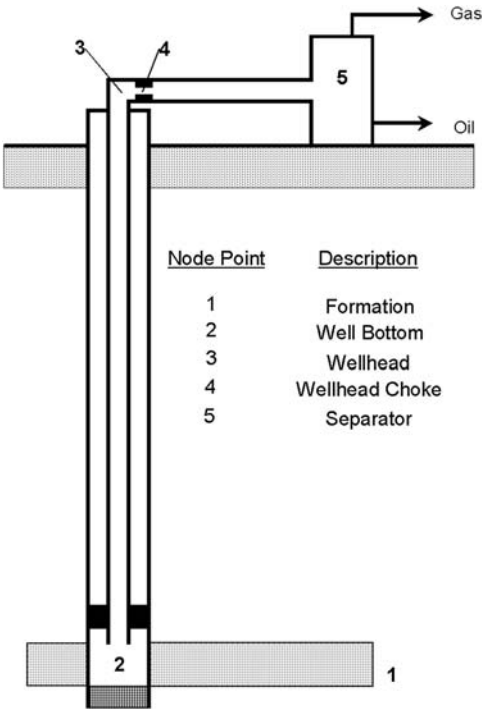


FIGURE 2.25 The production system of a flowing well with the node points.

decreases in the direction of the flow and the pressure drop can be calculated based on the characteristics of the given component. By this way, the performance of any component is established as the variation of the flow rate with the pressure drop across the given component. This task is accomplished through the application of the different pressure drop calculation models valid for the different kinds of hydraulic components.

Two node points deserve special considerations: the boundary of the drainage area (point 1) and the separator (point 5). These points constitute the two **endpoints** of the production system and their pressures are considered constant for longer periods of time. Formation pressure at the outer boundary of the drainage area of the well, of course, changes with the depletion of the reservoir but for production engineering purposes involving short and medium periods, this change can be **disregarded**. Separator pressure, at the same time, is usually set for the **whole life** of the field and is held constant. Thus, the two endpoint pressures the average reservoir pressure and the separator pressure can duly be considered **constant** values for the purposes of systems analysis.

2.5.3 Basic Principles

One of the many objectives of systems analysis is the determination of the **flow rate** of a given production system. The solution of this problem is illustrated here through the example of a flowing well.

As discussed before, in connection with Fig. 2.25, an oil well can be considered a **series-connected** hydraulic system made up of its components, which are bracketed by the appropriately placed **nodes**. Evaluation of the total system's performance permits the following conclusions to be made:

- Mass flow rate throughout the system is **constant**, although phase conditions change with changes in pressure and temperature.
- Pressure **decreases** in the direction of flow because of the energy losses occurring in the various system components.
- At node points, input pressure to the next component **must** equal the output pressure of the previous component.
- System parameters being **constant** for considerable periods are:
 - the **endpoint** pressures at the separator and in the reservoir,
 - the wellbore and surface **geometry** data (pipe diameters, lengths, etc.), and
 - the **composition** of the fluid entering the well bottom.

Taking into account these specific features of the production system, a procedure can be devised to find the flow rate at which the system will produce. This starts with dividing the system into two **subsystems** at an appropriately selected node called the “**solution node**.” The next step is to find pressure versus rate curves for each subsystem. These functions are constructed starting from the node points with known pressures at the separator and at the well bottom. The intersection of the two curves gives the **cooperation** of the subsystems and thus the desired rate.

A simple example is shown in Fig. 2.26 [16]. The well is a low producer and has no surface or downhole chokes installed. The well's production system is divided at Node 2 with one subsystem consisting of the flow line and the tubing string, and the other being the formation. The pressure versus rate diagram of the formation is the familiar **IPR curve**. The other curve is constructed by summing the separator pressure and the pressure drops in the flow line (wellhead pressure curve) and by further adding to these values the pressure drops occurring in the tubing string. The resulting curve is the **tubing intake pressure** versus production rate. The system's final rate is found at the intersection of this curve with the IPR curve and is 320 bpd in the present example.

The same procedure can be followed starting from a different node position but will give the same result. This shows that systems analysis allows a flexibility to study different situations.

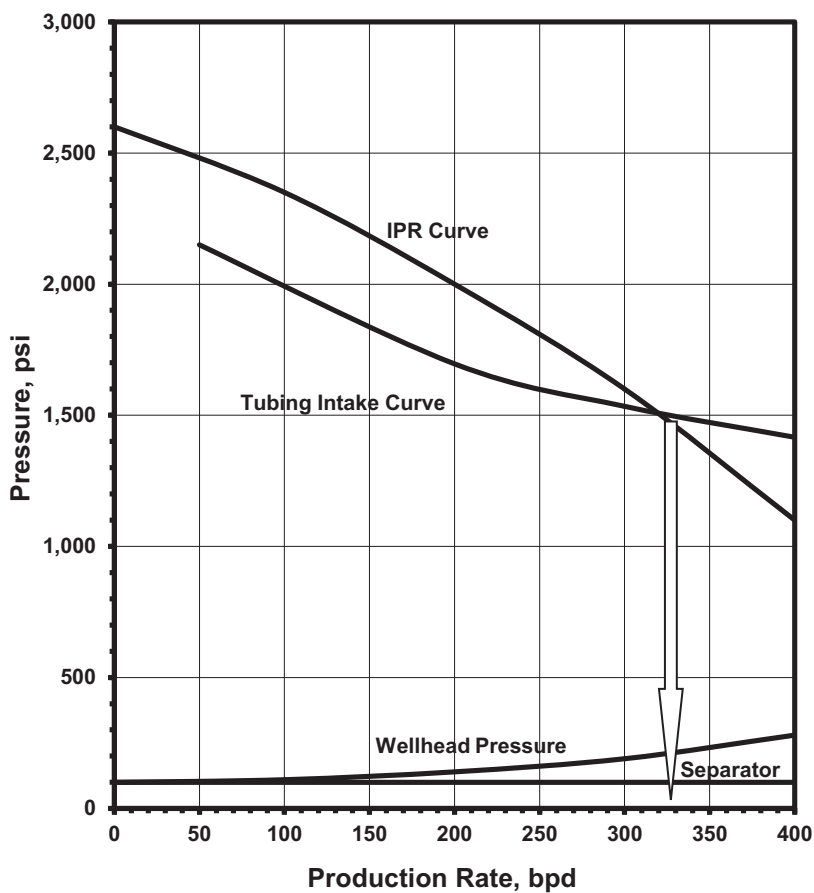


FIGURE 2.26 Example systems analysis for a flowing oil well after Ref. [16].

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Chapter 3

Electrical Submersible Pump Components and Their Operational Features

3.1 INTRODUCTION

During its long history, the electrical submersible pump (ESP) system proved to be an efficient means of producing liquid from oil and water wells. The classical or “**conventional**” installation is illustrated in Fig. 3.1, in which the ESP unit is run on the tubing string and is submerged in well fluids. The electric submersible **motor** is at the bottom of the unit and is cooled by the wellstream passing by its perimeter. It is connected to the **protector** (a.k.a. seal) section that provides many crucial functions for the safe operation of the unit. On top of the protector a pump intake or gas separator is situated, which allows well fluids to enter the centrifugal pump and, at the same time, can remove low quantities of free gas from the wellstream. The separated gas bubbles through the liquid column in the casing annulus and reaches the casinghead where it is directed into the flowline. Connection of the casinghead to the flowline ensures that the fluid level is kept constant above the submersible pump. Liquid is lifted to the surface by the **multistage centrifugal pump**, the heart of the ESP system.

The submersible motor is supplied with three-phase alternative current (AC) via an **electric cable** running from the surface all along the tubing string. Produced fluids flow through the tubing string to the surface, where a special **wellhead** ensures feeding of the electric cable into the well. The surface equipment includes a **junction box** where downhole and surface electric cables are joined and a control unit (called **switchboard**) that provides measurement and control functions. The ESP unit receives AC from a set of **transformers** (not shown) that supply the required voltage by stepping up or down the voltage available from the surface electric network.

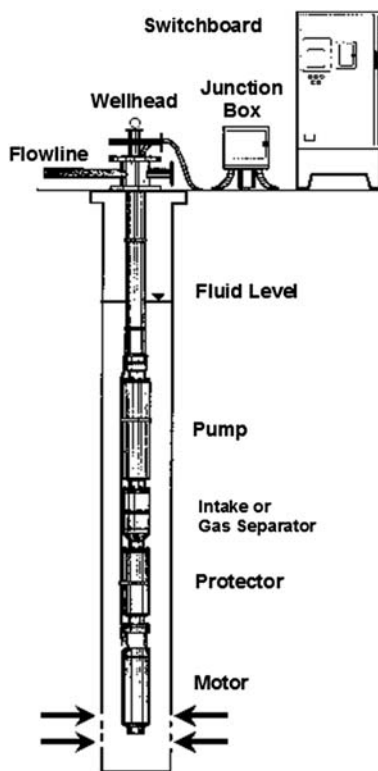


FIGURE 3.1 Conventional ESP installation.

The above type of installation has stood the test of time and is frequently used even today. The most important features making it a **conventional ESP installation** are as follows:

- Only **liquid** enters the centrifugal pump providing ideal conditions for the pump. The well must produce only a low amount of free gas at the pump suction that can easily be removed by the gas separator.
- The viscosity of the produced liquid is low, approaching the viscosity of water.
- The ESP motor is supplied with an AC of a **constant frequency**, thus its speed and consequently that of the centrifugal pump are constant.

Although the abovementioned conditions are not always met, the conventional ESP installation can be applied in a great variety of field conditions. This chapter will discuss the components of such conventional installations and also details their main operational features. Unusual conditions (greater gas production, viscous crude, etc.) require the use of special equipment; installations different from the conventional are the subject of Chapter 4.

3.2 THE SUBMERSIBLE PUMP

3.2.1 Basic Features

The **heart** of the ESP unit is the submersible pump and the design and analysis of the whole ESP system cannot be understood without a basic comprehension of the operation of the pump. Hence the description of the system's components must be started with a thorough analysis of the construction and operation of the centrifugal pumps.

The submersible pumps used in ESP installations are **multistage centrifugal pumps** operating in a vertical position. Although their constructional and operational features underwent a continuous evolution over the years, their basic operational principle remained the same. Produced liquids, after being subjected to great centrifugal forces caused by the high rotational speed of the **impeller**, lose their kinetic energy in the **diffuser** where a conversion of kinetic to pressure energy takes place. This is the main operational mechanism of radial and mixed flow pumps, as detailed in Chapter 2.

Fig. 3.2 illustrates the main parts of an ESP containing mixed flow stages. The pump shaft is connected to the gas separator or the protector by a

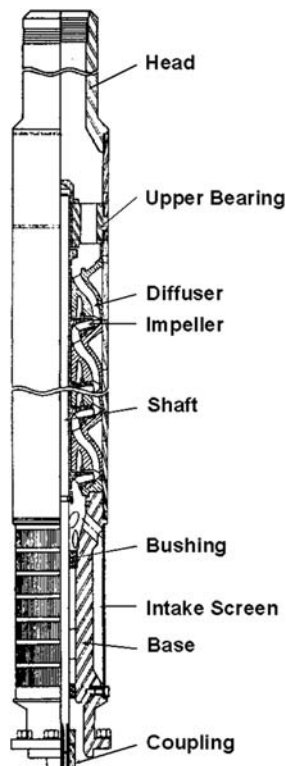


FIGURE 3.2 Main parts of an ESP.

mechanical spline coupling at the bottom of the pump. Well fluids enter the pump through an **intake screen** and are lifted by the pump stages. The figure illustrates a **standard** intake with several large (minimum 1") screened ports and bolted to the bottom of the pump housing with its shaft connected to the protector shaft by a spline coupling. **Integral** intakes are also available. Other parts include the radial **bearings** (bushings) distributed along the length of the shaft providing radial support to the pump shaft turning at high rotational speeds. An optional **thrust bearing** takes up part of the axial forces arising in the pump but most of those forces are absorbed by the protector's thrust bearing.

The liquid-producing capacity of an ESP depends on the following factors:

- the rotational **speed** provided by the electric motor,
- the **diameter** of the impeller,
- the **design** of the impeller (characterized by its specific speed),
- the actual **head** against which the pump is operating,
- the thermodynamic properties (density, viscosity, etc.) of the produced fluid.

The ESPs are designed to operate in the **proper** direction of rotation; as seen from the top, they can be driven in **clockwise** (CW) or **counterclockwise** (CCW) directions. The actual direction depends solely on the electric motor's rotation, which is easily set during installation in accordance with the pump's requirement. CW rotation, used by many manufacturers, involves the risk of **unscrewing** of threaded equipment parts (pumps, protectors, motors, etc.) during startup and running because of reactive torque. To prevent such failures "**lock plates**," i.e., steel pieces are welded across the equipment heads/bases and housings against possible separation of threaded connections. In contrary, if the pump is rotated CCW, unscrewing does not occur.

The pump will operate with the **wrong** direction of rotation as well but its output will be much reduced. This fact may be used to **detect** the proper rotation of the equipment by observing the developed wellhead pressure and/or the produced liquid rate immediately after starting the ESP unit. If measured performance data do not match design values, the direction of motor rotation must be changed.

Conventional ESP installations run on AC power with a constant frequency of 60 or 50 Hz. ESP motors in 60 Hz electric systems rotate at a speed of about 3,500 RPM, whereas in case of a 50 Hz power supply the motor speed is about 2,900 RPM. For constant speed applications, the most important factor is **impeller size**, which, of course, is limited by the inside diameter of the well casing. Pumps of bigger sizes can produce greater rates, although impeller design also has a great impact on pump capacity.

Present-day ESPs come in different capacities from a few hundred to around 80,000 bpd of liquid production rate and in outside diameters (ODs) from around 3" to 11". Smaller units contain "**pancake**" type impellers with **radial discharge** and are used up to the rates of 1,500–3,500 bpd, above which **mixed flow** impellers are used.

The **length** of individual ESPs is limited to about 20–25 ft, for ensuring proper assembly and ease of handling. **Tandem pumps** are made up of several pump sections (up to three) and are used to achieve higher operational heads usually required in deeper wells. Thus several hundreds of stages can be run, the maximum number of stages being limited by one or more of the following factors:

- the mechanical strength of the **pump shaft**, usually represented by the shaft's horsepower rating,
- the maximum **burst-pressure rating** of the pump housing,
- the maximum **allowed axial load** on the unit's main thrust bearing (usually situated in the protector section).

Individual stages in ESPs, provided they are of the same impeller design, handle the same liquid volume and develop the same amount of head. The heads in subsequent stages are **additive** so a pump with a given number of stages develops a total head calculated as the product of the total number of stages and the head per stage. This rule allows one to find the number of stages required to develop the total head to overcome the total hydraulic losses, valid at the desired liquid production rate in a well.

As the size of well casing limits the OD of the ESP equipment that can be run, pump selection is heavily restricted by the actual casing size. **Appendix B** lists the main dimensional data of common American Petroleum Institute (API) tubulars. For comparison, **Appendix C** contains the most important parameters (diameters, recommended liquid rate ranges) of submersible pumps available from a leading manufacturer.

Example 3.1

Using the total head calculated in Example 2.4 find the pump's required number of stages using the pump represented in Fig. 3.3.

Solution

The head developed by one stage at a flow rate of 2,000 bpd is read off Fig. 3.3 as 26 ft.

The required head is taken from Example 2.4 as 7,303 ft, and the number of stages is found easily:

$$\text{Stages} = \text{Head required} / \text{Head per stage} = 7,303 / 26 = 281 \text{ stages.}$$

3.2.1.1 Pump Performance Curves

The performance of ESPs is characterized by the **pump performance curves** introduced in Section 2.2.2.3. These are plotted in the function of the pumping rate and represent

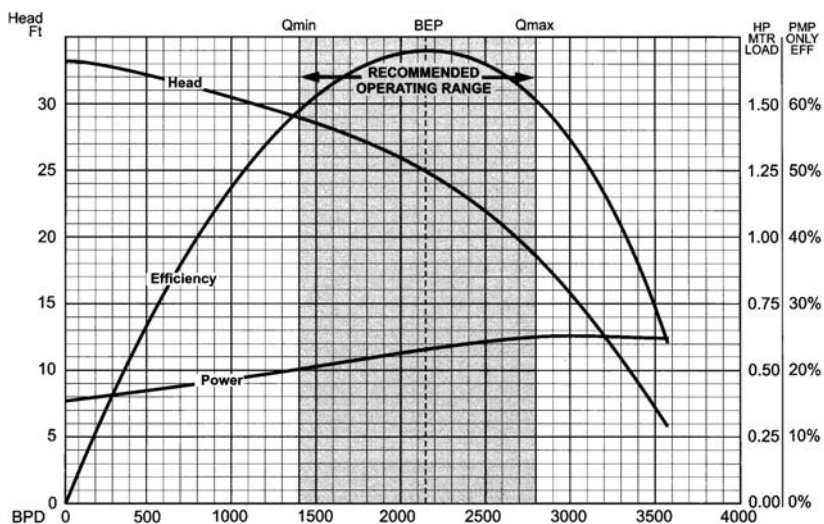


FIGURE 3.3 Example performance curves of an ESP. BEP, best efficiency point.

- the **head** developed by the pump
- the **efficiency** of the pump
- the mechanical **power** [brake horsepower (BHP)] required to drive the pump when pumping water.

These curves are **experimentally** obtained with freshwater under controlled conditions described in **API RP 11S2 [1]** at an operating temperature of 60°F. Tests on submersible pumps are made by driving the pump at a **constant** rotational speed, usually 3,500 RPM for 60 Hz service. This is the speed generally assumed as the actual speed of a fully loaded ESP motor having a synchronous speed of 3,600 RPM. Experiments with other fluids or speeds must be corrected to these standard conditions using the **affinity laws**. All performance parameters must represent the operation of one or a specified number of pump stages clearly indicated on the chart.

Performance curves of pumps in 50 Hz service are typically published for a fixed rotational speed of 2,917 RPM. As described by **Butlin [2]**, this number was chosen based on a misunderstanding of **slip** in electric motors, using the concept of the “percentage slip.” It was erroneously assumed that motor slip is found as a percentage of the synchronous speed. Slip (defined as the difference between the motor’s synchronous and actual speeds), however, depends on motor voltage and torque loading only. This means that the same motor at the same pump load will have the same amount of slip, independent of the electric frequency. The synchronous speed of ESP motors at 50 Hz service being

3,000 RPM, the actual full-load motor speeds must be around 2,900 RPM or even less for small-diameter motors.

Because of manufacturing tolerances and other factors, a variance in performance as compared to published performance curves can exist from one pump to another of the same design and the same manufacturer. The accepted **tolerances** between published curves and actual pump performance are specified as follows:

- The pump's best efficiency point (BEP) at the nominal flow rate should be at least 90% of the published value.
- The head—flow rate performance curve in the recommended operating range should have a tolerance of $\pm 5\%$.
- The tolerance of pump power—flow rate curve in the recommended operating range should be $\pm 8\%$.

The sample curves shown in Fig. 3.3 represent the performance of one stage of a given pump design. Pump performance curves include the **head** developed by the pump, the **power** required to drive the pump, and the **efficiency** of the pump, all in the function of the pumping rate. The performance parameters belonging to the **BEP** represent the criteria for an optimum utilization of the pump, around which the recommended range of operation is indicated.

The recommended range of pumping rates for any ESP is strictly related to the variation of **axial forces** occurring in the pump. As discussed in Section 2.3.2.5, static and dynamic axial forces occurring in pump stages are the results of different phenomena and may take the form of downthrust or upthrust. Fig. 3.4 presents the relationship between the axial forces and the recommended operating range of the pump; their interaction is detailed in the following.

The lower part of Fig. 3.4 schematically shows the change of **axial forces** in an ESP stage in the function of the liquid rate produced by the pump. **Downthrust** is basically determined by the **head** developed because its main component comes from the pump's discharge pressure acting on the top and bottom shrouds of the impeller. Its variation with pumping rate, therefore, follows the shape of the pump's head—rate performance curve, as shown in the figure. It is at a maximum at shut-in conditions (at a pumping rate of zero) and diminishes to zero where the pump's head decreases to zero. **Upthrust** forces arising in the pump stage, on the other hand, are the result of the change in **inertial** forces and are proportional to the kinetic energy of the liquid pumped. Thus their variation with pumping rate follows a second-order curve.

The sum of the up- and downthrust forces is shown in a bold line representing the **net thrust** arising in the pump stage. As seen, the operation of ESPs is **dominated** by downthrust forces because the net axial force points downward in the largest part of the operating pumping rates.

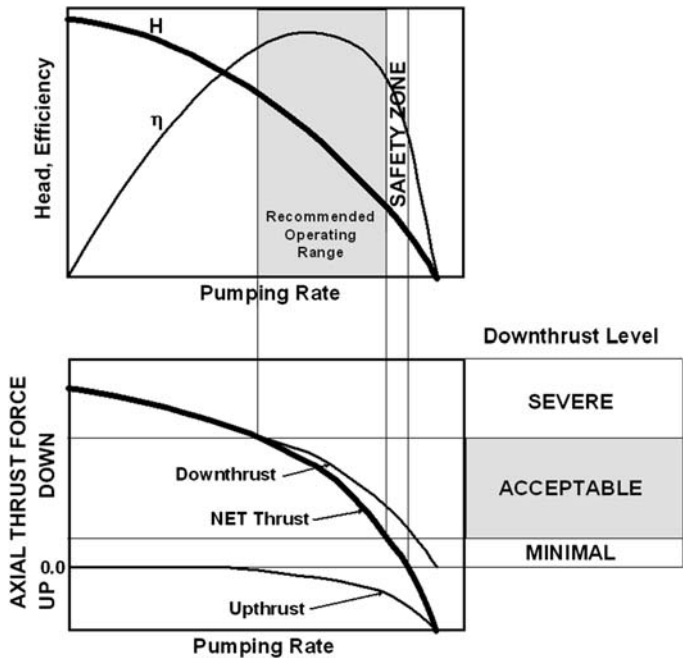


FIGURE 3.4 Axial thrust forces and the operating range of ESPs.

As discussed in Section 2.3.2.5, axial forces developed in ESPs must be compensated, otherwise the axial movement of the impellers and the pump shaft leads to mechanical **damage** of the stages. Elimination of such forces is accomplished differently in stages with **fixed** impellers from stages with **floating** impellers.

- In **fixed** impeller pumps all axial forces are transmitted to the pump shaft and must be balanced by the main thrust bearing, situated in the protector section of the ESP unit. This solution necessitates the use of thrust bearings of relatively large capacity.
- In **floating** impeller pumps, on the other hand, most of the axial forces are taken up by the forces arising in the up-, and downthrust **washers** installed on the impellers. The benefit of this impeller arrangement is that smaller-capacity thrust bearings are required in the protector.

In the following discussion, let us assume a **floating** impeller pump and determine the safe operating range of such ESPs. In these pumps, most of the net thrust illustrated in Fig. 3.4 must be absorbed by the thrust washers in the pump stage. Typical arrangement of thrust washers is illustrated in Fig. 3.5 from which it can be seen that the load-carrying area of the upthrust washer is

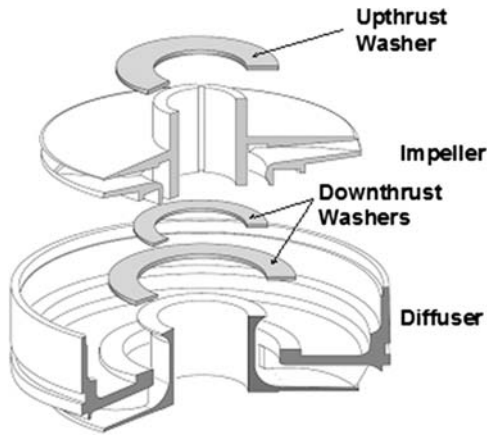


FIGURE 3.5 Comparison of upthrust and downthrust washer areas.

much smaller than that of the downthrust washers; therefore, upthrust is more **dangerous** for the pump's operation than downthrust. To safeguard against the occurrence of upthrust, a minimum of downthrust is assumed and a safety zone in pumping rates is created, as shown in Fig. 3.4. As illustrated in the figure, this defines the **upper** boundary of the pump's applicability in such a way that upthrust conditions are safely avoided.

As **downthrust** forces are absorbed by the combined effort of the washers and the unit's main thrust bearing situated in the protector, their combined capacity defines the maximum acceptable axial load that can be allowed. The pumping rate belonging to this load defines the minimum recommended pumping rate of the ESP.

Any given pump, therefore, must be operated in the **recommended** operating range defined by the two boundary rates described previously. Keeping the operation of an ESP inside the recommended pumping rate range poses the most severe **restriction** in the application of ESP equipment and should never be forgotten by users.

A floating impeller centrifugal pump's operating modes are illustrated in Fig. 3.6, in which the ranges of upthrust and downthrust conditions are defined. In the recommended range the impeller is said to "**float**" between the two diffusers but is forced against the diffusers when pumping liquid rates outside the range. Pump stages are damaged quickly because of the resultant mechanical wear that occurs in upthrust or downthrust conditions.

Although ESP performance curves are normally established by experiments, analytical models are also available [3] to calculate the pump's head performance curve. The use of pump performance curves in computer programs is facilitated by the fact that head—rate and power—rate functions can

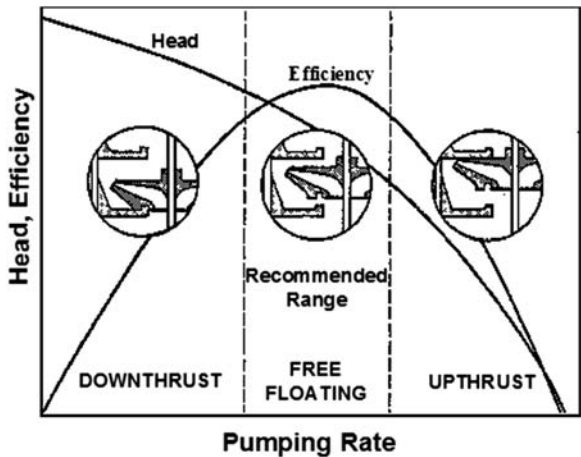


FIGURE 3.6 Illustration of pump operating modes.

be fitted very accurately by **polynomial** functions of the liquid rate. The use of the following mathematical formulas is universally accepted:

$$H = \sum_{i=0}^N a_i q^i \tag{3.1}$$

$$\text{BHP} = \sum_{i=0}^M b_i q^i \tag{3.2}$$

where

- H = head developed by pump, ft
- BHP = required pump power when pumping water, HP
- q = liquid rate, bpd
- a_i, b_i = coefficients
- N, M = maximum number of coefficients, —.

Because of the **regular** shape of the pump performance curves, they can be fitted very precisely by the above formulas, using a maximum of eight to nine coefficients: $N \leq 9$ and $M \leq 9$. Computer programs utilize this feature and can reliably describe the performance of different pumps with the help of a proper data base of pump coefficients.

In ordinary pumping situations the ESP, while pumping a given liquid rate, develops a positive head, i.e., both the liquid rate and the head are positive. Special cases, however, may necessitate the description of the pump’s operation at negative rates and/or heads, e.g., when injecting liquid into the well through an inoperative pump. In such cases a **four-quadrant** representation of the pump’s performance may be useful where the liquid rate (abscissa) and the

head developed (ordinate) axes are extended into negative regions [4]. Out of the four possible combinations of positive and negative values of rates and heads, three versions are possible:

- Normal operations take place in the first quadrant with positive rates and heads.
- In the second quadrant with positive flow rates but with negative heads due to the pressure drop occurring in its stages the ESP acts as a **turbine**. This can happen when the well is produced by another artificial lift method (e.g., gas lifting), with the pump not energized. The pressure drop across the stages creates great upthrust forces and the pump can easily be damaged.
- If fluid is **injected** through a pump not driven by its motor, flow rates are negative but head is still positive because the pressure decreases downward across the pump. The pump operates in the fourth quadrant and all thrust forces are in the downward direction. These unbalanced forces will result in heavy downthrust wear of the pump stages.

3.2.2 Floating Versus Fixed Impeller Pumps

Stages with **floating impellers** are the original and simplest constructions where the impellers are free to “float” axially (move up and down relative to the pump shaft) because they are not fixed to the pump shaft in the axial direction. As shown in Fig. 3.7, impeller hubs are not stacked so there is a vertical **distance** between the successive hubs. Furthermore, each stage contains downthrust washers to absorb the axial forces occurring during operation as well as to seal and minimize recirculation of fluids within the stage. Although called “floater” impellers, its washers are normally in contact with the diffuser pad because of the axial load on them. Because of this

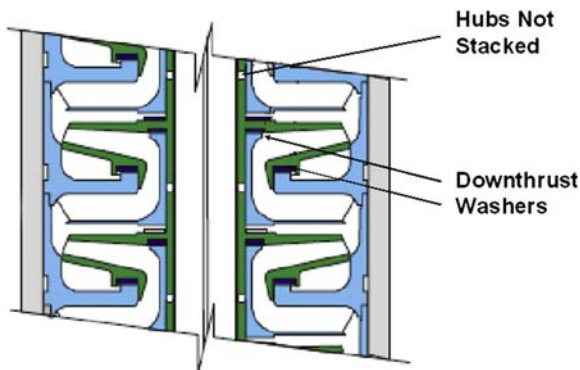


FIGURE 3.7 Construction details of floating impeller stages.

condition, the unit's main thrust bearing, situated in the protector section, has to carry only the downward thrust acting on the pump shaft.

The **downthrust** on the shaft is due to its cross-sectional area being exposed to a large pressure differential that equals the difference between the pump's discharge and suction pressures. This load is directly transferred to the protector and has to be taken into account when selecting the right protector.

The **benefits** of floating impeller design include

- the elimination of having to fix the impellers axially, a time-consuming work requiring high precision,
- the building of pumps with several hundreds of stages is possible,
- smaller-capacity thrust bearings are needed in the protector section because most of the hydraulic thrust is absorbed inside the pump
- lower investment cost, as compared to fixed impeller pumps.

Limitations are related to the load-bearing capacity of available thrust bearings, which, in turn, are restricted by the annular space available in different casing sizes:

- such pumps are usually manufactured in smaller diameters, up to a size of about 6",
- the recommended operating range is somewhat narrower than that for the same pump with fixed impellers.

Fixed impellers are locked on the pump shaft in the axial direction, their hubs being in contact, see Fig. 3.8. As the stages are not equipped with downthrust washers, the axial thrust developed on them must be fully carried by the unit's main thrust bearing in the protector section. Because of this, these pumps may be operated outside their normal operating ranges without much damage to the stages. Pumps with such stages are often called "**compression pumps**" and are commonly used in larger-sized ESP units (greater than 6" in diameter) capable of producing large volumes of liquids. As already mentioned, they may have a wider operating range than pumps of

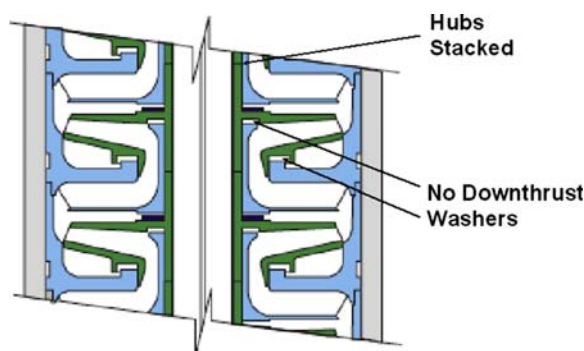


FIGURE 3.8 Construction details of fixed impeller stages.

the same type with floating impellers. Generally, they tolerate pumping of fluids containing abrasives much better than floating impeller type pumps.

The thrust to be carried by the main thrust bearing of the ESP unit is much greater than that for floater pumps. The total axial load has two components: (1) the shaft load due to the differential pressure acting on the shaft cross-sectional area and (2) the sum of the axial loads occurring in the pump stages. Clearly, thrust bearings of much greater capacity than that for floater pumps are required for proper operation.

Limitations of fixed impeller “compression” pumps include the following:

- they are more difficult to manufacture because impellers must be fitted very precisely along the pump shaft,
- investment costs are higher because of manufacturing requirements,
- the maximum number of stages in one pump is limited to about 80–100,
- protectors with high-capacity thrust bearings must be used.

3.2.3 Pump Temperature

During its operation, the centrifugal pump **converts** mechanical energy provided by the submersible motor into pressure increase of the fluid pumped. As the efficiency of energy transformation is inevitably less than 100%, there are energy losses involved. These losses end up as **heat** generated in the pump that is absorbed by the fluid flowing up the tubing string. To find the temperature increase in the produced fluid, a balance of the heat generated and the heat absorbed at steady-state conditions must be written up.

The power exerted by the pump to lift a given amount of liquid against the operating head equals

$$P_{\text{hydr}} = 7.368 \cdot 10^{-6} H q \gamma \quad (3.3)$$

where

P_{hydr} = pump hydraulic power, HP

H = head generated by the pump, ft

q = pumping rate, bpd

γ = specific gravity of the fluid, —.

The mechanical power required to drive the pump is found by considering the pump's efficiency:

$$\text{BHP} = \frac{P_{\text{hydr}}}{\eta_p} \quad (3.4)$$

where

BHP = power required to drive the pump, HP

P_{hydr} = pump hydraulic power, HP

η_p = efficiency of the centrifugal pump, —.

The energy **wasted** in the pump is the difference between the required BHP and the hydraulic power. This is the energy converted to heat, Q , in the pump; if expressed in British thermal units per minute, we obtain the following formula:

$$Q = 42.41 P_{\text{hydr}} \left(\frac{1}{\eta_p} - 1 \right) \quad (3.5)$$

where

Q = wasted energy converted to heat, BTU

P_{hydr} = pump hydraulic power, HP

η_p = pump efficiency, —.

The heat **absorbed** by the fluid flowing through the pump can be calculated from the fluid rate, the temperature increase, and the fluid's thermodynamic properties as

$$Q = \frac{350}{1,440} c q \gamma \Delta T_f \quad (3.6)$$

where

Q = heat absorbed by the fluid, BTU

c = specific heat capacity of the fluid, BTU/lb/°F

q = pumping rate, bpd

γ = specific gravity of the fluid, —.

ΔT_f = temperature rise in the fluid, °F.

As the heat generated in the pump is equal to the heat taken by the fluid, simultaneous solution of Eqs. (3.5) and (3.6) and substitution of P_{hydr} (Eq. 3.3) into the resulting equation gives the temperature rise of the fluid pumped:

$$\Delta T_f = \frac{H(1 - \eta_p)}{778 c \eta_p} \quad (3.7)$$

where

ΔT_f = temperature rise in the fluid, °F

H = head developed by the pump, ft

c = specific heat capacity of the fluid, BTU/lb/°F

η_p = pump efficiency, —.

As seen from the equation, the temperature rise of the fluid pumped does not depend on the pumping rate or the fluid gravity. The effects of the governing parameters are the following:

- Fluid temperature increases proportionally with the total **head** developed by the pump.

- Pump efficiency has a definite effect; at low efficiencies, more heat is generated and the fluid is heated up more.
- Fluids with high heat capacities (water or wellstreams with high water cut) provide more cooling of the pump.

The temperature rise found from Eq. (3.7) refers to the inflow temperature of the fluid entering the pump and the outflow temperature from the pump to the tubing string. In most cases the effect of this on the operation of the pump is negligible except when fluids with extremely high viscosity are pumped.

Example 3.2

Determine the temperature rise of the fluid in the ESP if pumping (1) water ($c = 1$ BTU/lb°F) and (2) oil ($c = 0.5$ BTU/lb°F). The pump develops 7,000 ft of head and has an efficiency of 64%.

Solution

Solving Eq. (3.7) for the two cases, we get

$$\Delta T_w = [7,000 (1 - 0.64)] / (778 \cdot 1 \cdot 0.64) = 5^\circ\text{F and}$$

$$\Delta T_o = [7,000 (1 - 0.64)] / (778 \cdot 0.5 \cdot 0.64) = 10^\circ\text{F.}$$

3.2.4 Novel Pump Stage Design and Manufacturing

For most of the history of centrifugal pump applications, the design of the pump stages was more an **art** than science. Development of new stages involved an iterative design process heavily relying on laboratory measurements and the final product usually did not meet all the requirements for efficiency and reliability. This situation has completely changed since the introduction of computational fluid dynamics (CFD) program packages that permit one to **design** and test different stage geometries on the computer. This novel method allows a complete control on the hydraulic processes that occur inside the pump stage and helps designers build stages for different requirements: optimum pump efficiency, maximum head development, etc.

Nowadays, most major manufacturers take advantage of CFD methods and offer centrifugal pump stages with significantly improved characteristics than the previous designs allowed. One of the first computer-designed stages was the “**Centurion**” offered by Centrilift [5] (Fig. 3.9). The objective of the designer was to develop the highest possible head and, at the same time, to increase the pump’s resistance against sand erosion. The most significant change in the stage’s geometry is the extra-large (more than 30% greater than in conventional pumps) vane **opening**, as shown in Fig. 3.10. Sand problems are minimized by the swirl-suppression **ribs** that divert the sand-laden fluid into the main flow path of the stage, thus decreasing the erosional damage

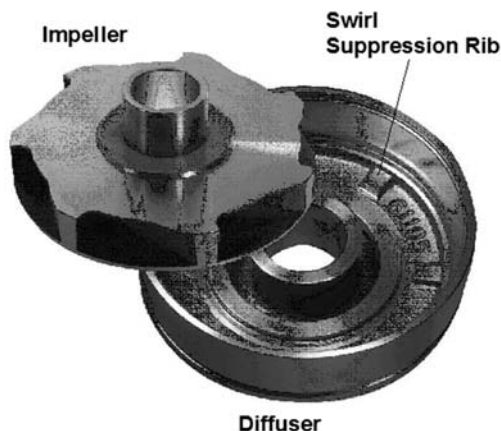


FIGURE 3.9 The “Centurion” pump stage.



FIGURE 3.10 The “Centurion” impeller.

caused by sand particles on the diffuser wall [6]. The high head developed by the Centurion stage has many advantages: **shorter** pumps are needed, the necessity of tandem pumps may be eliminated, and equipment installation time is reduced.

The advantages of CFD designs are illustrated in Fig. 3.11 that presents a comparison of two similar REDA stages, the DN2150 of the old design and the computer-designed D2400N [7]. As seen, the new stage attains an efficiency of 68% as compared to 56% of the old design; this proves the excellent hydraulic performance of the stage designed by CFD methods. The company offers several models with similarly high pump efficiencies for different production environments [8].

The materials and manufacturing processes have also improved over the years [9,10]. The first ESP stages were simple **grey iron** castings but stronger and harder materials soon followed to cope with enhanced requirements. Today the standard material of pump parts is **Ni-Resist**, an iron alloy with about 15% nickel content; it has increased hardness and improved resistance against corrosion and abrasion. The improved **Type 4** Ni-Resist contains approximately 25% nickel that further increases the metal’s corrosion resistance and allows its heat treatment, but it is rarely used because of its high cost.

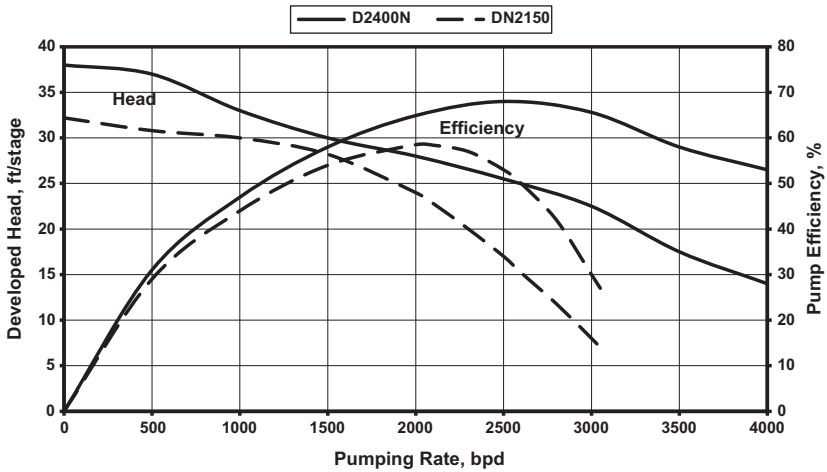


FIGURE 3.11 Comparison of two stage designs' performance.

The pursuit for better performance at lower costs resulted in the wide use of **coatings** applied to impellers and diffusers to increase their run life. Surface coatings applied include nickel, tungsten carbide, etc. All these have the disadvantage that sand erosion can remove them from the surface because sand is harder than the mostly used nickel. Diffusion coatings can eliminate the problems associated with simple coatings but require heating the parts during application and can thus distort the shape of impellers and diffusers.

The conventional manufacturing process associated with the traditional materials just discussed is sand molded **casting** in which molten metal is cast in sand **molds**. Sand molds are cheap and heat resistant and can provide the necessary surface **smoothness** for stage parts but the **complexity** of the stages that can be created is limited. Another disadvantage is that not every metal lends itself to the sand casting process; anyway, most pump stages are manufactured by sand casting everywhere today.

The latest innovation that revolutionized the industry is the introduction of **powder metallurgy** technology; the first application in pump stage manufacture was by the Russian company **Novomet** in 1991 [11]. Conventional powder technology, well known in other industries, involves mixing, molding, and sintering of metal **powders** into the required final shape. To reach the complex shapes of impellers and diffusers the process is divided into several steps:

- Powdered (finely ground) raw materials (iron, graphite, copper, etc.) are **mixed** using exact recipes.
- Properly selected parts of impellers and diffusers are injected into molds and **pressed** into the desired cold-formed shapes using high-pressure presses.

- Parts are **assembled** and joined into nonsintered blanks.
- The blanks are **sintered** (heat treated) at high temperature for extended periods while metal particles bond together to form the final solid object.
- Final **machining** finishes the manufacturing of impellers and diffusers.

The powdered metal manufacturing process produces pump stages with the following very important **improvements** as compared to stages made with the conventional sand casting method:

- More complex stage **geometries** can be manufactured especially in conjunction with CFD design.
- Geometrical **tolerances** and surface finishes are much improved.
- Optimum selection of metal **combinations** for specific pump parts is possible; pump stages for specific purposes (corrosion, erosion resistance, etc.) can be built.
- Pump stages are more **balanced** and produce lower amounts of vibrations thereby increasing the reliability of the pump. Thanks to this feature, pumps can be operated at much higher speeds (up to 6,000 RPM) than their conventional counterparts.
- The process lends itself to **cost-effective** mass production.

3.3 THE ESP MOTOR

3.3.1 Induction-Type Asynchronous Motors

3.3.1.1 Motor Construction

Most of the motors used in the ESP industry are **three-phase**, two-pole, squirrel cage **induction-type electric motors**; the basic operation of such motors was detailed in Section 2.4.4.1. These are, as detailed earlier, **asynchronous** motors and run at speeds less than the speed of the rotating magnetic field in their stators.

The construction of squirrel cage induction motors (IMs) is the simplest among electric motors. They are also the **most reliable** motors because their rotor is not connected to the electric supply. Also as these motors are the **most efficient** ones among the conventional electric motors available, they are very popular in oil field applications.

The basic construction of an ESP motor is shown in [Fig. 3.12](#) where a cross section of a common motor is depicted. Inside the motor housing and attached to it is the **stator**, a hollow cylinder made up of a great number of tightly packed thin magnetic steel discs called **laminations** that are press-fit into the housing. The shape of typical die-punched laminations (lams for short) is shown in [Fig. 3.13](#); they are thin steel discs covered on both sides with insulating coating. Pressed together, they provide the iron **core** for the stator windings; this solution prevents the creation of wasteful (heat producing) eddy currents in the stator, otherwise inevitable in stators made of solid steel. Stator

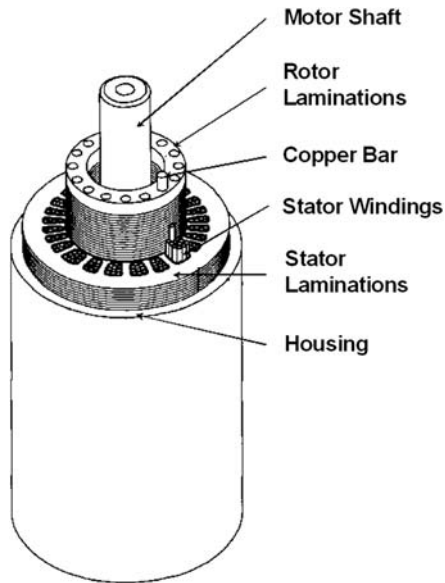


FIGURE 3.12 Construction details of an ESP motor's stator and rotor.

laminations contain a number of “**slots**” into which the electric windings are inserted by hand operations. The figure shows **closed** slots but earlier versions used open slots that did not have a continuous inner ring.

The stator slots accommodate the insulated (against wire-to-wire contact) **copper stator windings** (called **magnet wire**) connected to the AC power. There are three pairs of coils displaced at 120 degrees along the perimeter of

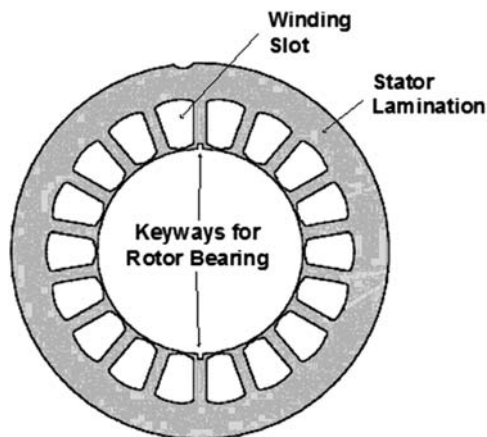


FIGURE 3.13 Closed stator lamination.

the motor, with the individual coils of each pair wound facing each other on opposite sides of the stator. The three coils are connected to the three phases in a “Y connection,” as illustrated in Fig. 3.14.

Originally, round insulated wire was used to make up the coils, but the “slot fill percentage,” i.e., the total **metal** cross-sectional area related to the slot’s area is not too high in this case. Usage of rounded “**squircular**” (intermediate shape between a square and a circle) wires increases the slot fill percentage by 18%, which means more available copper to carry motor current and more efficient use of electric power [12]. At the two ends of the lamination stack, coils make a 180-degree turn called the “**end turn**” where insulation against phase-to-phase faults is provided.

To prevent electric failures in windings, the motor must have a sophisticated **insulation system** including

- insulation of the individual wires making up the windings,
- insulation between the windings and the stator,
- protection against phase-to-phase faults.

When the stator is complete, it receives an additional insulation before final assembly. The traditional method is applying a thin coat of **varnish** on the magnet wire and the lamination slots. A much better and the current almost universally accepted method is the complete **encapsulation** of stator slots and end coils with **epoxy** that fills all the voids in the slots and end turns. The application of epoxy encapsulation has several benefits over a simple varnish coating:

- It improves the **mechanical strength** of windings and eliminates wire movement.
- It improves the **dielectric strength** of the windings.

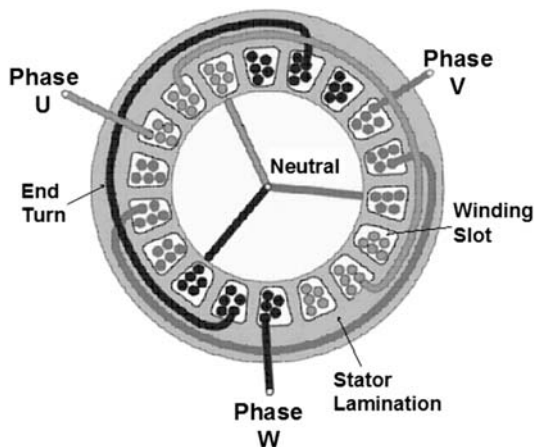


FIGURE 3.14 Y-connected windings in the stator.

- Because of the much greater **heat conductivity** of epoxy (about 300 times greater than that of the varnish) the dissipation of the heat generated in stator windings to the outside of the motor is greatly enhanced.
- It **protects** wires and end coils from contaminants.

Inside the stator and separated from it by an annular “**air gap**” is the **rotor**, consisting of magnetic rotor laminations (again to prevent eddy currents) similar to those used in the stator, refer to Fig. 3.13. Rotor laminations also contain slots (circular or shaped) into which a set of **copper bars** is inserted joined at their two ends by, and welded to, the so-called “**end rings**” (copper washers). This arrangement of connected copper bars makes up the “**squirrel cage**” inside the rotor, a schematic depiction of which is presented in Fig. 3.15. Traditional motors use **round** copper bars but high-capacity rotors have **shaped** slots and correspondingly shaped rods to increase the copper content and the power capacity of the motor. The center bore of rotor laminations has an axial **keyway** that accepts a **key** that fixes the laminations to the motor shaft in the radial direction and allows the transmission of torque created by the motor to the shaft.

In principle, rotors could be made in one piece but because of the high rotational speeds and their extreme slenderness, they would be extremely unstable radially. This is why ESP motors are manufactured with rotors made of short segments (usually 1 ft long) with radial bearings between them to stabilize the shaft’s operation.

The rotating magnetic field developed by the AC flowing in the stator windings **induces** a current in the rotor. Because of this induced current a magnetic field develops in the rotor, too. The **interaction** of the two magnetic fields turns the rotor and drives the motor shaft firmly attached to the rotor.

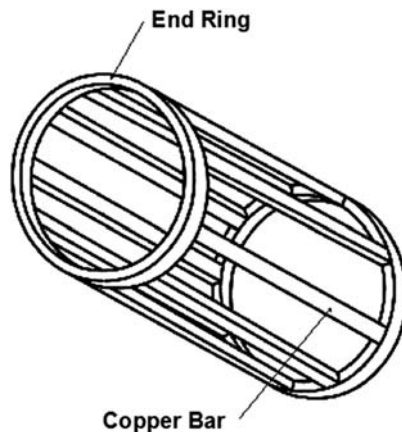


FIGURE 3.15 Construction of the “squirrel cage.”

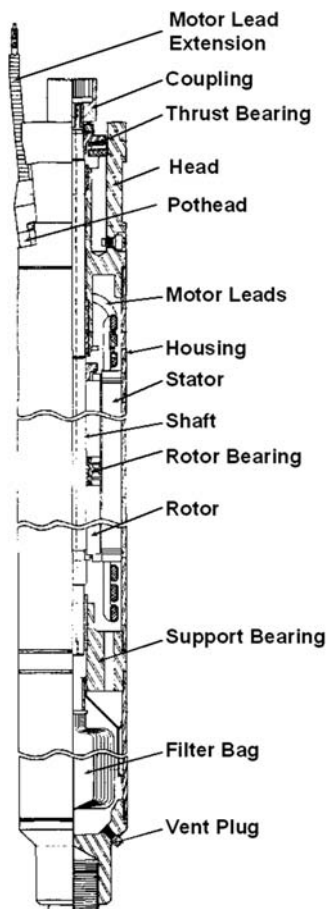


FIGURE 3.16 Main parts of an ESP motor.

The main parts of a common ESP motor are shown in Fig. 3.16. Because a single motor and its shaft can be as long as 30 ft, the elimination of radial **vibrations** is of prime importance. This is why **radial bearings** (simple journal bearings, in contrast to ball bearings used in surface motors) are located between rotor segments along the shaft's length. They provide the radial and axial support to the shaft and the rotor.

The typical construction of a **rotor bearing** (a simple journal bearing) is given in Fig. 3.17. A **sleeve** of usually bronze material is keyed to the shaft that carries the weight of the rotor laminations and turns with the rotation of the shaft. A radial **hole** in the sleeve, aligned with a similar hole in the motor shaft allows the flow of motor oil from the hollow shaft for lubrication and cooling. The bearing (often called a **T-ring**) is made of nonmagnetic materials to

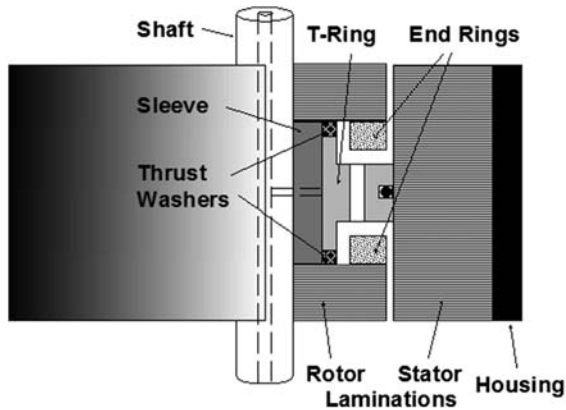


FIGURE 3.17 Schematic construction of the rotor bearing.

prevent its rotation due to the stator's magnetic field. It can be fixed to the stator laminations using different solutions, the traditional arrangement is to have stator laminations made of **bronze** inserted between the steel ones along the length of the rotor bearing. In this case the higher thermal expansion of bronze causes a tight fit between the stator and the T-ring when motor temperature increases during motor operation. Other solutions to prevent spinning of the bearing body use (1) keys inserted between the stator and the T-ring, see the keyways in a typical stator lamination in Fig. 3.13, or (2) elastomer seals as shown in Fig. 3.17.

Proper lubrication of the rotor bearing assembly is of prime importance; hence, the T-ring contains axial communication **holes** of sufficient total cross-sectional area where motor oil is circulated. Thrust **washers** of nonmetallic or metallic material are used to allow for the small vertical movement of the bearing body (T-ring).

To compensate for the axial load due to shaft and rotor weight, a **thrust bearing** of proper capacity is installed on top of the motor.

The whole motor is filled up with a highly refined **oil** of a specific gravity between 0.8 and 0.83 that provides

- the required **dielectric strength** to prevent short circuits between motor parts,
- proper **lubrication** for the bearings,
- good **thermal conductivity** required to carry the heat generated in the motor to the outside of the housing.

The motor shaft is **hollow** to allow the circulation of motor oil (1) to transfer the generated heat to the housing and (2) to lubricate the moving parts, especially the rotor bearings. The latter is facilitated by radial holes in the shaft at each rotor bearing position. A filter is also provided to remove solid particles from the lubricating oil.

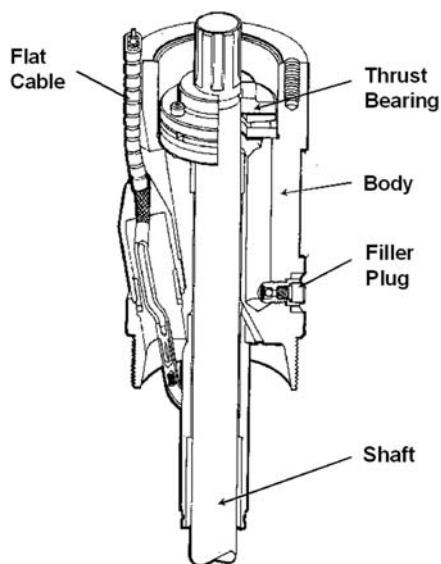


FIGURE 3.18 Tape-in connection of the ESP cable.

Electricity is supplied to the motor through the power cable from the surface. A **motor lead extension (MLE)**, usually a flat cable section) must be used along the ESP unit because of the limited annular space between the unit and the casing inside diameter. The connection of the MLE to the motor windings can be made with a simple **splicing** (“tape-in” connection), as shown in Fig. 3.18, which provides high reliability but requires relatively longer installation times. The use of a “plug-in” **connector** (see Fig. 3.19) that

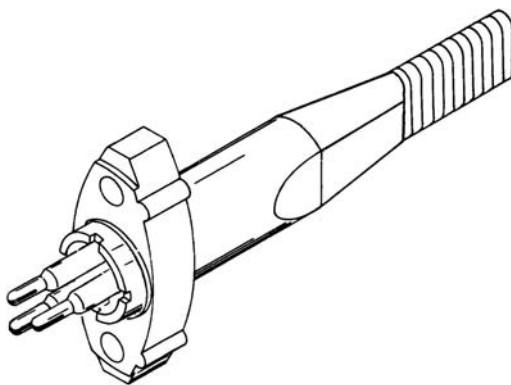


FIGURE 3.19 A pothead connector.

connects to the motor's **pothead** ensures easier installation but usually is not as reliable as the taped splicing.

Single motors are available in different sizes, usually between ODs of 3"—8", with BHP ratings in a broad range. For use in a given well casing size, i.e., for a constant motor OD, power can be increased only by increasing the length of the motor and **tandem motors** made up of several (up to three) individual motors are also available.

Tandem motors are composed of two or three standard ESP motors mechanically coupled at their shafts and fed by a single power cable. They come in different versions: **upper tandem (UT)** motors have the pothead at their head for electric connections, while **center tandem** or **lower tandem** motors are used in conjunction with an **UT** motor and have internal electric connections. Tandem motors are connected in series electrically: their voltage requirement is the sum of the individual nameplate voltages and they receive the same nameplate current. The main **rule** in making tandem motors is to select motors with **identical** power and voltage ratings; the tandem's power and voltage are doubled, whereas its current remains the same. Total power of tandem motors can go up well over 1,000 HP.

One specific case utilizes two tandem motors independently connected to two surface supply systems having their individual power cables, see [Fig. 3.20](#).

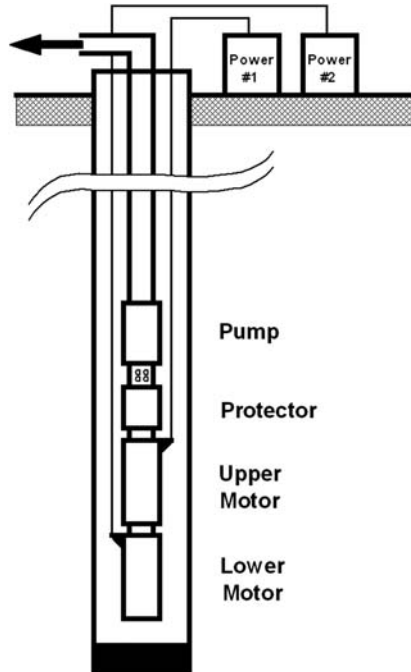


FIGURE 3.20 Two ESP motors connected parallelly to the power supply.

The two ESP motors are mechanically coupled (i.e., have their shafts connected to each other and to the rest of the downhole equipment) but work independently in an electric sense. As described in Ref. [13], motor power was reliably increased to 2,000 HP, allowing the production of 30,000 bpd liquid from several installations.

This solution has many beneficial features as compared to the conventional tandem motors connected in series to the electric supply, including

- the use of **standard** size equipment components [variable-speed drive (VSD), cable, switchboard] is sufficient,
- standard equipment is usually **readily** available from manufacturers,
- investment costs are lower.

3.3.1.2 Operational Features

ESP motors are very different from electric motors in everyday use on the surface. The most important differences [14] are listed in the following, a basic comprehension of which is necessary to fully understand the operational features of ESP motors.

- As they must be run inside the well's casing string, their **length-to-diameter** ratio is much greater than that of surface motors.
- Motor power can only be increased by increasing the **length** of the unit.
- Surface motors are usually cooled by the surrounding air, whereas ESP motors are cooled by the convective **heat transfer** taking place in the well fluid flowing past the motor.
- Because of the great difference between the heat capacities of air and liquids and the accordingly higher cooling effect, electric current densities that are 10 times higher than those in surface motors can be used in ESP motors without severe overheating.
- ESP motors have exceptionally low **inertia** and accelerate to full speed in less than 0.2 s when starting.
- ESP motors are connected to their power source by **long** well cables, where a substantial voltage drop occurs.

Motors with the same nameplate horsepower are available in different **voltage** and **current** combinations, the reason being that electric power is defined as the product of those two parameters. Thus a motor with a low voltage requirement draws a higher current and vice versa; however, both may provide the same rated electric horsepower. This feature of electric motors enables the designer to choose the right motor for every installation, the objective being the selection of a motor with a voltage–current combination assuring the **most efficient use** of electric power.

Motor **rotation** should comply with the preferred rotation of the submersible **pump**; it can be **CCW** or **CW**, the motor works exactly perfect in

either direction. Rotational direction is set and checked at equipment installation and is easily changed by changing two of the connections of the motor terminals to the phases of the power source. CW rotation has the disadvantage that threaded connections in the pump, motor, and other equipment may get **unscrewed** during operation. This must be prevented by plates welded to the outside of equipment pieces; CCW rotation prevents such problems.

Operational data of two series of motors available from a leading manufacturer are listed in **Appendix D**.

3.3.1.3 Motor Performance

IMs used in ESP service have operational features very similar to **NEMA (National Electrical Manufacturers Association) B** electric motors. A schematic torque—speed performance curve is shown in Fig. 3.21, valid at a constant input voltage. The synchronous speed of the motor is the rotational speed of the magnetic field in the stator. At speeds higher than this the motor becomes a **generator**; motoring is restricted to speeds lower than N_{synch} .

Stable motor performance is restricted to speeds higher than the speed belonging to the maximum (**breakdown**) torque value. This is because the load on the motor increases with speed and a stable common operating point of the motor and the submersible pump can be achieved in this region only.

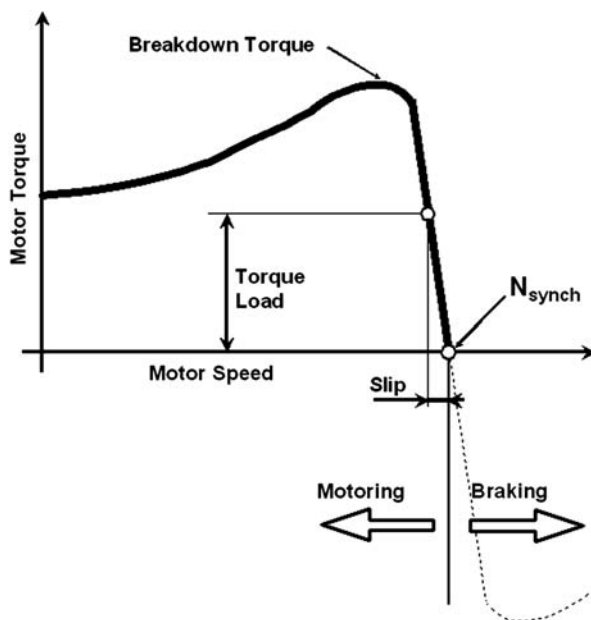


FIGURE 3.21 Typical torque—speed performance of an ESP motor at constant voltage.

The synchronous speed of ESP motors in 60 Hz service is $N_{\text{synch}} = 3,600$ RPM (see Eq. 2.33). Because the motor operates on the magnetic induction principle, the rotor always turns at speeds $N < N_{\text{synch}}$, the actual speed being the function of the torque load acting on the motor. The difference between the synchronous and actual speed is the motor's **slip**. As shown in Fig. 3.21, under normal operating conditions, motor slip is a sole function of the torque load on the motor [2].

At fully loaded conditions, ESP motors usually have a slip of 100–150 RPM, so normal rotational speeds are around 3,500 RPM. If supplied with a frequency at 50 Hz, the motor's synchronous speed becomes 3,000 RPM and its performance curve is shifted horizontally as in Fig. 3.21. The shape of the curve does not change because the constructional features of the motor have not changed, so at similar loads similar slips must occur. Hence, operational speeds around 2,900 RPM can be expected if the motor is supplied with electricity at 50 Hz where the synchronous speed is 3,000 RPM.

3.3.1.3.1 Motor Testing

Although testing procedures differ from manufacturer to manufacturer, during the developmental period, ESP motors undergo detailed testing to determine their performance parameters. Tests can be classified into three groups [15] detailed in the following.

- **Insulation/winding tests** are simple inexpensive tests to check the motor's general condition and include
 - direct current (DC) resistance measurements on the stator windings that are conducted to ensure that stator resistances are balanced within 3% of each other,
 - high-voltage insulation tests check the motor's insulation system against short-term voltage peaks,
 - testing of the motor oil for contamination and dielectric strength.
- **Idle tests** are performed on motors operating under no load conditions and can detect the electric and mechanical conditions of the motor.
 - Such measurements allow one to determine the internal losses of the motor such as (1) core losses, (2) oil friction losses, and (3) no-load heat losses.
 - Measuring the time taken for the motor to completely stop reveals the mechanical conditions of motor parts including thrust bearing, rotor bearings, etc.
- **Load tests**, perhaps the most important tests for the user, require expensive equipment to determine the performance parameters of the motor. They try to simulate the actual conditions in a well and use a dynamometer to apply a controlled torque load on the motor.
 - The parameters usually measured in a wide operating range (at several points from about 25% to about 125% of full load) are input power,

input current, revolutions per minute, motor efficiency, and power factor. The results are used to develop motor performance curves made available to customers.

- Winding temperature measurements are performed on motors running under full load and nameplate voltage and frequency. Results are used to check with recommended temperature ratings.

3.3.1.3.2 Performance Curves

ESP motor manufacturers publish typical motor performance curves representing average parameters from a number of tests. Sample **performance curves** of ESP motors, valid at nameplate voltages, are given in Fig. 3.22 as the function of motor loading [16]. Explanation of each curve follows.

- **Motor speed** drops as motor load increases. The full-load RPM of the motor is very important because it directly affects the performance of the submersible pump. Based on the **affinity laws**, it is easy to understand that a small difference in full-load speeds between two motors could result in a reduced pumping rate.
- **Motor input current** changes linearly with motor loading. It is the simplest to measure and the most useful parameter to determine the motor's actual load.
- **Input power** follows the variation of motor current.

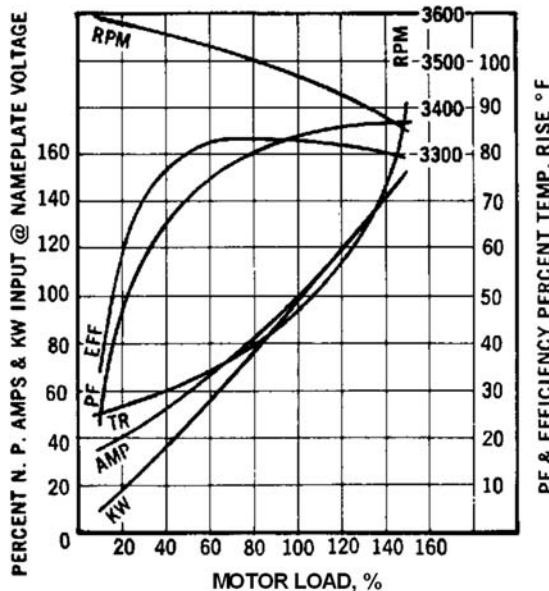


FIGURE 3.22 Average motor performance curves [16].

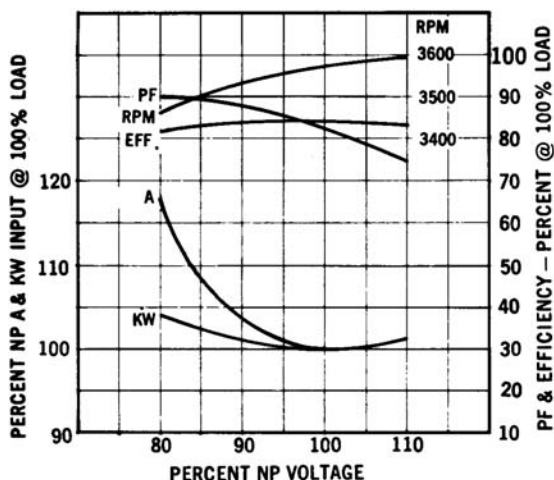


FIGURE 3.23 Average motor performance at varying input voltage [16]. NP, nameplate; PF, power factor.

- **Motor efficiency** quickly deteriorates at very low loadings but maintains its maximum around full-load conditions. As power costs constitute a major part of the total production costs in an ESP installation and are directly related to motor efficiency, use of motors with higher efficiencies can significantly improve profits.
- **Power factor** also drops at low motor loadings.
- **Temperature rise** follows the change of motor power and is shown for a fluid velocity of 1 ft/s around the motor.

The performance of the same motors under constant full load but varying **input voltages** is shown in Fig. 3.23. When the motor operates at less than nameplate voltage, its speed is reduced but the current and power drawn are significantly higher. Pump output is also reduced in accordance with the reduced speed but motor efficiency does not change much. The most significant change in motor performance under higher-than nameplate voltages is the drop in **power factor**. As there is a penalty for low power factors in most utility company's policy, it is necessary to **compensate** for low power factors to save on electricity costs.

Performance curves of typical 5.6"-OD ESP motors currently available from a different manufacturer [17] are given in Fig. 3.24. As seen, some improvement in motor efficiency values is apparent, which can be attributed to the use of up-to-date materials and technologies in the motor manufacturing process.

3.3.1.3.3 Startup Conditions

The two basic ways to start the operation of ESP equipment are (1) **across-the-line** start and (2) **soft** start [18]. The first one is the predominant, low-cost

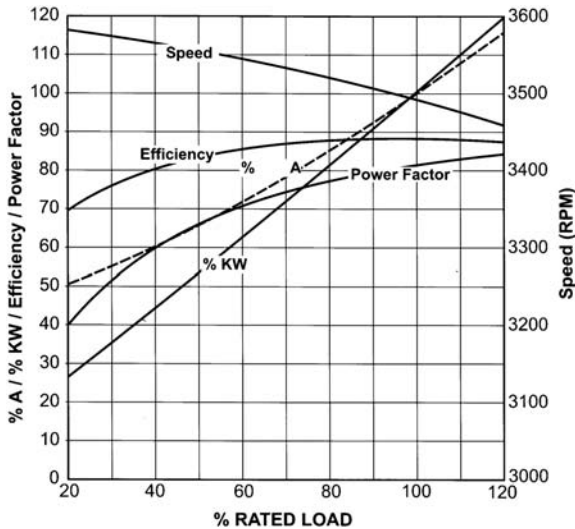


FIGURE 3.24 Average performance curves for 5.6" OD motors [9].

technique when the electric network is simply connected to the ESP system via the surface switchboard. The motor, receiving its nameplate voltage and rated frequency, immediately starts at full speed and develops a high torque. Soft starting, on the other hand, requires the use of a VSD unit; the motor accelerates quite slowly and this feature provides substantial operational benefits. The need for VSDs involves additional investment, whereas across-the-line starts do not. In the following the normal method of ESP starting is investigated first.

3.3.1.3.3.1 Across-the-Line Start The operational conditions of the ESP motor during an across-the-line startup necessitate special considerations. When the unit is started, there is an enormous instantaneous power requirement to set into motion the stagnant well fluid occupying the tubing string. Because the electric motor used has a low slip, it accelerates to its nominal speed in a fraction of a second no matter what the load is. As a result, because of the high instantaneous power requirement, the motor draws currents **several times** (four to six times) higher than its rated current during the short startup period, provided the actual line voltage stays above 95% of the rated value.

The advantages of across-the-line start are **low cost** and the high startup torque developed by the motor. The high available torque can be used to advantage in cases when the submersible pump is loaded or **plugged** with sand or other solids and needs an increased torque to start. The startup torque, however, depends on the voltage received by the motor, which, in turn, can drop below the nameplate value in deeper wells because of the high voltage

drop in the cable caused by the high inrush current. Fortunately, ESPs, as a rule, require quite low torques to start.

During the short startup period, voltage drop across the electric cable heavily **increases** because of the extremely high instantaneous current and, consequently, the voltage reaching the motor's terminals can be significantly reduced. Under unfavorable conditions the motor is **unable** to generate a sufficient torque to start or reach its operating speed. The general rule for a normal start is that about 50% of the nameplate voltage must be available at the motor terminals [19–21].

To illustrate the problem, let us take an example where motor nameplate voltage and current are 682 V and 22.5 A, respectively, and an AWG #4 cable of length 6,000 ft is connecting the motor to the surface power source. During normal operation, the voltage drop in the cable is found from Fig. E.1 in the **Appendix** at the normal motor current of 22.5 A as 9.5 V/1,000 ft. This gives a total voltage drop of 57 V in the 6,000-ft-long cable. The required surface voltage is found from the motor nameplate voltage plus the cable voltage drop and equals 739 V.

Now assume that the startup current is five times higher than the operating one and equals 112.5 A. The voltage drop in the cable for this current, found from the same table as before, has increased to 48 V/1,000 ft and the total drop in the cable reaches 288 V. Because the surface voltage is already set at 739 V, the voltage reaching the motor at startup is $739 - 288 = 451$ V. This value, expressed in fraction of the motor nameplate voltage, equals $451/682 = 66\%$; thus, the motor can safely start because it receives more than 50% of its nameplate voltage.

To ensure that the ESP motor can be started, one should check the voltage at the motor terminals during startup. Exact calculation models [20] consider the performance of the series-connected electric system consisting of the surface transformer, the power cable, and the ESP motor. The following simple procedure, although being an estimate only, permits a rapid check on the cooperation of the cable and the motor at startup conditions.

The approximate model is based on the assumption that the **inrush** voltage drop across the cable is four times the normal value. The operational voltage drop is easily calculated by multiplying the cable resistance, R_T , by the motor current. As the voltage at the motor's terminals during startup equals the nameplate voltage minus the inrush drop, its ratio to the nameplate value is found from

$$\frac{U_{\text{start}}}{U_{\text{np}}} = \frac{U_{\text{np}} - 4 I R_T}{U_{\text{np}}} \quad (3.8)$$

where

U_{start} = motor voltage at startup, V

U_{np} = motor nameplate voltage, V

I = nameplate motor current, A

R_T = cable resistance, ohms.

As discussed previously, no startup problems will occur if this ratio is greater than 0.5 because in this case, more than 50% of the nameplate voltage reaches the motor terminals.

Although the ESP motor's insulation system is normally designed to withstand higher than nameplate currents, the **damage** done during each startup is significant and is of a **cumulative** nature. The more number of startups a motor experiences, the more it deteriorates and its **run life** is reduced accordingly. To reach the maximum possible life of an ESP motor, it should be started only **once** at the beginning of its productive life and never be shut down. This ideal situation could be approached, if ever, only through a perfect installation design and under stable well conditions.

3.3.1.3.3.2 Soft Start As described earlier, normal starting of the electric motor has its inherent problems that can reduce the run life of the equipment. When VSDs (discussed in detail in Chapter 4) are used to control the operating conditions of ESP installations, their added benefit is that they provide the opportunity of a “**soft start**.” In this case the motor is started using a **low** frequency provided by the VSD unit to reduce the inrush current and prevent an excessive voltage drop across the electric cable. Starting currents only 1.5 times greater than the motor nameplate current can be expected, which is a drastic **reduction** as compared to the across-the-line start model.

VSD units operate at a constant voltage per frequency model, i.e., they regulate their output voltages according to the required frequencies. When controlling a soft start operation, VSDs supply the ESP motor with a low electric frequency at startup; typical starting frequencies are below 10 Hz. For example, if the motor nameplate voltage is 480 V at a rated frequency of 60 Hz, then at a starting frequency of 10 Hz the VSD supplies the motor with a voltage of 80 V. As shown, motor terminal voltages can be much lower than those experienced during across-the-line starts; motors may have difficulties in starting during complex situations. This problem is usually eliminated by the VSD's **voltage boost** function, which increases the output voltage at system startup.

During a soft start procedure, the electric frequency is continuously **increased** from the initial low value at startup. The designed operating conditions of the ESP system are reached when the electric frequency output from the VSD unit attains the design value. During this period the ESP motor's speed continuously increases and settles at the designed speed. This kind of startup procedure very substantially reduces the **loading** on the ESP system, especially that on the motor and can considerably increase the run life of the equipment.

3.3.1.4 Motor Temperature

ESP motors, like all electric motors, are very **sensitive** to operating temperature and their run lives are severely shortened if used at temperatures above

their design specifications. This is because motors contain a great number of materials (metals, plastics, composite materials, etc.), all of which have their own maximum temperature limits above which some or all their advantageous mechanical or electric properties are lost. The maximum allowed temperature of the motor, in this sense, is the **minimum** of the limits for all the materials in the motor.

Motor life is shortened not only at elevated temperatures but also by **aging**, i.e., degradation over time as a function of temperature. Materials used in the insulation of motor windings are especially prone to this harmful effect. The dielectric properties deteriorate according to **Arrhenius' rule** that says, for every 18°F (10°C) of operating temperature increase the life of the insulation material is reduced by 50%.

In light of the previous considerations, proper installation designs must ensure that no **overheating** of the electric motor occurs during operation. The responsibility of the manufacturer is to provide users with the maximum allowed inside temperature of the ESP motor.

The flow of well fluids past the motor is crucial for maintaining proper cooling of the unit. This is why in most installations the motor is set **above** the well's perforations to ensure fluid flow past the motor. The rule of thumb for proper cooling is a flow velocity greater than 1 ft/s. As seen in Fig. 3.25, **temperature rise** in the motor is at a minimum for velocities greater than this value. Also, as water has a much higher heat capacity than oil, the efficiency of motor cooling is greater for well fluids with higher water cuts. Because of the facts described previously, flow velocity around the ESP motor should be checked during installation design and special motor **shrouds** (attached to the outside of the motor and increasing its OD) must be used if the actual velocity falls below 1 ft/s.

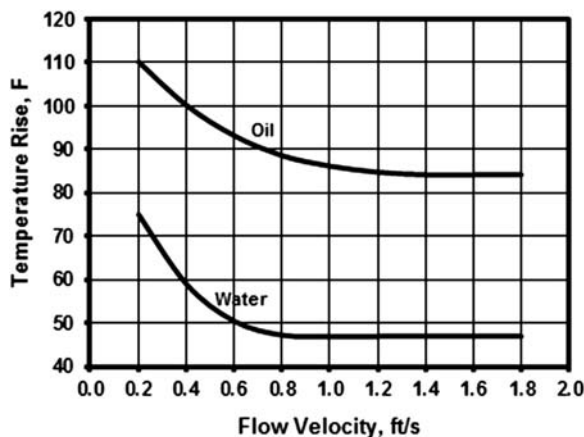


FIGURE 3.25 Motor cooling effects of oil and water at different flow velocities.

3.3.1.4.1 Heat Transfer Calculations

Generally the temperature of an operating ESP motor is the function of the following effects:

- **ambient** (geothermal) temperature
- the heat **generated** in the motor during operation
- the **cooling** effect of the wellstream flowing past the motor.

Of the previous factors, **ambient temperature** in a well depends on the geothermal gradient and the setting depth of the ESP unit. In hotter wells, therefore, motors cannot be set at the same depth as in colder wells.

The heat **generated** by the motor is closely related to motor efficiency because its magnitude is proportional to the energy lost during the conversion of electric energy to mechanical work. To find the energy wasted in the electric motor, let us first define the electric power input to the motor as follows:

$$P_m = \frac{\text{BHP}}{\eta_m} \quad (3.9)$$

where

P_m = electric power input to motor, HP

BHP = motor brake horsepower, HP

η_m = motor efficiency, —.

As the motor's output (mechanical) power equals the BHP, the power **wasted** in the motor is found by deducting BHP from the previous formula. After converting the wasted energy to **heat**, Q , and expressing it in British thermal units per minute, we get

$$Q = 42.41 \text{ BHP} \left(\frac{1}{\eta_m} - 1 \right) \quad (3.10)$$

where

Q = wasted energy converted to heat, BTU/min

BHP = motor brake horsepower, HP

η_m = motor efficiency, —.

Now the motor's BHP is expressed with the **hydraulic** power developed by the pump and the pump's efficiency (see Eq. 3.4) to arrive at

$$Q = 42.41 P_{\text{hydr}} \left(\frac{1}{\eta_m \eta_p} - \frac{1}{\eta_p} \right) \quad (3.11)$$

where

Q = heat generated in the motor, BTU/min

P_{hydr} = the hydraulic power developed by the pump, HP

η_m = motor efficiency, —

η_p = pump efficiency, —.

This formula gives the amount of heat **generated** in the motor during operation.

In steady-state conditions, i.e., when the temperature conditions are stabilized after an initial heating up, this heat is **absorbed** by the fluid flowing around the motor. The absorbed heat, Q , is found from the fluid rate and the temperature rise as given here:

$$Q = \frac{350}{1,440} c q \gamma \Delta T_f \quad (3.12)$$

where

Q = heat absorbed by the fluid, BTU/min

c = specific heat capacity of the fluid, BTU/lb/°F

q = pumping rate, bpd

γ = specific gravity of the fluid, —

ΔT_f = temperature rise in the fluid, °F.

The two heats must be equal and equating Eqs. (3.11) and (3.12) allows us to express the temperature increase in the fluid flowing past the motor. After substituting the definition of the pump's hydraulic power, P_{hydr} (see Eq. 3.3), into the resulting formula, we get the **temperature rise** in the bulk of the fluid flowing around the motor as

$$\Delta T_f = \frac{H(1 - \eta_m)}{778 c \eta_m \eta_p} \quad (3.13)$$

where

ΔT_f = fluid temperature rise, °F

H = head generated by the pump, ft

c = specific heat capacity of the fluid, BTU/lb/°F

η_m = motor efficiency, —

η_p = pump efficiency, —.

The formula, first derived by Powers [22] allows the calculation of the bulk temperature of the fluid flowing in the annulus between the casing inside wall and the ESP motor housing. It shows that temperature increase is proportional to the head developed by the pump and is very sensitive to the **efficiencies** of the pump and the motor. Low motor and/or pump efficiencies can substantially increase the heating up of the motor. On the other hand, the rise in temperature is independent of the pumping rate.

The previous formula can be used to find the average temperature of the fluid flowing in the annulus around the motor:

$$T_f = T_a + \Delta T_f \quad (3.14)$$

where

T_f = average fluid temperature in the annulus, °F

T_a = ambient (geothermal) temperature in the annulus, °F

ΔT_f = fluid temperature rise, °F.

The temperature of the motor also depends on the **cooling** effect of the well fluid flowing around the motor, which, in turn, is affected by the heat transfer in the well's annulus. Heat transfer between the motor and the produced fluid takes the form of **forced heat convection** that is responsible for an increased skin temperature on the outside of the motor housing. The amount of heat **transferred** by convection, Q , inside the fluid body is found from the familiar convection equation:

$$Q = 60 A h \Delta T_s \quad (3.15)$$

where

Q = heat transferred by convection, BTU/min

A = surface area of the pump's housing, ft²

h = convective heat transfer coefficient, BTU/(s ft² °F)

ΔT_s = skin temperature rise on motor housing, °F.

The amount of heat just calculated must be equal to the heat **generated** in the motor, as given in Eq. (3.10). From the balance of the two heats, the skin temperature increase, ΔT_s , can be found as

$$\Delta T_s = 2.7 \frac{\text{BHP}}{L_{\text{mot}} d_{\text{mot}} h} \left(\frac{1}{\eta_m} - 1 \right) \quad (3.16)$$

where

ΔT_s = skin temperature increase, °F

BHP = motor brake horsepower, HP

L_{mot} = motor length, ft

d_{mot} = motor diameter, in

h = convective heat transfer coefficient, BTU/(s ft² °F)

η_m = motor efficiency, —.

The calculation of the convective heat transfer coefficient, h , is a complex task because of the following problems [23]:

- The heat transfer occurs in the fluid body contained in the **annulus**. In such cases the use of the hydraulic diameter ($d_{\text{casing}} - d_{\text{mot}}$) is recommended.

- Fluid flow in the annulus may be **laminar** or **turbulent** and this must be taken into account when calculating the heat transfer.
- The conventional formulas available for the calculation of convective heat transfer assume single-phase **liquid** to be present in the annulus.

In the search for the solution of the problems stated, investigators agree that (except for pumping heavy, highly viscous oils [24]) flow in the annulus is usually **turbulent**. For such conditions, the formula recommended by **Powers** [22] is generally accepted, in which instead of the motor diameter, d_{mot} , the hydraulic diameter of the annulus ($d_{\text{casing}} - d_{\text{mot}}$), could be used:

$$h = 0.0378 \ k^{0.67} \ v^{0.8} \ \rho^{0.8} \ c^{0.33} \ \mu^{-0.47} \ d_{\text{mot}}^{-0.2} \quad (3.17)$$

where

h = convective heat transfer coefficient, BTU/(s ft² °F)

k = thermal conductivity of the fluid, BTU/(s ft °F)

v = fluid velocity, ft/s

μ = dynamic viscosity of the fluid, lb/(ft s)

ρ = fluid density, lb/ft³

c = specific heat capacity of the fluid, BTU/lb/°F

d_{mot} = motor outside diameter, in.

Calculation of the heat transfer coefficient requires an **iterative** scheme because almost all the parameters in Eq. (3.17) vary with temperature. The solution is to assume a temperature rise ΔT first and then calculate the thermodynamic parameters (thermal conductivity, density, viscosity, specific heat) of the fluid at the average conditions in the annulus. After finding the value of h , the actual temperature increase is found from Eq. (3.16). This is then compared to the assumed value and, if necessary, a new iteration step is performed until convergence is found.

Using the convective heat transfer coefficient, h , calculated from the previous formula the skin temperature rise of the ESP motor is found; finally the motor skin temperature is calculated as

$$T_s = T_a + \Delta T_f + \Delta T_s \quad (3.18)$$

where

T_s = motor skin temperature, °F

T_a = ambient (geothermal) temperature in the annulus, °F

ΔT_f = temperature rise in the fluid, °F

ΔT_s = skin temperature rise on motor housing, °F.

The schematic temperature distribution around and inside an ESP motor is shown in Fig. 3.26. Outside the casing string the **ambient** geothermal temperature, T_a , exists, inside the annulus the bulk temperature of the flowing fluid is T_f , while the motor skin temperature equals T_s . Temperature inside the motor

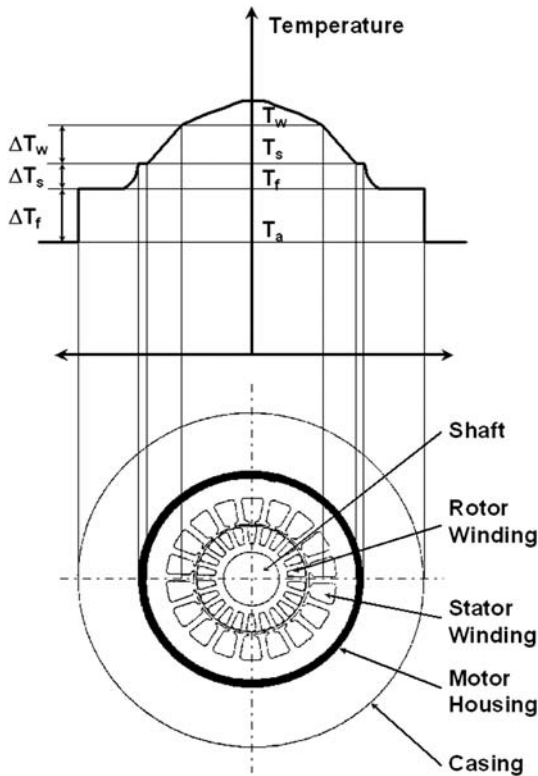


FIGURE 3.26 Temperature distribution around and inside an ESP motor.

is greater than that at the outside, reaching its maximum somewhere in the motor shaft. The average motor winding temperature, T_w , is normally assumed to increase by $\Delta T_w = 30\text{--}40^\circ\text{F}$ above motor skin temperature. In the axial direction the highest temperatures exist at the **head** of the motor.

Example 3.3

Calculate the steady-state skin temperature of an induction-type ESP motor for the following conditions.

Pump Head	9,000 ft
Motor OD	4.5"
Motor BHP	60 HP
Motor Efficiency	0.75
Motor Length	15 ft
Pump Efficiency	0.60
Water Cut	0%
Bottomhole Temp.	150°F
Oil Gravity	25 API
Fluid velocity	1 ft/s

Solution

First let us assume the temperature rise in the annulus to be $\Delta T_f = 13^\circ\text{F}$. Average temperature in the annular fluid is thus

$$T_{\text{avg}} = 150 + 13/2 = 156.5^\circ\text{F}.$$

Specific heat of the light oil at this temperature is found to be $c = 0.5 \text{ BTU/lb}^\circ\text{F}$.

Calculated temperature rise, according to Eq. (3.13), is

$$\Delta T_f = 9,000 (1 - 0.75) / (778 \cdot 0.5 \cdot 0.75 \cdot 0.6) = 12.9^\circ\text{F}.$$

Because this is very close to the assumed value, no iteration is needed and the fluid average temperature is found from Eq. (3.14):

$$T_f = 150 + 13 = 163^\circ\text{F}.$$

For calculating the heat convection in the annulus, first assume a motor skin temperature increase $\Delta T_s = 140^\circ\text{F}$. The average temperature in the annulus then becomes

$$T_{\text{avg}} = T_f + \Delta T_s/2 = 163 + 70 = 233^\circ\text{F}.$$

The required thermodynamic parameters of the oil at this average temperature, as found from the appropriate tables or computer calculations, are listed here:

$$k = 1.9\text{E-}5 \text{ BTU/(s ft }^\circ\text{F)}$$

$$c = 0.5 \text{ BTU/lb}^\circ\text{F}$$

$$\rho = 52.3 \text{ lb/ft}^3$$

$$\mu = 2.19\text{E-}3 \text{ lb/(ft s)}.$$

Now the calculation of the convective heat transfer coefficient is possible from Eq. (3.17):

$$\begin{aligned} h &= 0.0378 \cdot 1.9\text{E-}5^{(0.67)} \cdot 1^{0.8} \cdot 52.3^{0.8} \cdot 0.5^{0.33} \cdot 2.19\text{E-}3^{(-0.47)} \cdot 4.5^{-0.2} \\ &= 6.43\text{E-}3 \text{ BTU/(s ft}^2\text{F)}. \end{aligned}$$

The motor skin temperature increase is calculated from Eq. (3.16):

$$\Delta T_s = [2.7 \cdot 60 / (15 \cdot 4.5 \cdot 6.43\text{E-}3)] (1/0.75 - 1) = 124.4^\circ\text{F}.$$

As the calculated temperature increase is different from the assumed value, a new iteration step is required. The new assumed value is $\Delta T_s = 124.4^\circ\text{F}$. The average temperature in the annulus then becomes

$$T_{\text{avg}} = T_f + \Delta T_s/2 = 163 + 62.2 = 225.2^\circ\text{F}.$$

The required thermodynamic parameters at this average temperature are

$$k = 1.9\text{E-}5 \text{ BTU/(s ft }^\circ\text{F)}$$

$$c = 0.5 \text{ BTU/lb}^\circ\text{F}$$

$$\rho = 52.4 \text{ lb/ft}^3$$

$$\mu = 2.49\text{E-}3 \text{ lb/(ft s)}.$$

The new convective heat transfer coefficient is found from Eq. (3.17):

$$h = 0.0378 \cdot 1.9\text{E-}5^{(0.67)} \cdot 1^{0.8} \cdot 52.4^{0.8} \cdot 0.5^{0.33} \cdot 2.49\text{E-}3^{(-0.47)} \cdot 4.5^{-0.2}$$

$$= 6.08\text{E-}3 \text{ BTU}/(\text{s ft}^2\text{°F})$$

The skin temperature increase is calculated from Eq. (3.16):

$$\Delta T_s = [2.7 \cdot 60 / (15 \cdot 4.5 \cdot 6.08\text{E-}3)](1/0.75 - 1) = 132^\circ\text{F}.$$

As the calculated temperature increase is different from the assumed value, a new iteration step is required. The new assumed value is $\Delta T_s = 132^\circ\text{F}$. The average temperature in the annulus then becomes

$$T_{\text{avg}} = T_f + \Delta T_s / 2 = 163 + 66 = 229^\circ\text{F}.$$

The required thermodynamic parameters at this average temperature are

$$k = 1.9\text{E-}5 \text{ BTU}/(\text{s ft °F})$$

$$c = 0.5 \text{ BTU}/\text{lb°F}$$

$$\rho = 52.3 \text{ lb/ft}^3$$

$$\mu = 2.35\text{E-}3 \text{ lb/(ft s)}.$$

The new convective heat transfer coefficient is found from Eq. (3.17):

$$h = 0.0378 \cdot 1.9\text{E-}5^{(0.67)} \cdot 1^{0.8} \cdot 52.3^{0.8} \cdot 0.5^{0.33} \cdot 2.35\text{E-}3^{(-0.47)} \cdot 4.5^{-0.2}$$

$$= 6.2\text{E-}3 \text{ BTU}/(\text{s ft}^2\text{°F}).$$

The skin temperature increase is calculated from Eq. (3.16):

$$\Delta T_s = [2.7 \cdot 60 / (15 \cdot 4.5 \cdot 6.2\text{E-}3)](1/0.75 - 1) = 129^\circ\text{F}.$$

The calculated and the assumed values are close enough (computer calculations can achieve better accuracies) and the motor skin temperature is found from Eq. (3.18) as

$$T_s = 150 + 13 + 130 = 293^\circ\text{F}.$$

By comparison, if the well produced only water, the motor skin temperature would be $T_s = 167^\circ\text{F}$ only; this shows that water is a much better coolant than crude oil.

The determination of more exact temperature conditions in ESP motors requires advanced heat transfer calculations and/or the use of **CFD** techniques [25,26]. Results of such studies proved that the use of **epoxy** in stator slots greatly improves heat transfer inside the electric motor and thus reduces the temperature of stator windings. Lower stator temperatures increase motor run life because the service life of insulating materials is increased; motors for

higher-temperature service are always manufactured with epoxy instead of varnish.

Studies of internal temperature distribution in ESP motors [27,28] revealed that motor **winding** temperatures can effectively be reduced by decreasing the thermal **resistance** to heat transfer across the different parts of the motor. Motor components close to the housing play the biggest role, including the thin oil **film** between the housing and the stator laminations. Based on these findings, improved motor designs try to increase the heat transfer in this oil film as well as through the motor housing; high-temperature motors designed according to this principle are described in Chapter 4.

3.3.1.4.2 Allowed Motor Temperature

The calculation model described in the previous section allows one to find the temperature distribution around the **head** of the ESP motor. This is where skin temperature, T_s , is at its **maximum** because the fluid flowing at that level contains the heat generated by the entire motor. Here the cooling effect of the annulus fluid is at its **minimum** and the top of the motor is subjected to the greatest **heat load**.

The temperature inside the motor increases radially from the housing to the shaft, as shown in Fig. 3.26. Although temperature reaches its maximum somewhere around the center of the motor shaft, the temperature in the stator **windings** is considered to be the **critical** factor. This temperature is usually calculated from the motor skin temperature (Eq. 3.18) as follows:

$$T_w = T_a + \Delta T_f + \Delta T_s + \Delta T_w \quad (3.19)$$

where

T_w = motor winding temperature, °F

T_a = ambient (geothermal) temperature in the annulus, °F

ΔT_f = fluid temperature rise, °F

ΔT_s = skin temperature rise on motor housing, °F

ΔT_w = temperature rise from the motor housing to the windings, °F.

The difference between motor skin temperature and stator winding temperature, ΔT_w , depends on many factors, the most important being the **thermal conductivity** of the insulating material used in the manufacturing of stator windings. The use of **epoxy encapsulation**, as compared to a simple **varnish coating** applied earlier, has enormously increased the thermal conductivity inside the motor. Heat flow toward the motor housing is therefore improved, allowing the dissipation by well fluids of a greater amount of the heat generated in the motor.

At average conditions, temperature increases from the motor housing to the stator windings of $\Delta T_w = 30\text{--}40^\circ\text{F}$ [22] are suggested. Actual values, however, heavily depend on the loading of the motor and increase with higher

loads. ESP motor manufacturers should make such data available [29] along with other motor performance parameters. The maximum allowed insulation temperatures, T_w , vary with different manufacturers but values above 400°F are common.

At extremely high temperatures, the ESP motor must be **derated** and must be run at less than full load to prevent overheating and damage to its windings. The maximum allowed well temperature changes with the motor **loading** and the **water cut** of the well fluid, as shown in Fig. 3.27 for an example unit. Because of the great amount of heat generated in the motor, the maximum allowed well temperature rapidly decreases with increased motor loadings. As seen, water is a much better coolant than oil and wells with higher temperatures can be easily lifted when the water cut approaches 100%.

3.3.1.5 High-Performance Motors

The quest for reducing operating costs resulted in more efficient and powerful motors that utilize the electric energy better than conventional designs. The main improvements [30] in motor manufacturing include the following:

- All stator laminations are made of **steel** and have a **closed-slot** design.
- The magnet wire is fully encapsulated using **epoxy**. This modification alone may increase the insulation life of the motor by 2 years on average.

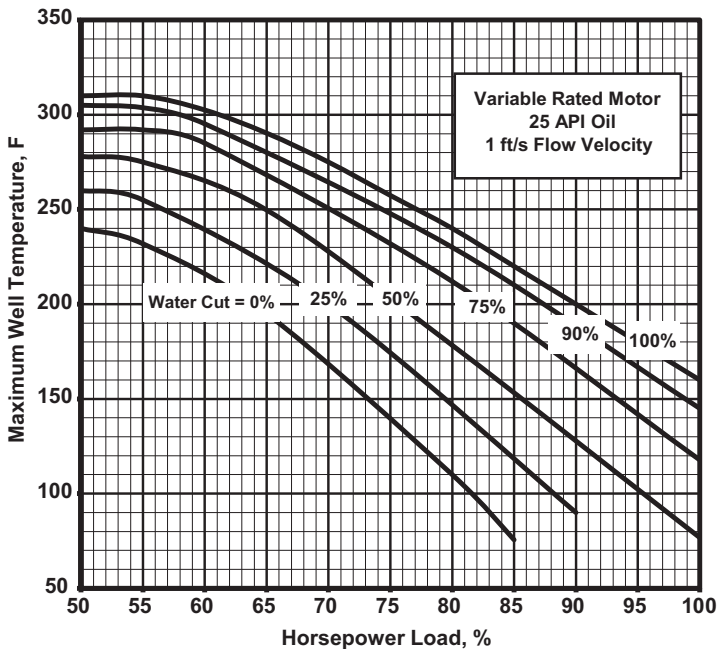


FIGURE 3.27 The effect of motor loading on allowed well temperature.

- Rotor bars are **shaped** (coffin shape) instead of round, increasing the copper content of rotor slots by about 20%; motor power is increased.
- Rotor bearings are manufactured from nonmagnetic material with increased axial **length** to eliminate spinning of the bearing.
- The bronze sleeve in the rotor bearing is replaced with a harder material such as **tungsten carbide** that offers greater mechanical stability and better heat transfer.
- Plug-in **pothead** connection of the MLE to the motor reduces installation time and increases reliability.

The Baker Hughes **SP Series** motors [31] and the Summit **Corsair** motors [32] offer the features just discussed and provide superior performance as compared to conventional motors. The REDA **Maximus** motors, in addition to these, come prefilled with the proper motor oil and have a compensating shipping cap that ensures that the right amount of oil is always present [33]. This feature makes running the equipment in the well shorter and more reliable.

3.3.2 Permanent Magnet Motors

3.3.2.1 Introduction

Before the features of permanent magnet motors (PMMs) are discussed in detail, the most important operational and constructional differences between PMMs and induction-type asynchronous electric motors (IMs) must be understood [34]. These can be summarized as follows.

- The main constructional difference is in the PMM's rotor that contains, instead of the "squirrel cage" structure of an IM, permanent magnets (PMs) to provide the magnetic flux in the rotor. The stators of the two motors are practically similar. However, the PMM's stator windings create a shallower magnetic field that does not penetrate down to the shaft of the rotor thus saving on energy consumption.
- A PMM used in ESP service, in contrary to an IM, cannot be connected directly to the three-phase power source, a special motor controller must be used at all times. Although PMMs are brushless synchronous AC motors that could be operated using a sinusoidal power source the need for additional equipment (rotor position sensor, etc.) would make this solution impossible in an oil well. For this reason, only sensorless motor control from the surface is feasible in ESP operations.

3.3.2.2 Motor Controllers for Permanent Magnet Motors

The PMMs in ESP service belong to the **brushless synchronous** AC motors. The basic operational principle of these motors is, like that of all electric motors, the interaction of two magnetic fields: the stator's and the rotor's.

The poles of same polarity (N–N or S–S) repel, whereas poles of different polarity (N–S or S–N) attract each other, and this makes the rotor to turn around. It is easy to understand that the torque developed on the motor shaft is maximum when the stator and the rotor magnetic fluxes are at right angles to each other. As development of the maximum torque is the objective of any motor design, it must be ensured that the two fluxes are kept **perpendicular** to each other.

In a PMM the magnitude of the rotor's magnetic flux provided by the PMs is constant (neglecting the stator's reaction) and the **direction** of flux turns with the rotor's movement. The direction of stator flux, on the other hand, depends on the AC connected to the stator coils and can be **adjusted** by shifting the phase of the applied voltages. So stator flux can be made to fall at right angles to the rotor flux. To achieve this condition the exact knowledge of the **position** of the rotor at any moment is necessary. Rotor position can be sensed, for example, by the Hall effect sensors that are attached to the motor frame and provide signals representative of the rotor position. Control of PMMs used on the surface is accomplished according to the model just described; they are driven by continuous sinusoidal three-phase power. The basic limitation of this solution in ESP service is that the motor with the sensors is downhole, while their signal must be led to the surface where the motor controller is located. The necessary cabling and other complications make this solution unfeasible in oil well environment.

It must be obvious from the previous discussion that the only feasible way to control a PMM in an oil well is by **sensorless** control (without any physical sensors) where all control functions are accomplished at the **surface** with no modifications to the downhole motor. Currently there are two different ways to control PMMs: scalar and vector control.

Although PMMs are often used at different frequencies, the following discussions suppose operation of the motors at a constant electric frequency and give details of the different motor controllers. If the use of the motor at variable frequencies is desired then special devices called VSDs must be used; their features and operation will be detailed in Section 4.7.2.

3.3.2.2.1 Scalar Control

One widely used solution in the industry utilizes controllers that connect only **two** of the three phases at any time to the motor's terminals [35,36]. Fig. 3.28 presents the rectangular **voltages** applied to the three phases in function of the electric revolution, which in a two-pole motor is equal to the mechanical revolution of the rotor (In multipole motors the number of electric revolutions is greater than the mechanical one by the number of poles.). As seen in the figure, motor operation follows a “**six-step**” fashion with the length of one step being 60 degrees of rotation. In each step, only two phases are energized, the third one is idle, i.e., not connected to the power

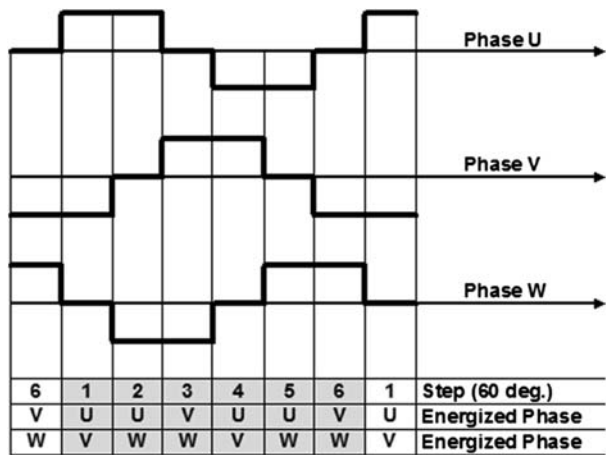


FIGURE 3.28 Typical three-phase voltage waveforms in a permanent magnet motor.

source. This six-step pattern (the shaded part in the figure) is repeated perpetually while the motor is operated.

Because at any time only two phases are conducting current and the third winding is **open**, this situation makes it possible to utilize the idle phase for detection of rotor position. This is made possible by the fact that the **shape** of the voltage induced by the rotating PMs (called the **back-EMF**) positively indicates the **position** of the rotor.

Sensorless determination of rotor position is done by sampling the **induced voltage** [back electromotive force (EMF)] caused by the rotor in the idle coil. PM rotors are designed so that the shape of the voltage induced into the stator winding due to rotor movement is **trapezoidal**. The position detector detects the **zero crossing** points of the back-EMF. Based on this signal the motor controller calculates the exact position of the rotor and controls the **switching** on and off of the phases to the motor's stator in order to keep the magnetic poles of the stator and rotor fields perpendicular. This condition, as described earlier, is the prime requirement of maximizing the torque at the motor shaft.

The variation of back-EMF and the stator current in each phase is schematically depicted in Fig. 3.29. As discussed, the motor controller, using the accurate rotor position, is able to change the **phasing** of stator currents so that current and back-EMF are always **in phase**. In that case the angle between the stator flux and the rotor flux is kept close to 90 degrees and the generated torque is at a maximum. As seen from the figure, the torque developed by the different phases follows each other so that the **net** torque on the motor shaft is quite constant. This kind of motor operation ensures a very low fluctuation of output torque, called "**torque ripple**."

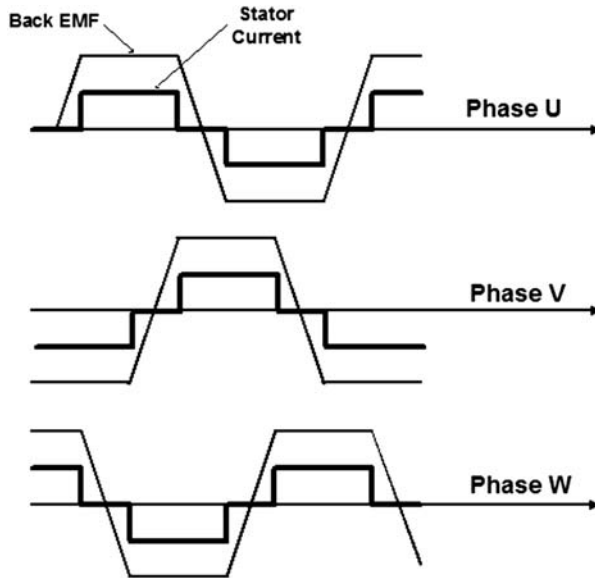


FIGURE 3.29 Variation of stator current and back electromotive force (EMF) in a permanent magnet motor.

The previously described operation of PMMs is available through the utilization of special motor controllers that use three-phase power at a constant frequency and have a rectifier and an inverter section that controls the switching on and off of the different phases according to signals received from the idle phase's back-EMF value. Controller output is voltage pulses, their magnitude is proportional to the required frequency, the voltage/frequency ratio is kept constant. Under those conditions the motor becomes a **constant-torque, variable-speed** device.

The motor controller needs to have special built-in operational behavior during motor **startup** because this control method is not self-starting. This is because the back-EMF voltage is **zero** when the motor is at **rest** and it becomes significant from a certain rotor angular velocity only. As motor control is based on the observed back-EMF values, it can be used only after the motor is at a certain minimal speed, about 5% of the motor's nameplate rotational speed. Motor starting is done by an "open loop" procedure when stator magnetic field is applied with an increasing frequency profile until the rotor's magnetic field is attracted to the stator field and the rotor starts to turn. After a short stabilization period the rotor speed reaches a threshold value when the back-EMF can already be detected; from then on, the controller switches to normal synchronous operation.

The inherent **limitations** of scalar control of PMMs are the use of long electric cables and high rotational speeds because the accuracy of rotor position determination by the back-EMF method significantly decreases in such cases. The field-oriented vector control model eliminates these problems, as discussed in the following.

3.3.2.2.2 Vector Control

Vector control (a.k.a. sensorless field-oriented control) of PMMs treats the whole electric system of the ESP unit in its **entirety** including the motor, the controller, the cable, and the transformer. This is why proper operation requires all functional data of these components to be input in the controller's memory.

In vector control applications, **all** three electric phases with **sinusoidal** voltages are connected to the motor at all times. In this case, too, it is necessary that the magnetic field of the PM and that of the stator be at right angles to provide the maximum possible torque at the motor shaft. In a PMM the **magnitude** of the rotor's magnetic flux provided by the PMs is constant and the direction of flux turns with the rotor's movement. The **direction** of stator flux, on the other hand, depends on the ACs running in the stator coils and can be **adjusted** by shifting the phase of the applied voltages. So stator flux can be forced to fall at right angles to the rotor flux. To achieve this condition the exact knowledge of the **position** of the rotor at any moment is necessary.

Motor controllers using sensorless vector control detect the position of the rotor by using sophisticated electronic circuits (microprocessors). They control the magnetic **flux vector** in the stator that rotates in phase with the current creating it. The two components of motor current are (1) **magnetizing** (inductive) current that produces the magnetic flux in the stator and (2) the **resistive** current that produces the torque at the motor's shaft. Magnetizing current always lags the voltage by 90 degrees and the torque producing current is always in phase with the voltage.

The algorithm used in vector control units is based on the decomposition of the stator current into its two **components** and then to control those components **separately**. For this purpose the currents in the three phases are projected, using special software, to a rotating rectangular coordinate system. The *d* (direct) axis of this coordinate system is fixed to the rotor, whereas the *q* (quadrature) axis is perpendicular to the direct axis. The net stator current is then decomposed into its two components on these axes, with the *d* axis current corresponding to the field current and the *q* axis current corresponding to the armature (stator) current. The recommended value for the **torque producing** current, I_q , is set to provide the required torque on the motor shaft, whereas the recommended **magnetizing** current, I_d , is set to zero because the required magnetic flux is supplied by the PMs of the rotor. The microprocessor then checks the **deviation** of the actually measured currents from the

recommended values at regular (very short) intervals and calculates the necessary voltages in the three phases to provide the desired I_q and I_d currents. By applying the calculated voltages to each phase, the motor current and magnetic flux in the stator will attain the recommended value and the motor will develop the required torque. This kind of operation helps maintain the system's power factor at a maximum that very significantly improves the power efficiency of the ESP system.

Vector controllers utilize complex software packages that, in addition to basic motor control, provide the solution to many problems in ESP operations. System startup with stuck pumps, soft start, elimination of gas lock conditions, etc. are accomplished using preprogrammed procedures [37]. The controllers usually provide advanced safety features and easy connectivity to SCADA (supervisory control and data acquisition) systems.

The advantages of vector control over scalar control are much higher power efficiency, higher power factor (around 0.95), and stable operation on long cables and at higher operating speeds. However, the use of vector control necessitates very accurate current measurements so exact input data on motor, cable, and transformer behavior is vital.

3.3.2.3 Constructional Details

As discussed in Section 2.4.4.2, the **stators** of PMMs are identical to those of IMs. Because PMMs, compared to IMs, have rotors with **smaller** diameters, there is more radial space in the motor for stator windings. Therefore stator slots can be made larger and this permits placing **more** copper wires in the windings; thus, stator losses decrease and the power density in the stator increases. An added feature is that the magnetic field created by the stator penetrates only to the depth of the rotor magnets and not to the motor shaft as in IMs. These features increase the power density in the air gap and the efficiency of the PMM.

PMMs contain **rotors** equipped with PMs made from **rare earth** metals. Such magnets have at least 10 times higher maximum energy density than traditional ferrite magnets (25–32 kJ/m³ for ferrites, 190–240 kJ/m³ for SmCo, and 290–400 kJ/m³ for sintered NdFeB); the strongest magnets are made from Neodymium Iron Boron (NdFeB) metal [34]. These magnets provide the magnetic flux in the rotor without any outside energy requirement; the distribution of magnetic flux lines created by a four-pole rotor is illustrated in Fig. 3.30. In comparison, IMs take electric energy from the stator's magnetic field to induce an electric current in the “squirrel cage” of the rotor, which, in turn, creates the rotor's flux necessary to operate the motor. This difference alone guarantees that PMMs are more **efficient** than IMs in converting electric energy to mechanical work.

PMMs, just like IMs, have rotors made up of individual sections of about 1 ft length each. The rotors, in contrary to those in IMs, do not contain

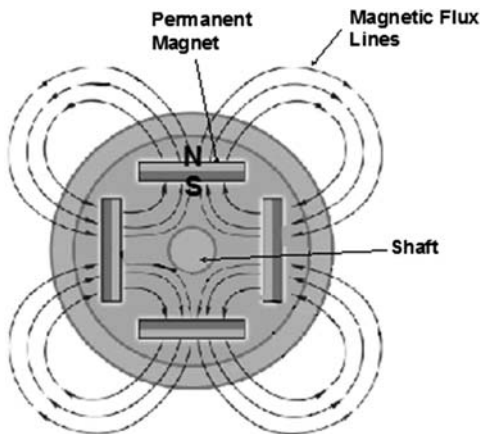


FIGURE 3.30 Magnetic flux lines created by a four-pole permanent magnet rotor.

laminations, e.g., an iron core; the PMs may be surface mounted or embedded in the body of the rotor. The surface of the magnets is normally metal coated or covered by some resin to prevent corrosion of the rear earth material.

The most popular rotor constructions (surface or interior mounted) are illustrated in Fig. 3.31. For ESP service, motors with **two** or **four poles** are normally used; increasing the number of poles decreases the speed of the

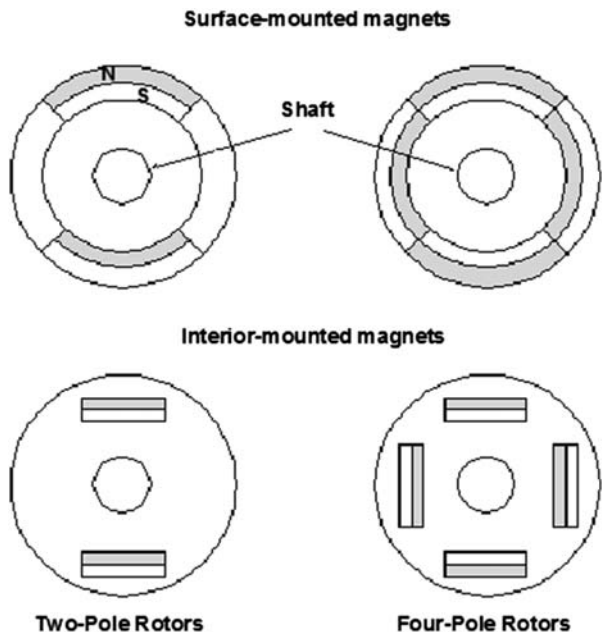


FIGURE 3.31 Some typical rotor configurations in permanent magnet motors.

motor. For low-speed service, like in progressing cavity pump (PCP) installations, 10-pole rotors are used.

The PMs used in PMMs are about 10 times stronger than traditional ferrite magnets so they provide an exceptionally strong magnetic flux in the motor's air gap. As motor power is proportional to the magnitude of magnetic **flux**, PMMs generate much greater powers than their induction-type counterparts of the same size. This is clearly seen if the power capacities of one **single** rotor of the two motor types are compared. The traditionally used rotor in IMs with **round copper bars** develops a power of 5 HP, improved models with **shaped bars** provide 10 HP, whereas PM rotors reach 22 HP for the same length [38,39]. This feature of PM rotors allows building **shorter** motors with the same power capacity; motor length and weight are reduced, as compared to IMs.

The synchronous speed of PMMs found from Eq. (2.33) is repeated here:

$$N_{\text{synch}} = \frac{120 f}{p} \quad (3.20)$$

where

N_{synch} = synchronous speed, RPM

f = frequency of the AC power, Hz

p = number of poles in the stator, —.

As seen, the speed of PMMs is proportional to the frequency of the electric power and inversely proportional to the number of poles in the rotor. According to their speed ranges, PMMs used in the petroleum industry can be classified in the following groups:

Group	Speed Ranges (RPM)	Applications
Low Speed	100–1,500	PCP operations
Standard Speed	2,400–4,200	ESP operations
High Speed	2,400–6,000	ESP operations
Ultrahigh Speed	up to 12,000	Special ESP

Motor speeds for PMMs used in the ESP industry for the most frequently used pole numbers are given in Table 3.1 in function of the electric frequency. As seen, use of four-pole motors in the 50–200 Hz AC frequency range provides motor speeds that match the possible pumping speeds of submersible centrifugal pumps, especially those manufactured using the powdered metal metallurgy process.

3.3.2.4 Operational Features

PMMs are **synchronous** machines so their speed does not change with the torque load, they keep their speed in the normal torque loading ranges. This behavior is shown in Fig. 3.32 where the motor speed equals the synchronous speed, N_{synch} at the actual torque load. Under the same loading an

TABLE 3.1 Synchronous Speeds of Permanent Magnet Motors		
Electric Frequency (Hz)	Motor Speed (RPM)	
	Two-Pole Motors	Four-Pole Motors
50	3,000	1,500
60	3,600	1,800
100	6,000	3,000
133	8,000	4,000
150	9,000	4,500
167	10,000	5,000
183	11,000	5,500
200	12,000	6,000

asynchronous motor, like the IMs described in [Section 3.3.1](#), runs at a speed less than the synchronous speed. The difference is the “slip” of the motor that changes with the load.

The operation of PMMs can be described by two formulas that determine their performance. The first involves the relation between the motor **current** and the **torque** developed and can be written [\[34\]](#) in the following schematic form:

$$T = k_l \ \Phi \ I \tag{3.21}$$

where

T = torque developed by the motor
 k_l = conversion constant

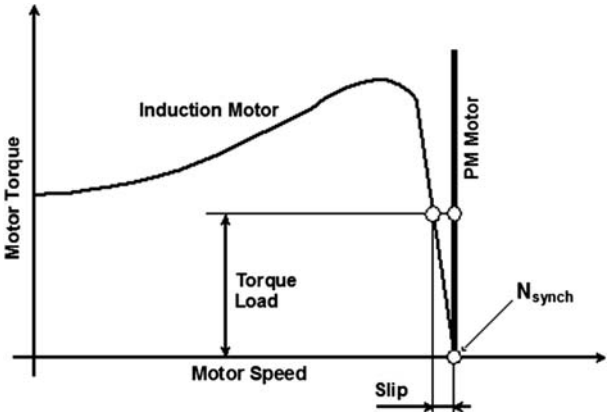


FIGURE 3.32 Torque—speed performance of induction and permanent magnet (PM) motors.

Φ = magnetic flux developed by the PMs

I = motor current.

The other basic formula permits the calculation of the **voltage** (back-EMF) induced in the stator coils by the rotating PMs in function of rotor's angular **speed**; it is written in a schematic form:

$$U = k_2 \Phi N \quad (3.22)$$

where

U = induced voltage in the stator

k_2 = conversion constant

Φ = magnetic flux developed by the PMs

N = motor rotational speed.

Because in a given motor the magnetic **flux** developed by the PMs is **constant**, two basic conclusions can be derived from the above formulas:

1. the **torque** developed by PMMs varies directly **proportional** to the **current** taken by the motor,
2. the rotational **speed** of the motor changes linearly with the **voltage** applied to the motor.

These are very essential observations that govern the operation and analysis of PMMs and will be used in the following.

Typical performance curves for PMMs at full loading are presented in Fig. 3.33 according to a major manufacturer's catalog data [40]. The four-pole motor's nameplate parameters are 60 HP power at 3,000 RPM. As

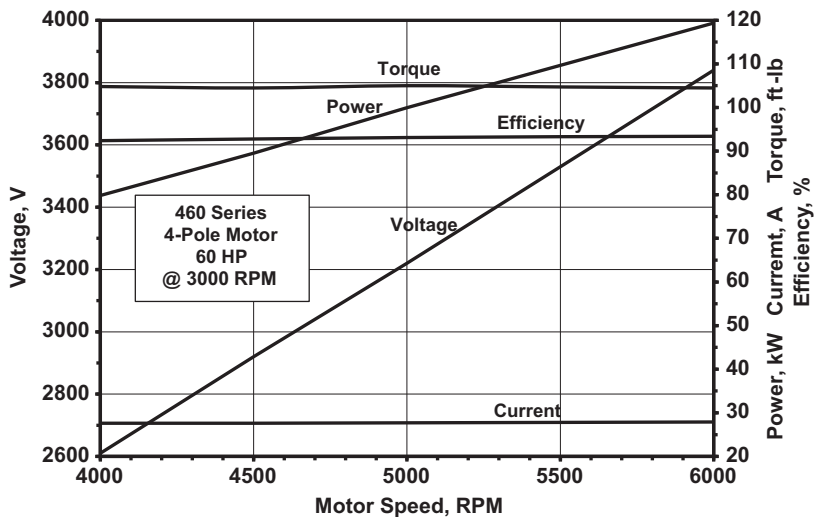


FIGURE 3.33 Performance curves of a four-pole permanent magnet motor.

seen, motor speed changes linearly with the applied voltage; this is the basis of speed control applied to PMMs. The relationship between motor current and torque can also be observed: torque depends on motor current. The PMM, just like the IM, provides a constant torque at variable speeds, and 'this is exactly needed for driving a submersible centrifugal pump. Efficiency is quite constant over the operational speed range of the motor and it is greater than 90%.

Motor performance at a constant speed and partial load conditions is presented in Fig. 3.34, typical for a major manufacturer's PMMs [41]. The most striking difference from the behavior of IMs is the variation of motor **current** with the load on the motor. PMMs have a linear current versus load curve and take considerably less amperage at low loading conditions than IMs do. This is explained by the fact that the rotor of a PMM does not need outside energy to maintain the magnetic flux. The IM, on the other hand, takes relatively high currents to keep up the rotor's magnetic field even at no-load conditions. For comparison, Fig. 3.34 shows also the current for an IM in dashed line, as taken from Fig. 3.24. Other significant differences [42] are the variations of efficiency and power factor versus motor load: these, in contrary to IM behavior, do not change significantly with the load on the motor.

Appendix E contains several tables with basic information on selected PMMs offered by two major providers including operational parameters as well as mechanical data.

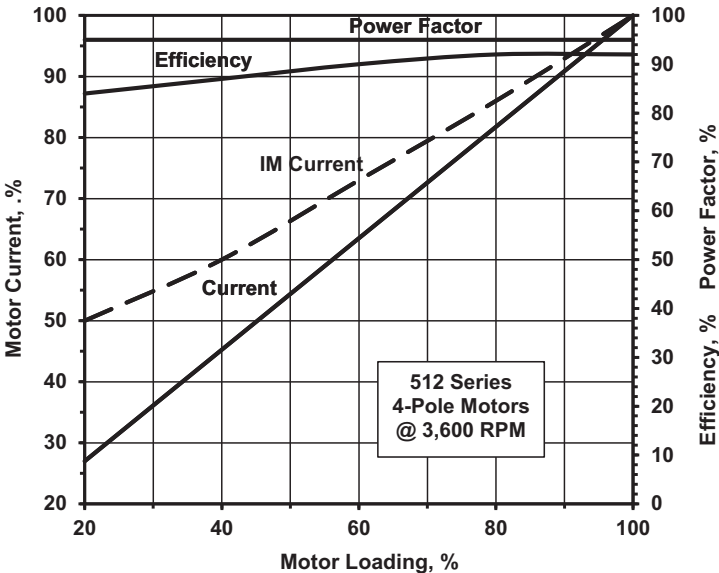


FIGURE 3.34 Performance curves of partially loaded four-pole permanent magnet motors. *IM*, induction motor.

TABLE 3.2 Overview of Available PM Motors From Novomet Company [40]

Motor Series	Motor OD (in)	Speed Range (RPM)	Power Range (HP)
Motors Rated at 3,600 RPM			
406	4.06	1,000–4,200	10–225
460	4.60	1,000–4,200	35–360
512	5.12	1,000–4,200	51–480
744	7.44	1,000–4,200	96–1,290
Motors Rated at 6,000 RPM			
319	3.19	1,000–6,000	19–120
406	4.06	1,000–6,000	19–450
460	4.60	1,000–6,000	54–540
512	5.12	1,000–6,000	80–620
<i>OD</i> , outside diameter; <i>PM</i> , permanent magnet.			

Main parameters of PMMs offered by two major manufacturers are presented in [Tables 3.2 and 3.3](#); they contain motor dimensions and the power ranges of available motors [40,41].

3.3.2.5 Ultrahigh Speed Permanent Magnet Motor Applications

The application of PMMs with extra high speeds (up to 12,000 RPM) in ESP systems has definite advantages but necessitates the use of special equipment. More details are given in the following, based on the data from a major manufacturer [43].

The operating speed range of ultrahigh speed (UHS) equipment is between 1,000 and 12,000 RPM, although their special pumps can operate at speeds up to 15,000 RPM. Because of this extremely high operating speed the whole ESP

TABLE 3.3 Overview of Available PM Motors From Borets Company [41]

Motor Series	Motor OD (in)	Maximum Motor Power (HP)	
		50 Hz	60 Hz
456	4.56	400	150
512	5.12	760	350
<i>OD</i> , outside diameter; <i>PM</i> , permanent magnet.			

system is significantly **shorter**, less than half to one-third of the length of conventional units. The compact design of the ESP system makes it possible to **assemble** and test the unit at the manufacturer's **facility**; it is delivered to the well fully assembled and is ready to run in the well. Before installation, the MLE must be attached to the motor only; this feature considerably decreases installation time and labor costs as well as minimizes rig time and any human errors. Typical installations require one-third to one-fourth of the installation time of a traditional ESP system. The compact design of the UHS ESP system permits deeper pump setting depths and running through high dogleg severity sections.

The UHS submersible pump is a modular multistage centrifugal pump designed for a speed range of 1,000–12,000 RPM. The **float**er design pump is equipped with stages distributed in several pump **modules**, with seven to nine stages per module depending on pump model. Pump stages are manufactured on computer numerical controller machines from **solid** pieces of high-grade stainless steel. Radial **stability** of pump stages at UHSs is crucial, all bearings and bushings must be of abrasion-resistant quality and made from hard materials (bimetallic, tungsten carbide facing radial bearings and silicon carbide toward pump shaft). Axial loads in the floating impellers are taken up by downthrust **washers** made of a hard material and every module has its thrust bearing to carry the load from the discharge pressure. The use of hard alloys in manufacturing maximizes the ESP's abrasion and corrosion resistance. The shaft of low-rate pumps may be **hexagonal** and may have double keyways with separate keyways for each module to minimize shaft dimensions and increase pump efficiency.

Typical performance curves of an UHS submersible pump are shown in Fig. 3.35 at a nameplate speed of 10,000 RPM [43]. As seen, extremely high heads are developed by the stage because of the high speed; this usually results in a low number of required stages and a short pump.

PMMs used in UHS service of a major manufacturer [43] have six **poles** and a rated speed of 10,000 RPM at a frequency of 500 Hz. Their rotors are considerably **shorter** than those of standard IMs and thus have a much higher power density. The heat generated by the motor's operation at such high speeds required the development of an enhanced **cooling** system. Cooling is provided by an **active** system (in contrast to the passive cooling system used in IMs) that consists of

- an oil-circulating **pump** and heat **exchanger** built in the motor base
- a protector with an **increased** oil capacity.

Thanks to the much higher motor oil **volume** and the active cooling system the UHS motors' thermal behavior is better than that of the IMs. Heat transfer between motor housing and reservoir fluid depends on the same parameters (fluid rate, casing size, etc.) as in standard systems and similar fluid velocities around the motor are required for proper cooling. UHS motors are designed for

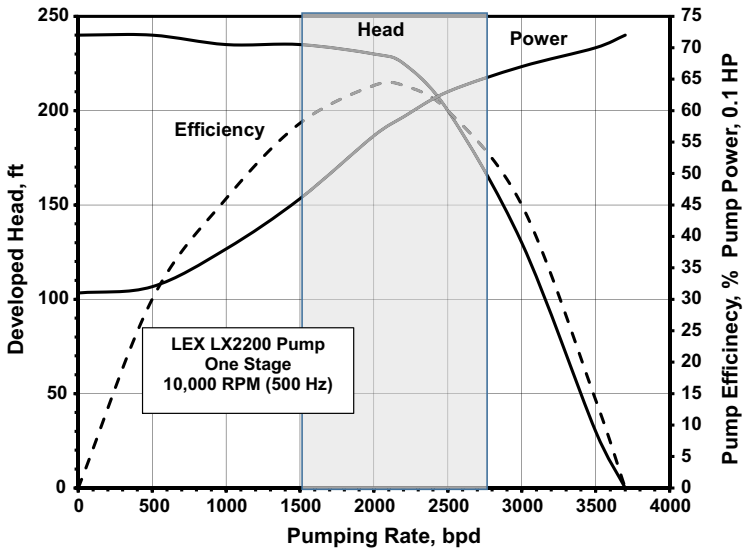


FIGURE 3.35 Performance curves of a ultrahigh speed pump.

low voltage and low current and have a flat, at least 91%, power efficiency over a wide range of motor loading.

Comparison of UHS PMM and standard IM installations has shown that considerable power **savings** can be realized with UHS equipment because of the higher motor efficiency and lower current requirement [44]. However, when UHS systems are compared to normal- or high-speed (up to 6,000 RPM) PMM installations, similar power efficiencies can be expected. In such cases the advantages of using UHS systems come from other features such as shorter pump and motor lengths and better corrosion and abrasion tolerance.

It is also important to note that UHS ESPs have significantly better performance in **low-rate** and slim-hole applications. For example, the efficiency of UHS pumps producing 100–200 bpd is greater than 50%, whereas standard-speed pumps have efficiencies typically below 40%. The higher efficiency of the UHS pumps leads to longer system run life and less problems associated with high pump temperatures, such as scale deposition, MLE overheating, and bearing problems.

3.3.2.6 Comparison with Induction Motors

Electric motors convert electric **energy** to mechanical **work** and while doing so they create irreversible energy losses that reduce the motors' power efficiency. The power losses that occur in electric motors are classified as given in the following.

- **Copper Losses.** These are the largest part of the total losses created by the heating effect of the current running in the stator and rotor windings.

- **Iron Losses.** Iron losses are generated in the conducting core laminations by the alternating magnetic field due to hysteresis and eddy currents.
- **Mechanical Losses.** Such energy losses are divided into friction and windage (FW) losses. Friction occurs in the motor’s bearings and windage losses arise at the “air gap” by the friction of the air or, in case of ESP motors, the motor oil.
- **Stray Losses.** Any additional energy losses.

If power losses in IMs and PMMs of the same power rating in ESP service are compared, the most striking difference is that PMMs have **zero** copper losses in their rotors because there are no windings in their rotors. On average, motor losses in PMMs **decrease** by about 20% because of this effect alone. Iron and FW losses are also lower in PMMs because of the **shorter** length of the PMM due to the higher magnetic flux provided by the PMs [38,45]. These effects are clearly seen in Table 3.4 in which power losses in two motors of the same power (160 kW) are compared [45]. The net result of the lower amount of energy waste in PMMs is their increased power **efficiency** that is usually above 90%.

Table 3.5 compares the main operational parameters of an IM to a standard-speed and a high-speed PMM of the same power made by a major manufacturer [40]. PMMs use less electric **current** and are more **efficient** and have much higher power factors; their constructional length and weight, at the same time, are considerably less. All these features indicate the superior performance of PMMs in ESP service.

The combination of the high power efficiency and the reduced constructional length results in a lower amount of **heat** generated inside the PMM. This condition reduces the cooling required for proper operation so lower liquid

TABLE 3.4 Power Losses in 160 kW Induction and Permanent Magnet Motors				
Loss Type	Motor Losses			
	IM (kW)	PMM (kW)	IM (%)	PMM (%)
Stator Copper Loss	8.34	7.76	44.2	78.5
Rotor Copper Loss	5.32	0.00	28.2	0.0
Iron or Core Loss	2.91	0.71	15.4	7.2
Friction and Windage	1.70	1.19	9.0	12.0
Stray Losses	0.59	0.22	3.1	2.2
Total	18.86	9.88	100.0	100.0

TABLE 3.5 Comparison of Sample Induction and PM Motors

Parameter	160 HP 3,600 RPM Induction Motor	160 HP 3,600 RPM PM Motor	160 HP 6,000 RPM PM Motor
NP Voltage, V	2,400	3,230	2,480
NP Current, A	40	27	30
Efficiency, %	84.0	90.8	91.7
Power Factor	0.86	0.95	0.95
Length, ft	25	18	10
Weight, lb	1,244	772	452

NP, nameplate; PM, permanent magnet.

velocities around the motor are necessary. Cooler motor temperatures considerably increase the run **life** of the motor because the insulating materials do not deteriorate so quickly and maintain their dielectric properties longer at lower temperatures. The following example problem illustrates the reduction of skin temperature in a PMM.

Example 3.4

Calculate the skin temperature of a PMM under the conditions given in [Example 3.3](#). For comparative purposes, assume that all input variables are the same as those of the original example except the much higher motor efficiency $\eta_m = 0.92$ and the shorter length of the motor $L_{\text{mot}} = 6.4$ ft.

Solution

First let us assume a temperature rise in the annulus equal to $\Delta T_f = 3^\circ\text{F}$. Average temperature in the annular fluid is thus

$$T_{\text{avg}} = 150 + 3/2 = 151.5^\circ\text{F}.$$

Specific heat of the light oil at this temperature is found to be $c = 0.5$ BTU/lb/°F.

Calculated temperature rise, according to [Eq. \(3.13\)](#), is

$$\Delta T_f = 9,000 (1 - 0.92) / (778 \cdot 0.5 \cdot 0.92 \cdot 0.6) = 3.4^\circ\text{F}.$$

Because this is very close to the assumed value, no iteration is needed and the fluid average temperature is found from [Eq. \(3.14\)](#):

$$T_f = 150 + 3.4 = 153.4^\circ\text{F}.$$

For calculating the heat convection in the annulus, first assume a motor skin temperature increase of $\Delta T_s = 90^\circ\text{F}$. The average temperature in the annulus then becomes

$$T_{\text{avg}} = T_f + \Delta T_s/2 = 153.4 + 45 = 198.4^\circ\text{F}.$$

The required thermodynamic parameters of the oil at this average temperature, as found from the appropriate tables or computer calculations, are listed here:

$$k = 1.9\text{E-}5 \text{ BTU}/(\text{s ft } ^\circ\text{F})$$

$$c = 0.5 \text{ BTU}/\text{lb}/^\circ\text{F}$$

$$\rho = 53.1 \text{ lb}/\text{ft}^3$$

$$\mu = 3.567\text{E-}3 \text{ lb}/(\text{ft s}).$$

Now the calculation of the convective heat transfer coefficient is possible from [Eq. \(3.17\)](#):

$$\begin{aligned} h &= 0.0378 \text{ } 1.9\text{E-}5^{(0.67)} \text{ } 1^{0.8} \text{ } 53.1^{0.8} \text{ } 0.5^{0.33} \text{ } 3.567\text{E-}3^{(-0.47)} \text{ } 4.5^{-0.2} \\ &= 5.2\text{E-}3 \text{ BTU}/(\text{s ft}^2^\circ\text{F}). \end{aligned}$$

The motor skin temperature increase is calculated from [Eq. \(3.16\)](#):

$$\Delta T_s = [2.7 \text{ } 60/(6.4 \text{ } 4.5 \text{ } 5.2\text{E-}3)](1/0.92 - 1) = 94^\circ\text{F}.$$

Because the calculated temperature increase is different from the assumed value, a new iteration step is required. The new assumed value is $\Delta T_s = 94^\circ\text{F}$. The average temperature in the annulus then becomes

$$T_{\text{avg}} = T_f + \Delta T_s/2 = 153.4 + 47 = 200^\circ\text{F}.$$

The required thermodynamic parameters at this average temperature are

$$k = 1.9\text{E-}5 \text{ BTU}/(\text{s ft } ^\circ\text{F})$$

$$c = 0.5 \text{ BTU}/\text{lb}/^\circ\text{F}$$

$$\rho = 53 \text{ lb}/\text{ft}^3$$

$$\mu = 3.41\text{E-}3 \text{ lb}/(\text{ft s}).$$

The new convective heat transfer coefficient is found from [Eq. \(3.17\)](#):

$$\begin{aligned} h &= 0.0378 \text{ } 1.9\text{E-}5^{(0.67)} \text{ } 1^{0.8} \text{ } 53^{0.8} \text{ } 0.5^{0.33} \text{ } 3.41\text{E-}3^{(-0.47)} \text{ } 4.5^{-0.2} \\ &= 5.3\text{E-}3 \text{ BTU}/(\text{s ft}^2^\circ\text{F}). \end{aligned}$$

The skin temperature increase is calculated from [Eq. \(3.16\)](#):

$$\Delta T_s = [2.7 \text{ } 60/(6.4 \text{ } 4.5 \text{ } 5.3\text{E-}3)](1/0.92 - 1) = 92^\circ\text{F}.$$

As the calculated temperature increase is different from the assumed value, a new iteration step is required. The new assumed value is $\Delta T_s = 92^\circ\text{F}$. The average temperature in the annulus then becomes

$$T_{\text{avg}} = T_f + \Delta T_s / 2 = 153.4 + 46 = 199^\circ\text{F}.$$

The required thermodynamic parameters at this average temperature are

$$k = 1.9\text{E-}5 \text{ BTU}/(\text{s ft } ^\circ\text{F})$$

$$c = 0.5 \text{ BTU}/\text{lb}/^\circ\text{F}$$

$$\rho = 53 \text{ lb}/\text{ft}^3$$

$$\mu = 3.46\text{E-}3 \text{ lb}/(\text{ft s}).$$

The new convective heat transfer coefficient is found from Eq. (3.17):

$$\begin{aligned} h &= 0.0378 \cdot 1.9\text{E-}5^{(0.67)} \cdot 1^{0.8} \cdot 53^{0.8} \cdot 0.5^{0.33} \cdot 3.46\text{E-}3^{(-0.47)} \cdot 4.5^{-0.2} \\ &= 5.3\text{E-}3 \text{ BTU}/(\text{s ft}^2\text{F}). \end{aligned}$$

The skin temperature increase is calculated from Eq. (3.16):

$$\Delta T_s = [2.7 \cdot 60 / (6.4 \cdot 4.5 \cdot 5.3\text{E-}3)](1/0.92 - 1) = 92^\circ\text{F}.$$

The calculated and the assumed values are close enough and the motor skin temperature is found from Eq. (3.18) as

$$T_s = 150 + 3.4 + 92 = 245.4^\circ\text{F}.$$

The motor skin temperature would be $T_s = 158^\circ\text{F}$ if the well produced only water.

When comparing the skin temperature calculated here with the results of Example 3.3, in which a conventional induction-type electric motor was used, an important **benefit** of using PMMs is clearly seen. The skin temperature is less by almost 50°F in the PMM, which results, according to the **Arrhenius' rule**, in a very substantial increase in the **life** of motor insulator materials. PMMs, compared to IMs, therefore, can be used without failure much longer at the same well temperature. They can also be used at higher temperatures because of their much greater power efficiency and smaller size.

3.3.2.7 Case Studies

The earliest reports on PMM operations came from Russia [46] where two companies installed altogether 48 units starting in 2005. Two years later, 32 motors were still operating and their use increased the mean time between failures (MTBF) of ESP systems in the two fields by 160 and 540 days. Interestingly, none of the ESP system failures were caused by the failure of the PMM unit. In another case [47] an average reduction of 30% on power consumption was experienced by a major Russian operating company, Gazprom Neft, after installing 268 ESP units with PMMs in 2011.

The use of UHS ESP units with PMMs to produce low-rate wells (less than 1,000 bpd) in a West Siberian oil field was introduced in 2014 [48]. The high

nameplate speed of the motors (10,000 RPM) allows for an exceptionally short ESP system with inherent advantages in running and pulling operations; they can handle high dogleg severities as well. Currently, more than 40 units operate with an average mean time between failures of more than 1,000 days. Average power savings reached 40%, as compared to the use of IMs.

A comprehensive laboratory testing of ESP motors [49] involving four IMs and four PMMs was performed between 2014 and 2015. Tests were conducted in a horizontal bench with the main task of comparing the operation of induction and PMMs. Most conclusions of the tests have substantiated the usual claims of PMM vendors: PMMs, compared to IMs of the same power rating, (1) are shorter and lighter, (2) have lower-rated currents and temperature rises, (3) have higher power efficiencies, and (4) have much higher power factors. The paper also estimated the possible run life of PMMs based on 250 cases and predicted that PMMs might last twice as long as IMs mainly because of their lower operating temperatures.

3.3.2.8 Conclusions

As discussed in the previous sections, PMMs offer many advantageous features over IMs and thus will surely replace those in the near future. This is clearly seen in Russia where about 10%–12% of the approximately 60,000 ESP installations have been already converted to PMMs [49].

The main **advantages** of PMM use in ESP operations are given in the following.

- PMMs have inherently higher power **efficiencies** than IMs because
 - all electric and magnetic losses (copper and iron losses) in “squirrel cage” rotors are eliminated,
 - the stator’s magnetic flux only penetrates the rotor to the depth of the PMs.
- The power efficiency of PMMs is quite **constant** in their entire load range; in contrast, IMs have low efficiencies at low loading conditions.
- Because of their higher power efficiency (up to 93%), PMMs use less **power**. Power savings in the range of 10%–30% can be attained that result in production cost **savings** of the same magnitude.
- The amount of **heat** generated in PMMs is very low because of the relatively high efficiency of the motor. Motor cooling is not **critical**, the required liquid velocity around the motor is lowered by about 50%, compared to IMs. This effect has several consequences on the operation of the ESP system:
 - Under the same conditions, ESP installations with PMMs can be operated at lower flow **rates** than with IMs.
 - PMMs can be used without overheating problems in **bigger** casing sizes than IMs of the same size.
 - Motor run **life** increases because insulators at lower temperatures keep their dielectric properties at design levels for longer periods.
 - Scaling problems, normally associated with IMs, are eliminated.

- The higher operating speed of PMMs (UHS motors may have nameplate speeds of 10,000 RPM) permits to **optimize** the utilization of submersible pumps:
 - Running the pump at greater than normal speeds substantially increases the **head** capacity of the pump and also reduces the required **number** of pump stages.
 - The wide variation of pumping speeds makes it possible to always use the pump at its BEP.
- PMMs are **shorter** (by about 30%) than IMs of the same power rating and can negotiate tighter well trajectories. They can be used in deviated wells with a maximum buildup rate of 24 degrees per 100 ft [48].
- Because of their compact size, PMMs weigh less and are easier to run into and retrieve from the well; installation and pulling operations require less time.
- Lower electric **currents** are used by PMMs than by IMs because of their higher power density; the results are
 - the required size of electric cables is **smaller**, giving a definite reduction in the investment cost of the ESP system,
 - startup currents are below nameplate values, and note that IMs require four to five times the normal current at system startup.
- PMMs have quite high and basically **constant** power factors (approaching 1.0), this feature can reduce the power bill.
- The no-load current taken by a PMM is about 80% less than that of an IM. This means that, in contrary to IMs, PMMs can safely be used if the well is produced in **cycles** because the motor does not overheat when the liquid rate drops.

One negative feature of PMMs related to system **safety** must be taken very seriously. This is related to operations when the motor is **inadvertently** rotated by the pump by

- liquid flowing **back** from the tubing string during system shutoff or
- liquid **movement** in the tubing string during running or pulling of the ESP unit.

Under these conditions the submersible pump is **rotated** by well fluids in the tubing string in the proper or the opposite (back spinning) direction and turns the motor's rotor firmly connected to it; the motor becomes an electric **generator**.

Spinning of the motor poses no problems in IMs because the voltage generated is low, maximum 12 V; it is caused by residual magnetism in the motor's coils. In a PMM, however, the magnetic flux is always provided by the PMs and any rotor movement induces **high** voltages in the stator windings. Such voltages reach the junction box and/or the open end of the cable on the

surface and pose **lethal** danger to the operating personnel. Proper **grounding** to the wellhead is necessary to prevent such accidents.

3.4 THE PROTECTOR OR SEAL SECTION

3.4.1 Main Functions

In small nonindustrial submersible pumps the electric motor is completely **sealed** against the produced liquid to prevent short circuits and burning of the motor after it is contaminated with well fluids. As the motor must be filled up with a high dielectric strength oil, ESP motors operating at elevated temperatures, if completely sealed, would burst their housing because of the great internal pressure developed by the **expansion** of the oil. This is why ESP motors must be kept **open** to their surroundings but at the same time must still be protected from the harmful effects of well fluids. This is provided by connecting a **protector** (a.k.a. seal) section between the motor and the centrifugal pump.

An ESP **protector** performs the following five crucial functions [50] and in so doing ensures the proper operation of the whole installation:

1. It ensures that no axial thrust load developing in the ESP's stages during operation is transmitted to the motor shaft. Thrust loads transmitted to the pump shaft are supported by the protector that contains the ESP unit's **main thrust bearing**. This thrust bearing must be capable of overcoming the net axial force acting on the pump shaft.
2. The protector **isolates** the clean dielectric oil with which the motor is originally filled up from well fluids that are usually loaded with dirt, water, and other impurities. It must ensure that no well fluid enters the motor during operation. This is a basic requirement because contamination of the clean motor oil can cause premature motor failures due to
 - the loss of **lubrication** in the structural bearings, and the consequently increased wear in bearing surfaces,
 - the decrease of the electric **insulation** strength of the motor oil causing short circuits in the motor's stator or rotor windings.
3. It allows for the **expansion** and **contraction** of the high quality oil the motor is filled up with. As the protector is connected directly to the motor, motor oil expanding due to both well temperature and the heat generated in the motor can enter the protector during normal operation. Similarly, during shutdowns, the oil contained in the motor shrinks because of the decreased motor temperature and part of it previously stored in the protector can be sucked back to the motor space.
4. By providing **communication** between well fluids and the dielectric oil contained in the motor, the protector **equalizes** the inside pressure with the surrounding pressure in the well's annulus. The inside and outside pressures being approximately equal, **leakage** of well fluids past the sealed joints and into the motor is eliminated. This feature

- allows the use of low-pressure and consequently lower-cost seals,
 - greatly increases the reliability of the ESP system.
5. It provides the mechanical **connection** between the motor and the ESP and transmits the torque developed by the motor to the pump shaft. The **couplings** on the protector's shaft ends must be capable of transmitting not only the normal operating torques but also the much greater torques occurring during system startup.

3.4.2 Basic Operation

The ESP protector's required functions detailed previously are accomplished by different component parts of the protector listed here:

- The two **couplings** attached to the protector's shaft transmit the torque developed by the motor to the pump. The top coupling, in addition to torque transfer, also transmits the axial loads coming from the pump shaft.
- The **thrust bearing** supports the axial loads developed by the ESP's action.
- **Isolation chambers** containing motor oil provide the rest of the necessary functions of the protector related to the isolation, expansion, and contraction, as well as pressure equalization, of motor oil and well fluids.
- Mechanical **shaft seals** prevent communication between the motor oil and well fluids along the protector shaft.

The main parts of a commonly used ESP protector are schematically shown in [Fig. 3.36](#). Being connected between the centrifugal pump and the submersible motor, the protector's shaft is connected at the bottom to the motor and to the pump at the top. It is filled with the same high dielectric strength oil as the motor and provides a direct communication to the motor oil, allowing for its expansion and contraction. The thrust bearing carries all the **axial loads** that may come from the centrifugal pump, transmitted by the shaft. As shown, the thrust bearing is situated at the base of the unit to ensure it will be the last part to be contaminated by well fluid. Earlier models had the bearing above the chambers.

The figure depicts a protector with two isolation chambers, a common solution to increase the reliability of ESP systems. The two chambers are of different designs to be detailed later. At this point, it is only important to note that the use of more than one isolation chamber can significantly increase the operational life of the ESP motor because it delays the contamination of the motor oil by well fluids. The minimum requirement is two chambers connected in series but protectors with more chambers and **tandem** protectors are also available.

Let us now discuss how the **expansion** of motor oil is handled by the protector. During operation, the heat generated in the ESP motor causes the expansion of the motor oil that rises from the bottom along the protector shaft up to the mechanical seal of the bottom isolation chamber. Not being able to

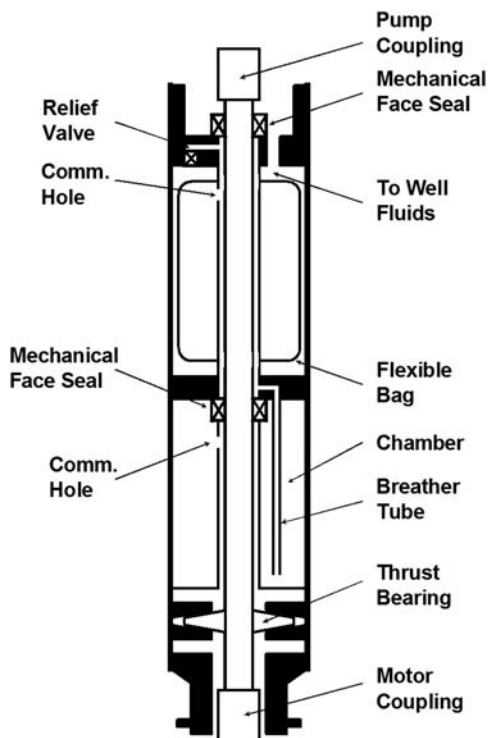


FIGURE 3.36 Schematic drawing of an ESP protector.

rise higher, the oil passes the communication hole and flows downward in the chamber space. Because the chamber was originally filled up with the same oil, part of this oil is displaced from the bottom of the chamber through the breather tube into the upper chamber, see Fig. 3.36.

Oil transferred from the bottom chamber travels along the protector shaft and before reaching the mechanical seal enters the flexible bag situated in the upper chamber. The expanding oil inflates the bag and at the same time equalizes the pressure of the motor oil and the surrounding well fluids. The relief valve prevents damage to the bag by releasing oil to the well environment as soon as the inside pressure reaches a predetermined threshold. After the ESP motor reaches its maximum operating temperature, oil pressure inside the bag stabilizes and the relief valve closes.

The **contraction** of motor oil occurs during system shutdowns when the motor cools down to ambient temperature. As the oil cools down, its volume decreases and the bag deflates. Oil from the bag travels downward along the protector shaft and through the breather tube to the bottom of the lower chamber. From here, it is sucked into the cooled-down ESP motor.

The **thermal cycle** explained previously will repeat as soon as the ESP unit is started again. As shown in this elementary description of the protector's operation, all five crucial functions required for the proper operation of the ESP system are accomplished by protectors of this basic design.

3.4.3 Main Parts

3.4.3.1 The Thrust Bearing

As already discussed, ESPs (especially those with fixed impellers) develop considerably high **axial forces** that are detrimental to the operation of the pump, the electric motor, and the whole ESP unit. Although some of these forces are absorbed in the thrust bearing installed in the pump, the ESP unit's main thrust bearing contained in the protector is responsible for the elimination of the bulk of the thrust forces.

ESP systems predominately use pad-type **plane thrust bearings** because these have the following advantages over bearings with rolling elements usually employed in other applications:

- the heat generated in the bearing during normal operation is much less,
- plane bearings can operate with lubricating oils containing solid contaminants,
- they can support higher thrust loads.

The operational principle of these bearings is the load-carrying ability of a **fluid film**. For proper operation the stationary and the moving parts are always separated by an oil film preventing metal-to-metal contact. This is one of the reasons ESP protectors are filled with the same lubricating oil as the submersible motor.

The usual construction of an ESP **thrust bearing** is schematically depicted in Fig. 3.37. The thrust **runner**, a metal disk rigidly connected to

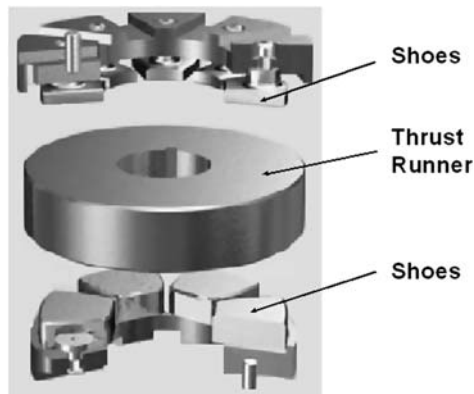


FIGURE 3.37 Construction of an ESP thrust bearing.

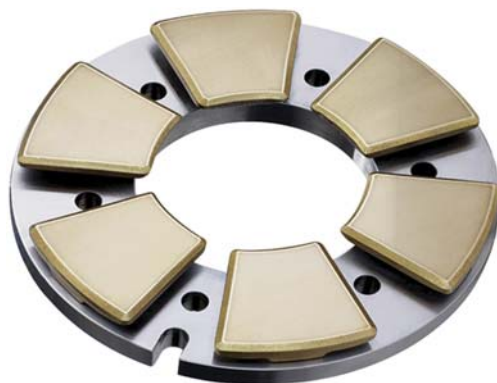


FIGURE 3.38 Construction of metal shoes in thrust bearings.

the pump shaft, turns with the speed of the motor. Segmented metal pads called **shoes** are situated above and below the runner to take up axial loads. The figure shows shoes with support pedestals in the middle of the metal pads; this feature permits shaft rotation in both directions. Because the shoes are allowed to **pivot**, the oil wedge developed between the shoes and the thrust runner prevents metal-to-metal contact of the lapped mating surfaces. As the oil film can be maintained only for a limited distance, the thrust bearing contains several (usually six) metal pads or shoes arranged circularly, see Fig. 3.38.

Shoes with tilting, **off-center** pedestals (located closer to one side of the shoe) are more efficient because they develop a wider gap with the thrust runner at the leading edge and create a more pronounced oil wedge between them and the runner. This works only if the shaft turns in the **proper direction**, and the wrong direction of rotation can damage the equipment in a short time.

Depending on the service and loading conditions, different metal alloys can be used for the bearing material. The proper material must be

- corrosion resistant
- not prone to seizure
- able to **embed** dirt particles.

Commonly used bearing materials [51] are listed along with their basic features in the following.

- Tin-based **Babbitt** is the traditional bearing material with excellent corrosion resistance and embedding capability. It cannot operate above 250°F; its fatigue strength is also low.
- **Bronze** can be used at higher temperatures up to 500°F but its embedding ability is poor and it is susceptible to galling.
- **Silver plating** on bearing surfaces meets all requirements but is costly.

- **Polymer-coated** shoes are the latest development and provide the most advanced solution for today's high-load and high-temperature applications. The polymer compound usually contains a dry lubricant such as **graphite** to ensure low friction and to facilitate the embedding of dirt particles.

Because the complete thrust bearing operates in the same oil as the motor does, a continuous **oil film** is maintained between the thrust runner and the shoes. Should this lubricating film be destroyed, a metal-to-metal contact develops with catastrophic consequences. A **failure** of the protector's thrust bearing usually means the end of the ESP unit's service life and can be attributed to one or a combination of the following factors:

- a significant reduction of the oil's **viscosity** because of elevated temperatures can almost completely eliminate the oil film,
- **misalignment** (either static or dynamic) of the thrust runner and the shoes causes a loss of oil film,
- **vibrations** transmitted to the thrust bearing by the firmly connected shafts of the ESP unit's components destroy the film between the runner and the shoes,
- **solid particles** present in the motor oil can scratch the contact surfaces and can finally damage the thrust bearing.

When properly assembled the shaft **end play** of the protector should comply to manufacturer specifications. End play is calculated from two measurements performed between the protector's top flange face and the end of the shaft. First the shaft is moved to its bottommost position when the thrust runner sits on the downthrust shoe then to its uppermost position with the thrust runner against the upper shoe. The difference between the two measurements gives the end play of the protector shaft.

3.4.3.2 Isolation Chambers

Isolation chambers provide many of the crucial features of ESP protectors and can vary from one manufacturer to the other. Their main differences are in the way well fluids and motor oil are isolated from each other. They can be classified into three broad groups: (1) the **labyrinth**, (2) the **bag** (or **bladder**), and (3) the **metal bellows** type chambers [52].

In the **labyrinth-type** units direct contact between the motor oil and the well fluids occurs. The expansion and contraction of motor oil causes the oil to follow a labyrinth path in the chamber, hence its name. The two fluids usually being **immiscible** with each other, the contamination of motor oil seldom happens. The proper operation of such chambers depends on

- the labyrinth path the motor oil has to follow
- the difference between the densities of the motor oil and the well fluid.

Because the typical specific gravities of motor oils are between 0.8 and 0.83, proper operation can only be expected for well fluids with a minimum

specific gravity of 0.9. These chambers are designed to operate in a **vertical** position and their ability to absorb the oil expanding from the motor diminishes as the angle between the vertical and the protector's shaft increases.

Bag or metal bellows type chambers, on the other hand, are closed vessels with no physical contact between motor oil and well fluids. They can be used even when the gravities of the motor oil and the well fluid are quite similar. Because their oil storage capacity does not change with the deviation from the vertical position, the use of bladder-type chambers is recommended in wells having deviations of 30 degrees or even more from the vertical.

3.4.3.2.1 Labyrinth-Type Chambers

The operation of a labyrinth-type chamber with **breather tubes** is illustrated in more detail in [Fig. 3.39](#). Before installation the protector is filled up with motor oil and well fluid is prevented from entering any of the chambers by the seals on the shaft. During the operation of the motor, expanding oil is forced upward from the **lower chamber** through one of the breather tubes into the **upper chamber**. Because the upper chamber, too, is full, oil flows downward

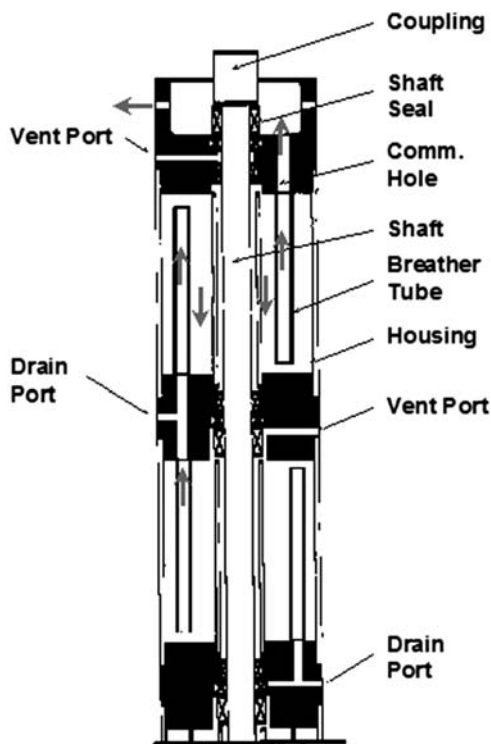


FIGURE 3.39 The operation of a chamber with breather tubes.

along the shaft then it enters the other breather tube and is directed upward. After traveling along this labyrinth path, a certain quantity of oil is displaced and is **discharged** into the well.

This situation, indicated by arrows in the figure, changes when the ESP unit is shut down. As motor oil cools down and contracts, flow directions **reverse** and well fluid enters the upper chamber where, because of its greater density, it **settles** at the bottom of the chamber. Throughout the cycles of operation and shutdown periods, the amount of well fluid may **increase** in the upper chamber but it will take a long time to completely fill the chamber. Only after the upper chamber is full, the well fluid can enter the lower chamber but it still cannot reach the motor. Thus the required dielectric properties of the motor oil can be maintained for a long time.

The **annular-type** labyrinth chamber operates similar to the unit described previously, as shown in Fig. 3.40 where flow directions during oil expansion are indicated. Oil from the lower chamber enters the annulus between the shaft and the shaft tube and then flows downward inside the middle tube. At the bottom of the chamber the flow direction changes in the annulus between the outer housing and the middle tube. Finally, part of the oil originally contained in the protector enters the well. As before, during shutdown periods well fluid is **sucked in** the upper chamber and the flow directions indicated in the figure are reversed. The end result is, as before, that the **contamination** of the motor oil is **postponed** for a sufficiently long period.

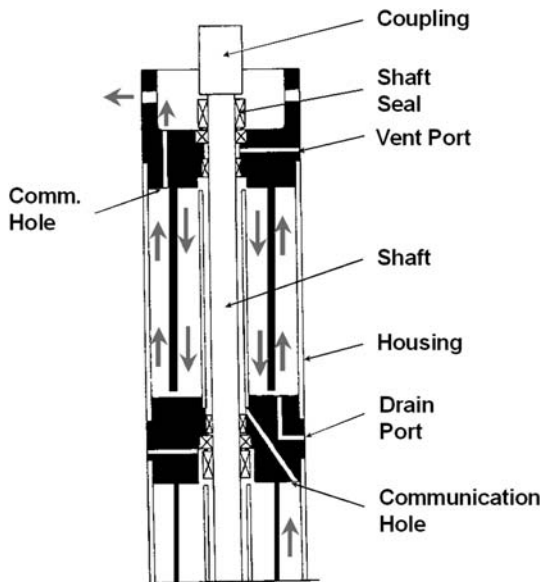


FIGURE 3.40 The operation of an annular-type chamber.

For the proper operation of the protector the isolation chamber must have the right expansion/contraction **capacity**. Chambers with breather tubes have relatively great capacities because of the small volumes of the breather tubes that discharge the motor oil up from the bottom of the chamber. Annular-type chambers, on the other hand, use a large amount of volume for passing the oil upward; hence, their expansion/contraction capacities are considerably less for the same main outside dimensions.

Deviation from the vertical of the protector's shaft at the setting depth significantly affects the **capacity** of all labyrinth-type isolation chambers. At increased well deviations the expansion/contraction capacity of the protectors rapidly decreases. This can lead to a severe reduction of the time until well fluid can get into the ESP motor and to a consequent shortening of the ESP system's operational life span. This is why labyrinth-type chambers are not used in wellbore sections with more than **30 degrees** inclination from the vertical.

3.4.3.2.2 Blocking Fluids

In any of the isolation chambers discussed so far, the reliability of separating the wellstream from the motor oil can be enhanced if a **blocking fluid** is used in the protector's chambers. The blocking fluid is a high-specific-gravity, chemically **inert** liquid placed at the contact of the motor oil (with a usual specific gravity between 0.8 and 0.83) and the well fluid as shown in [Fig. 3.41](#).

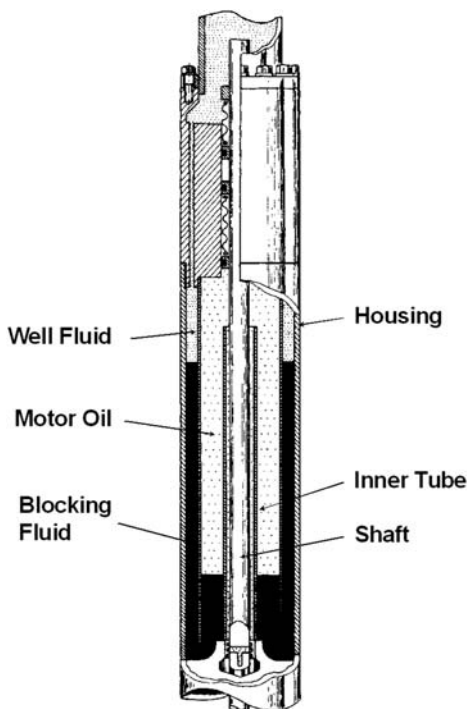


FIGURE 3.41 The use of a blocking fluid in an ESP protector.

Here an annular labyrinth-type chamber is illustrated where the blocking fluid is placed at the bottom of the outer and middle annuli. The expanding oil from the motor enters the chamber through the inner tube and reaches the top of the blocking fluid displacing it to the outer annulus. Because well fluid is situated above the blocking fluid in the outer annulus, the **contact** between the motor oil and well fluid is avoided. Thus contamination of the motor oil with well fluid possibly containing water and solid particles is effectively prevented.

The expansion/contraction **capacity** of annular-type isolation chambers with blocking fluids equals the volume of the blocking fluid contained in the chamber. Chambers with breather tubes, on the other hand, have a low capacity because the volume of blocking fluid expelled in one thermal cycle is very close to the total volume because of the small capacity of the breather tube. Breather-tube chambers, therefore, are not recommended if the use of a blocking fluid is desired.

The use of blocking fluids in isolation chambers is almost completely obsoleted by the introduction of bag-type chambers. Their use is also restricted by an increased price.

3.4.3.2.3 Bag-Type Chambers

Bag- or **bladder-type** chambers contain a positive **barrier** between the well fluid and the motor oil, see Fig. 3.42. The bladder is a **flexible bag** made of a

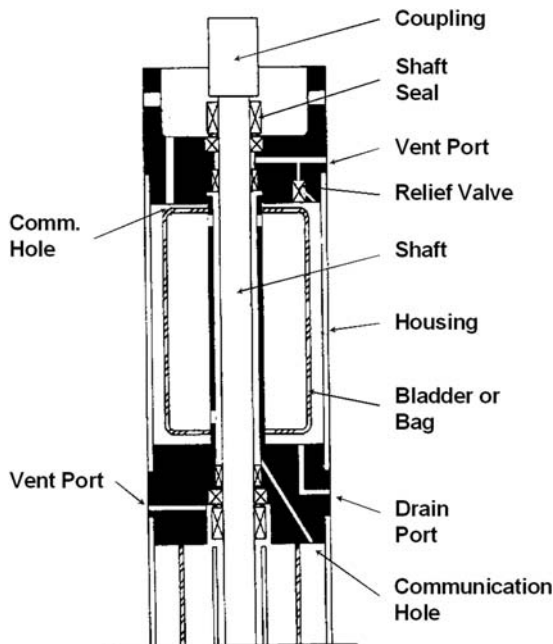


FIGURE 3.42 Construction of a bladder-type chamber.

high-performance elastomer that can withstand the harsh downhole environment. During motor oil expansion, oil from the lower chamber travels up and enters the flexible bag through holes in the shaft tube. As more and more oil leaves the motor because of the elevated temperature, the bag **expands** accordingly. After the bag is fully inflated, further inflow of motor oil increases the internal pressure causing the oil to leak upward along the shaft but is prevented to leave by the shaft seal. As the oil inside the bag communicates with the **relief valve**, this valve opens and releases oil into the environment. After the motor temperature stabilizes at its normal operating level, the relief valve closes and no more oil is released from the chamber.

When the motor is shut down and its oil contracts because of the decreased temperature, the flexible bag **collapses** because oil is drawn from it downward to the chamber below. Well fluid from above is sucked to the outside of the bag but it still cannot contaminate the oil in the motor. As seen from this operational sequence, the chamber's inside pressure is always equalized with the outside well pressure and contact between the motor oil and well fluids is prevented.

The expansion/contraction **capacity** of bag-type chambers equals the volume between the inside of the fully expanded bag and the outside of the shaft tube. Because this capacity is not affected by the inclination of the protector's center line from the vertical, the use of protectors containing flexible bags is recommended in **deviated wells**; they may also be used in **horizontal** wells.

In spite of their advantageous features, elastomer bags have their limitations, which are mainly caused by the materials used in bag manufacturing. The usual plastic materials are **permeable** to free gases, and gas bubbles may displace motor oil when the motor is hot. The bag may become **hard** and can **blister** and **fail** if contacted by some chemicals. All these problems are eliminated if exotic materials such as “**Viton**” or “**Aflas**” are used to manufacture bags.

3.4.3.2.4 Metal Bellows Type Chambers

Although isolation chambers with elastomeric bags highly improve the operation of protectors in challenging environments, their reliability and life expectancy is greatly reduced by (1) elevated **temperatures**, (2) **aggressive** gases, and (3) **corrosive** fluids. All these problems are solved if a metal bellows (usually made of **Inconel** material) is used instead of the elastomer bag to contain and isolate motor oil from the wellstream [53].

The main advantages of metal bellows in ESP protectors are

- the protector can be used at very high temperatures, usually above 400°F,
- migration of gases into the motor oil, often occurring through elastomeric bags, is prevented,
- metal bellows are resistant to corrosive gases such as H₂S and other corrosive materials.

Owing to the advantageous features just described, protectors with metal bellows are used under severe environmental conditions such as in gassy, corrosive, and high-temperature wells.

3.4.3.3 Shaft Seals

Each isolation chamber of an ESP protector is equipped with a rotating **mechanical seal**. Usually located at the top of the chambers, these seals prevent communication of well fluids and motor oil along the protector shaft. Because of this vital function, they can be considered as the **heart** of the protector. The mechanical seal illustrated in Fig. 3.43 contains a stationary seal ring fixed to the protector's housing and a rotating ring turning with the shaft's movement, both made of metal. These parts provide the **dynamic seal** function and prevent the migration of any fluids along the rotating shaft.

The main sealing action is provided between the stationary and the rotating metal rings, usually made of silicone carbide material, where a **wet seal** is present on the mating faces. Across the contact of the two faces, motor oil from the inside and well fluids from the outside give the necessary lubrication. The two faces must be extremely **flat** and must rotate **concentrically** to each other; otherwise, mixing of motor oil and well fluids causes a pumping action across the seal face. Surface defects on mating surfaces initiate leaking; hence, in wells producing sand or scale the use of tungsten carbide or ceramics on seal faces is preferred.

In addition to the dynamic or rotating seal, the shaft seal in Fig. 3.43 contains three supplementary **static seals**. These use rubber or other resilient

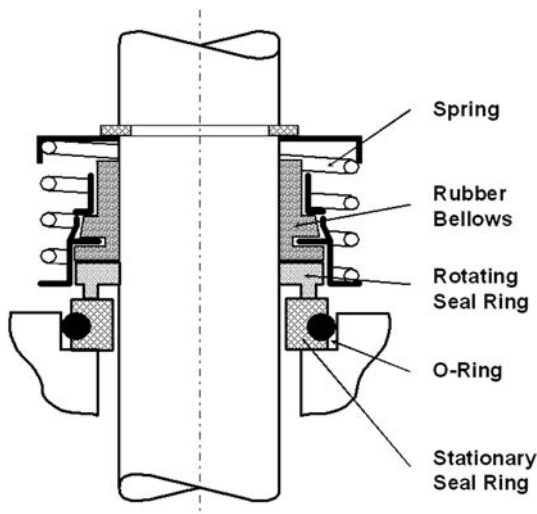


FIGURE 3.43 Construction of a mechanical seal on the protector shaft.

materials compressed against hard surfaces to provide the necessary sealing action between parts stationary in relation to each other:

1. Sealing between the protector housing and the stationary seal ring is provided by an **O-ring**.
2. An elastomeric **bellows** (usually made of rubber) radially fixed to the shaft seals on the shaft.
3. The same bellows, forced downward by a spring, provides the seal on the rotating seal ring.

Mechanical seals of other constructions (using a metal bellows instead of an elastomeric bellows, lip seals) are also available and are used in protectors from different manufacturers.

3.4.4 Other Features

The most important operational features of protectors used in the selection of the proper protector for a particular application are

- thrust bearing capacity
- shaft torque capacity
- oil expansion capacity.

The **main thrust bearing** of the ESP unit, contained in the protector, must be capable of withstanding the maximum anticipated axial force coming from the pump. As the viscosity of motor oil is lower at elevated temperatures, the oil film between the thrust runner and the thrust shoes becomes thinner at higher operating temperatures. This is why, as seen in Fig. 3.44 [54], the load rating of a

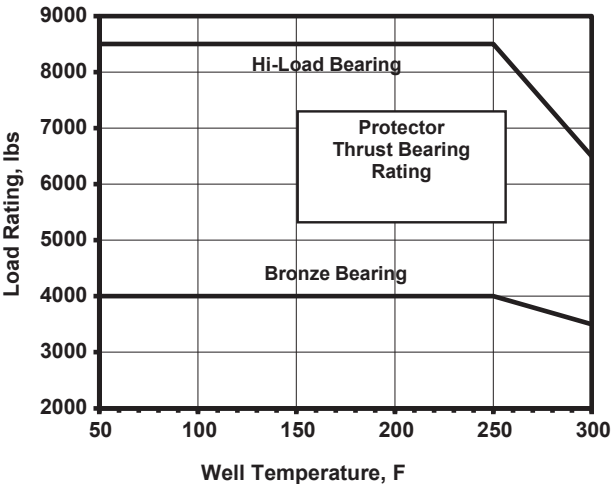


FIGURE 3.44 Load rating of a protector as a function of well temperature.

given protector must be **decreased** at higher temperatures. The capacity of thrust bearing increases with the diameter of the protector because of the increased load-bearing surfaces; bigger series units have higher load capacities.

The torque capacity of the protector shaft is checked against the maximum possible torque that may occur during startup operations. Finally, the required expansion capacity is found from the combined oil volume contained in the motor plus the protector and the maximum differential temperature. The temperature range of the operational cycles starts at installation temperature and reaches the maximum anticipated temperature during normal continuous operation.

Protectors are manufactured in different **sizes**, conforming to the sizes of other ESP components. The availability of **modular protectors** allows the operator to select the right combination for the well. These protectors can combine in a single unit bladder and labyrinth sections in different configurations. The isolation chambers are connected in series and up to three can be used with a single thrust bearing at the bottom. This solution provides the necessary **redundancy** in the ESP system for extending the operational life of the installation.

Tandem protectors are composed of two individual protectors connected between the pump and the motor. As both have their main thrust bearings, one would think that they could work in unison and their thrust ratings would be additive. In the real world, however, it is practically **impossible** to equally distribute the thrust load between the two bearings. To reach proper cooperation between the bearings requires the clearance between the moving parts of each thrust bearing must be controlled to very close tolerances. The ideal tolerance (less than 1/1,000th of an inch) can never be achieved because of the many factors (thermal expansion of metal parts, shaft compression, etc.) involved.

In practice, two solutions are available to operate tandem protectors, as discussed in the following.

- Thrust is carried in the thrust bearing of the **upper tandem** alone if there is a sufficient **axial gap** between the shafts of the two protectors. The isolation chambers of the upper tandem protect the thrust bearing as long as they get contaminated by well fluids. At this point an increased wear of the thrust bearing starts and leads to a downward movement of the protector shaft. This, in turn, allows the mechanical damage of pump stages due to a downthrust condition. Axial load will be transferred to the lower tandem only after the pump fails, but contamination of the motor is still prevented by the chambers of the lower tandem. Thus the motor survives but the pump and the upper protector are sacrificed. This solution is justified if expensive motors, as compared to service costs, are used.
- If, during installation, the shaft of the lower tandem contacts and slightly lifts the upper tandem's shaft, axial loads are carried entirely on the **lower** thrust bearing. This ensures that all chambers in the two protectors will

protect the main thrust bearing and the motor. In the normal course of operations, contamination of each chamber with well fluids starts from the top and proceeds downward. First the upper thrust bearing is damaged, then the lower one, followed by the pump. By now, well fluid can get into the motor damaging it, too. Finally, both protectors, along with the motor and the pump, are damaged but the time to system failure is greater than that in the previous case.

3.5 THE GRAVITATIONAL GAS SEPARATOR

It follows from the operational principle of centrifugal pumps that free gas entering the pump suction **deteriorates** the pump's performance. This is caused by the great difference between the **specific gravities** of liquids and gases. The centrifugal pump, being a **dynamic device**, imparts a high rotational velocity on the fluid entering its impeller but the amount of kinetic energy passed on to the fluid greatly depends on the given fluid's density. Liquid, being denser than gas, receives a great amount of **kinetic energy** that, after conversion in the pump stage, increases the pressure. Gas, on the other hand, although being subjected to the same high rotational speed, cannot produce the same amount of pressure increase. Hence, centrifugal pumps should always be fed by **gas-free**, single-phase liquid for ensuring reliable operation.

The existence of free gas at pump suction conditions (pressure and temperature) affects the operation of the ESP in several ways.

- The **head** developed by the pump decreases as compared to the performance curve measured with water.
- The **output** of a pump producing gassy fluids fluctuates; **cavitation** can also occur at higher flow rates causing mechanical damage of the pump stages.
- In cases with extremely high gas production rates, **gas locking** may occur when no pumping action is done by the pump completely filled with gas.

Because of its many detrimental effects, the production of a considerable amount of free gas requires special treatment from the ESP operator. The possible solutions and the necessary equipment will be detailed in Chapter 4 in which unconventional ESP installations are discussed. Here the simplest solution for producing wells with low gas production rates is detailed: the use of the **reverse-flow** gas separator.

The simplest and least efficient gas separator has been in use almost since the early days of ESP operations. It may also act as an **intake** for the centrifugal pump and can separate **low to moderate** amounts of gas with limited efficiency. The separator is connected between the protector and the pump and directs the separated gas into the well's casing-tubing annulus. It is, therefore, essential that the conventional installation, as given in [Fig. 3.1](#), be used where the casing annulus is not sealed off by a packer.

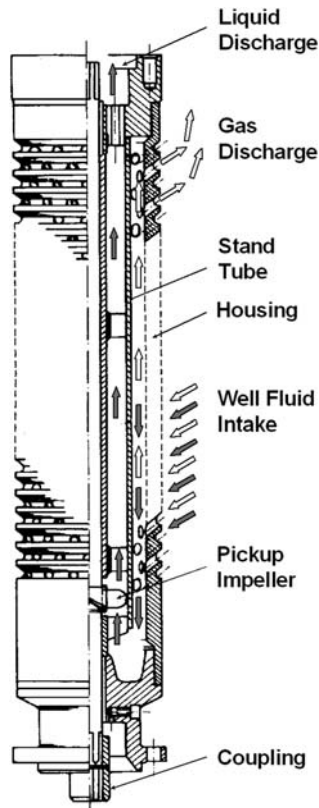


FIGURE 3.45 Construction of a reverse-flow gas separator.

The construction and operation of this type of gas separator, often called a **gravitational** (or **reverse-flow**) **gas separator**, is illustrated in Fig. 3.45. It works on the principle of **gravitational separation** by forcing the fluid flow to change direction and allowing free gas to escape into the well's annulus. Well fluid containing free-gas bubbles enters the separator through the perforated housing. In the annular space formed by the housing and the stand tube, gas bubbles **rise** but liquid flows downward. If **bubble rise velocity** is greater than the countercurrent liquid flow velocity, gas bubbles rise to the top of the separator and **escape** into the well's annulus through the upper perforations of the separator's housing. Liquid containing a reduced amount of free gas is sucked in by the pickup impeller at the bottom of the separator and is transferred to the ESP connected to the top.

The simplest gas separator described here is still in use in wells with low to moderate liquid and gas rates where the low separation efficiency achieved by this construction is sufficient. The use of the reverse-flow gas separator is inherently limited to **low liquid rates** because its operation is based on the

existence of low fluid velocities inside the housing—stand tube annulus. At higher rates normally encountered in ESP service, the liquid velocity in that annulus is too high to facilitate the separation of free-gas bubbles. Therefore, in wells with substantial amounts of free-gas production, more advanced separators with higher efficiencies have to be used, as discussed in Chapter 4.

3.6 THE ESP CABLE

Electric power from the surface is transmitted to the ESP motor through a special three-phase electric **power cable** leading from the surface to the motor connection. ESP cables work under extremely harsh conditions and must meet the following important requirements:

- **small diameter**, so that they can fit in the annulus along the well tubing,
- they must maintain their **dielectric properties** under harsh well conditions such as
 - high temperatures
 - aggressive fluid environments
 - the presence of hydrocarbon and/or other gases,
- they must be well **protected** against mechanical damage occurring during running and pulling as well as normal operations.

The operating conditions of ESP cable are very **unforgiving** and the proper choice of its type and size in many cases has a direct impact on the life of the ESP installation. A properly selected and designed cable system, on the other hand, can be expected to stay operational for many years, provided all handling and other recommendations are strictly followed.

3.6.1 Cable Materials

ESP power cables are complex structures with the following structural parts:

- the three metal (usually copper) **conductors** carrying the AC,
- the individual **insulation** of each conductor preventing short circuits and leakage currents between the conductors,
- supplementary **coverings** over the insulation providing additional strength and protection to cable components,
- the **jacket**, the protective cover of the three conductors that provides structural strength and mechanical protection to the cable and prevents contact of the insulations with the downhole environment,
- a metal **armor** (optional) providing enhanced mechanical protection during running and pulling operations as well as reducing the swelling (due to contact with well fluids) of underlying insulator materials.

In the following, the usual materials of each cable component will be described along with their main features.

Conductors in ESP cables can be made of **copper** or **aluminum** and are available in **solid** circular, **stranded**, or **compacted** wire configurations. Solid wires are the cheapest and smallest in diameter, stranded wires are more flexible, and compacted ones have about 10% smaller diameter than stranded round wires.

Aluminum is less expensive but it provides only 61% of the conductivity of an equivalent copper wire and is more difficult to splice. Copper, on the other hand, is very sensitive to H_2S corrosion but this problem can be solved with a continuous lead sheath on the insulation.

To meet the highest requirements, solid copper conductors are recommended because they are smaller than stranded ones and prevent gases or fluids from migrating up the cable. An added advantage is that they permit more reliable splicing operations.

Insulator materials used in the manufacture of individual insulations of the conductors and the cable jacket must

- withstand the high operating **temperatures** in the well,
- be oil resistant,
- resist the **migration** of free gas into the cable body.

One of the most important operational limitations of various plastic materials used for insulation in ESP cables is **temperature**; cable life decreases exponentially as temperature increases. High cable temperature over long periods causes **embrittlement** of insulation materials and leads to loss of mechanical and dielectric properties and eventual failures. As a rule of thumb, the life of insulation materials is reduced by 50% for every 18°F (10°C) of operating temperature increase (**Arrhenius' rule**).

In addition to this, cables must be resistant to the harmful effects of crude oil and hydrocarbons and must not **soften** or **swell** when in contact with well fluids. An even tougher requirement is the resistance against **penetration** by natural and other gases under pressure. If the material is **permeable** then gases can easily enter the body of the cable sections situated deeper in the well, permitting the upward migration of gas molecules in the cable's insulation. As ambient pressure around the cable decreases at lower well depths, gas bubbles expand and can rupture the insulation causing **decompression** damage. The small **cracks** thus formed not only decrease the mechanical strength of the cable but also allow aggressive and electrically conductive well fluids to reach the metal wires causing **short circuits** and an eventual failure of the whole cable. Prevention of these conditions requires sophisticated choice of materials. Most manufacturers use proprietary plastics; for a detailed treatment of available materials and techniques see **API RP 11S5 [55]**.

Frequently used **[56,57]** insulating materials include the following.

- **Polypropylene** (a thermoplastic material) can be used up to a temperature of around 200°F and is generally applied as conductor insulation because of its low cost. It is susceptible to **softening** by crude oils and other

chemicals, as well as to stress cracking caused by CO₂ and hydrocarbon gases. Contact with copper accelerates its aging; therefore, copper conductors have to be coated with tin.

- **Ethylene propylene diene monomer (EPDM)**, a thermosetting plastic material, can be used at operating temperatures well above 400°F. It has a good chemical resistance but must be prevented from decompression. It is mostly used for conductor insulation as well as for manufacturing cable jackets.
- **Nitrile rubbers** are also used for making cable jackets because they are oil resistant.

Coverings are supplementary layers of special materials to increase the protection and enhance the performance of the cable. They are applied over cable insulations and may take the following forms.

- **Braids** are woven layers of synthetic materials, usually nylon, and increase the strength of the insulation against the harmful effects of gases that migrate into the insulation material.
- **Barrier tapes** are helically wrapped on the insulation with an overlap between layers and prevent the entrance of well fluids into the cable. Their use results in increased cable life.
- **Barrier extrusions** provide seamless covering of the insulation without any gaps found with conventional braids or tapes. The different, usually thermoplastic, materials provide chemical and mechanical protection and may improve the cable's dielectric strength.
- **Lead sheath** provides a continuous seamless covering and the **ultimate** protection against gas penetration as well as the highest operating temperature up to 550°F. In environments containing H₂S the use of copper conductors necessitates that insulations be lead-sheathed.

Jackets keep the three conductors together and protect the insulations from mechanical damage during running and pulling operations, and their properties may determine the application range of the cable, especially the cable temperature rating. Usual jacket materials are **nitrile rubber** and **EPDM**, with temperature limits of 280 and 400°F, respectively. Nitrile rubber has excellent resistance to oil but swells when in contact with water. EPDM, on the other hand, can be used at higher temperatures but swells if contacted by crude oil.

Metal armor is the outer covering providing mechanical protection to the ESP cable during handling. In addition to this function, the armor also constrains the **swelling** or expansion of the insulating materials when they are exposed to well fluids. Armor is available in different materials (galvanized or stainless steel, or Monel) for different corrosive environments and is manufactured in different configurations:

- **flat** armor ensures the smallest OD,
- **interlocked** armor minimizes unraveling,
- helically applied round wires provide the greatest longitudinal strength.

3.6.2 Cable Construction

ESP cables are available in two configurations: **round** and **flat**. Round cables are generally used along the tubing string where annular space is not too critical and the cable can fit between the tubing coupling OD and the casing's drift diameter. On the other hand, flat cables with a much smaller radial space requirement are necessary for small annular configurations or along the ESP unit whose OD is considerably greater than that of the tubing string.

The construction of a typical **round** and a typical **flat** cable is shown in Figs. 3.46 and 3.47, respectively. Both cables shown have an optional metal **armor** on the outside that prevents mechanical damage during running and pulling operations. Their **jacket**, in addition to mechanically fixing the three insulated conductors, provides a **barrier** against crude oil and natural gas. Fig. 3.47 shows a flat cable construction with lead encapsulation for use in H₂S environments.

Cable **size** is defined by the size of the individual conductors used in the cable. It is specified by utilizing the **American Wire Gauge** (AWG) numbering system commonly used in the wire industry. A greater AWG number designates a smaller wire; the usual sizes manufactured in the ESP industry are No. 1, 2, 4, and 6.

Many times, it is necessary to increase the length of the ESP cable when two individual cables have to be joined by properly executed **splices**. Splicing is required also for **repairing** damaged cable, for **joining** the MLE to the main cable, etc. Most splices are made by wrapping tape around the connected conductors but **molded** or **vulcanized** splices are also known. Integrity of cable splices depends on the materials used and the skill of the person making the splice. In most cases a **splice-free** cable is preferred for extra reliability. For proper splicing instructions, refer to the manufacturers' literature [54,58].

3.6.3 Operational Features

The main properties of cable materials (conductors, insulators, etc.) as well as their testing procedures are presented in the API publication **RP 11S6** [59].

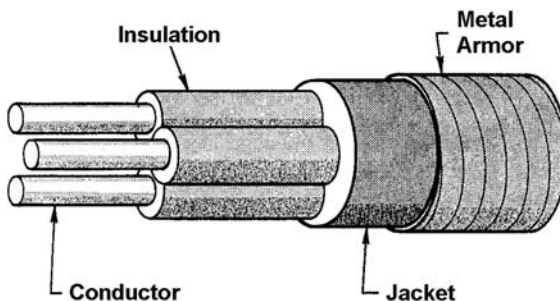


FIGURE 3.46 Construction of an ESP round cable.

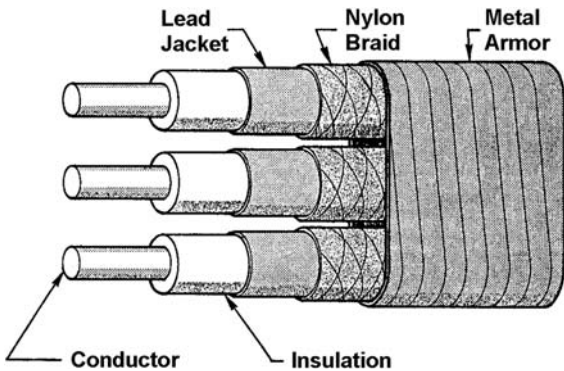


FIGURE 3.47 Construction of an ESP flat cable.

TABLE 3.6 Main Parameters of a Round ESP Cable [60].						
AWG No.	Conductor Type	Conductor Diameter (in)	Overall Diameter (in)	Resistance at 400°F (Ω/1,000 ft)	Reactance at 60 Hz (Ω/1,000 ft)	Cable Weight (lbs/1,000 ft)
1	Stranded	0.331	1.44	0.226	0.034	1,825
1	Solid	0.289	1.35	0.220	0.033	1,700
2	Stranded	0.292	1.35	0.294	0.035	1,575
2	Solid	0.258	1.28	0.286	0.035	1,480
4	Solid	0.204	1.16	0.441	0.037	1,135
6	Solid	0.162	1.07	0.700	0.040	910

Technical parameters of one type of round cable rated at 400°F service with copper conductors and a 0.075"-thick EPDM insulation from a leading manufacturer [60] are given in Table 3.6.

As seen from the table, the inductive reactance of ESP cables is generally low; this is why in engineering calculations, only the resistive component of their impedance is considered. For example, let us take 1,000 ft of an AWG 4 cable and calculate the **impedance** of the cable according to Eq. (2.28):

$Z = [0.441^2 + 0.037^2]^{0.5} = 0.442$ ohms, only slightly more than its resistance of 0.441 ohms.

The operating conditions of the ESP cable are extremely affected by its **temperature**. As the cable is suspended in the well's annulus in close contact

with the tubing string, its steady-state temperature is the result of the following factors:

- ambient (**fluid**) temperature valid in the tubing string,
- the amount of heat **generated** in the cable due to the operating electric current,
- the amount of heat **dissipated** in the stagnant fluid (liquid or gas) present in the annulus.

Because of these effects, cable temperature changes along the tubing string in such a way that it **parallels** the tubing temperature but is always greater than that. The difference depends on the **balance** of the generated and dissipated heats. Heat is dissipated in the annulus by **natural convection**, the effectiveness of which greatly depends on the type of fluid filling the annular volume. This is why cable differential temperature is different **below** and above the dynamic liquid level in the annulus.

As natural convection is more efficient in liquids than in gases, the cable is cooled better **below** the dynamic liquid level than in the upper well sections where the annulus contains gas only. The depth of the annular liquid level, therefore, greatly affects the maximum temperature of the cable. With high liquid levels, maximum temperature occurs at the setting depth of the motor. On the other hand, if the liquid level is close to the depth of the motor, this is where maximum cable temperature may occur.

The electric **impedance** of the cable increases with temperature and the following formula is used to find the resistance (because inductive reactance is negligible) at elevated temperatures (see also Eq. 2.36):

$$R_T = \frac{L_c r}{1,000} [1 + 0.00214(T - 77)] \quad (3.23)$$

where

R_T = cable impedance at temperature T , ohms

L_c = cable length, ft

r = resistance of conductor at 77°F, ohm/1,000 ft

T = cable temperature, °F.

The **voltage drop** in cables is a very important parameter and defines the inevitable **energy losses** occurring in the cable because voltage drop multiplied by the current gives the electric energy wasted in the ESP cable. Voltage drop changes with the material, the size, and the temperature of the conductor but the main parameter is the magnitude of the current flowing in the cable. The following formula is used to find the three-phase voltage drop across the cable:

$$\Delta U_{\text{cable}} = \sqrt{3} R_T I = 1.732 R_T I \quad (3.24)$$

where

ΔU_{cable} = voltage drop in cable, V

R_T = cable impedance at temperature T , ohms

I = motor current, A.

TABLE 3.7 Typical Voltage Drops in Copper Conductors	
AWG	Voltage Drop per 1 A (V/1,000 ft at 20°C)
1	0.26
2	0.33
4	0.53
6	0.84
8	1.32

After Brown KE. The technology of artificial lift methods, vol. 2b. Tulsa, Oklahoma: PennWell Books; 1980.

Table 3.7 contains typical values of voltage drops in copper conductors for each 1,000 ft of cable length, assuming a current of 1 A [61].

Fig. E.1 in **Appendix E**, as given by most cable manufacturers, can be used to calculate the voltage drop in usual ESP cables. Because temperature has a direct effect on the electric resistance of metal wires, the values read from the figure must be corrected for elevated temperatures by using the correction factors given. The correction factors can also be calculated from Eq. (3.23) introduced earlier.

The three-phase **power loss** (in kilowatts) in the cable is calculated from the cable resistance and the motor current as follows:

$$\Delta P_c = \frac{3 I^2 R_T}{1,000} \tag{3.25}$$

where

- ΔP_c = power loss in cable, kW
- R_T = cable impedance at temperature T , ohms
- I = motor current, A.

Example 3.5

Calculate the voltage drop and the energy wasted in an **AWG 4** size 4,500-ft-long cable, if the average cable temperature is 350°F, the resistance of the conductor at 77°F is 0.271 Ω/1,000 ft, and motor current is 30 A.

Solution

The total resistance of the cable at the given temperature is found from Eq. (3.23) as

$$R_T = 4.5 \, 0.271 [1 + 0.00214(350 - 77)] = 1.22 \, 1.58 = 2.5 \, \Omega.$$

Voltage drop across the cable from Eq. (3.24) is

$$\Delta U_{\text{cable}} = 1.732 \cdot 2.5 \cdot 30 = 130 \text{ V.}$$

Finally, power loss is calculated from Eq. (3.25) as

$$\Delta P_c = 3 \cdot 30^2 \cdot 2.5 / 1,000 = 6.8 \text{ kW.}$$

When analyzing the operating conditions of the ESP cable, one has to consider that cable temperature is a function of the well's ambient temperature and the **heating effect** of the AC flowing through it. Hence, at higher well temperatures, in order not to exceed the allowed maximum cable temperature, the current level in the cable must be restricted. The accepted solution is the construction of so-called **ampacity charts** (from “ampere capacity”) for available cables, which specify the maximum allowed cable current as a function of maximum well temperature. Fig. 3.48 presents an example for a given cable type of a major manufacturer [62] and shows the adverse effect of well temperature on the allowed amperage in the ESP cable. Although several years ago the use of ESP equipment was severely limited by higher temperatures, today's most advanced cables can be used at well temperatures above 450°F.

All cables, new, rerun, or spliced must be periodically checked in the field according to the recommendations of the API, set forth in **RP 11S6** [59].

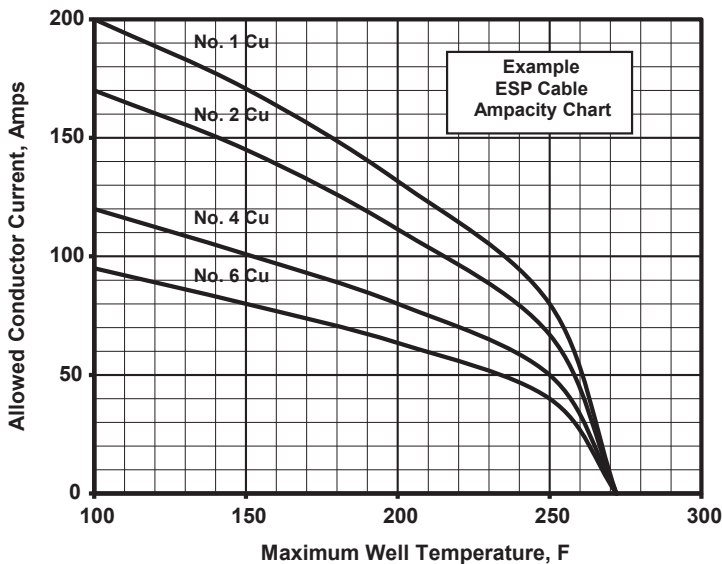


FIGURE 3.48 Typical ampacity chart for an ESP cable [62].

3.7 MISCELLANEOUS DOWNHOLE EQUIPMENT

In addition to the equipment detailed so far, the proper operation of an ESP installation requires several additional pieces of equipment to be run into the well.

The **MLE** is the part of the ESP cable running along the pump, the gas separator, and the protector and is directly connected to the motor. As radial distance along the ESP unit in the annulus is severely restricted, typically a piece of a flat cable is used. The top end of the MLE is spliced to the main cable while its bottom end is connected to the motor terminals by (1) direct splicing or (2) using a connector (see [Fig. 3.19](#)) that fits into the motor's pothead.

The MLE operates under extremely **adverse** conditions because of the restricted size, the high mechanical stresses, and the high temperatures involved. As the heat load is at a maximum at the head of the motor, this is the point where cable temperature is the greatest. Because of these effects, MLEs are usually **replaced** every time a cable is reused.

Cable bands are used to strap the power cable to the tubing string and prevent mechanical **damage** to the cable during running and pulling as well as during normal operation. The recommended distance between cable bands is 15 ft.

Flat cable guards should be used along the ESP equipment to protect the flat cable from mechanical damage by firmly **fixing** it to the outside of the ESP components. They come in lengths of 18 in; use as many guards as necessary to cover the total length of the flat cable.

A **check valve** is installed about two to three joints above the ESP to maintain a **full liquid column** in the tubing string during equipment **shutdown** periods. It is a simple gravity valve that uses a seat and a poppet to block downward flow in the tubing string, see [Fig. 3.49](#). The check valve prevents leaking of the fluid from the tubing down through the pump when the pump is not running. Fluid flowing through the pump would result in **reverse rotation** of the pump and consequently the whole ESP unit. If, in order to start the ESP unit, power is applied during the period while the pump is turning in the wrong direction, several kinds of damage can occur: the shaft may **break** and the motor or the cable may **burn**.

In installations without a check valve, it must be ensured that the tubing string completely drains back to the well through the pump **before** the motor is started again. This typically requires about 30 min. In cases where **gas locking** is a possibility, the check valve is installed higher than normally, about five to six joints, above the ESP. The increased height of liquid column provides enough volume to break a gas lock should the system automatically shut down.

Drain valves (a.k.a. bleed valves) are used whenever a check valve is present and are installed immediately above the check valve. They prevent pulling a wet tubing string and need not be run if a check valve is not installed.

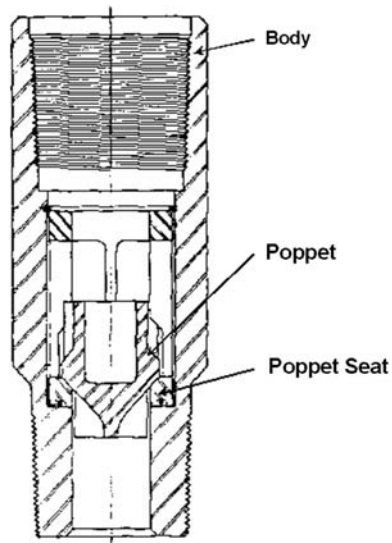


FIGURE 3.49 Main parts of a check valve.

As shown in Fig. 3.50, the drain valve contains a **break-off plug** that, after being sheared, opens a hole in the tubing through which liquid can flow back to the well bottom. Shearing of the plug is best accomplished by using a sinker bar on a wireline but dropping a short bar in the tubing may also be done.

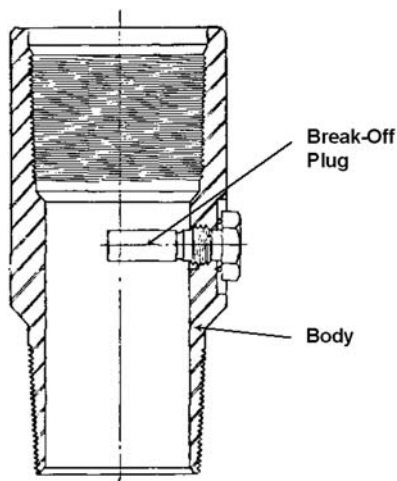


FIGURE 3.50 Main parts of a drain valve.

Backspin relays are useful in times when running a check valve is not recommended. A typical case is when plugging of ESP equipment with scale, sand, etc. is expected and cleaning by pumping down chemicals would be impossible because of the check valve installed at the bottom of the tubing string.

Fluid flowing down the tubing during equipment shutdown turns the pump and the motor in the wrong direction. As the rotor of the motor turns, it generates a phase-to-phase voltage of about 8–12 V in the stator due to residual magnetic flux in the motor. The generated voltage is transmitted through the ESP cable to the surface where it is detected by the backspin relay and the controller prevents restarting the unit.

Centralizers are used to center the motor and pump in the wellbore for proper **cooling** and to prevent rubbing of the power cable against the casing string. They are especially useful in **deviated** wells where the ESP unit tends to stick to one side of the casing. Centralizers also prevent damage of the coating applied to the outside of ESP equipment in corrosive environments.

The **Y-tool** is a special **crossover assembly** [63] of an inverted Y shape (see Fig. 3.51) installed at the bottom of the tubing string, with one side being in line with the tubing and the other side being offset. The straight section (plugged under normal ESP operations) provides a straight run down the hole, whereas the ESP unit is connected to the offset section.

Since the Y-tool makes it possible to access the wellbore sections below the ESP unit, the following main operations can be achieved:

- formation treatment: acidizing, fracturing, etc.,
- well completion: perforation of new pays, etc.,

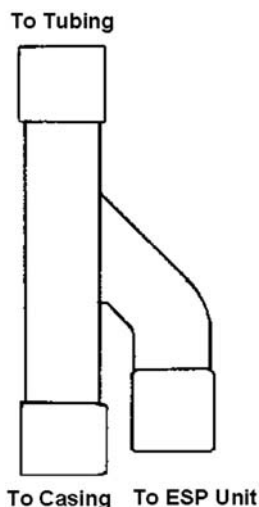


FIGURE 3.51 Schematic drawing of the Y-tool.

- running of **pressure surveys** using wireline or coiled tubing,
- other well logging tasks.

Downhole instrumentation (described in more detail in Chapter 7) is usually installed just below the ESP motor and consists of the required **measuring devices** to continuously monitor downhole **pressure and temperature**, as well as other parameters. Signals from the transducers are sent to the surface through either (1) the regular power cable by adding a DC signal to the motor AC power or (2) a special wire in the power cable. The continuous monitoring of well pressure and temperature gives valuable information on the **inflow properties** of the well and allows the analysis of the ESP unit's operating conditions.

Modern downhole instrumentation packages are solid-state units and use extremely accurate transducers: strain gauges for pressure and resistive thermal devices for temperature [64]. The monitored data can be stored in the unit's memory, as well as displayed in graphical format. The parameters usually monitored include

- pump intake pressure, temperature,
- motor oil or motor winding temperature,
- pump discharge pressure, temperature,
- data on mechanical vibrations,
- electric current leakage, etc.

3.8 SURFACE EQUIPMENT

3.8.1 Wellhead

At the wellhead of an ESP well the casing vent line is normally **connected** to the **flowline** to continuously bleed down any gas that would otherwise accumulate above the dynamic liquid level. If this vent is closed, then gas pressure builds up in the casing annulus and depresses the liquid level below the pump intake causing gas slugs to enter the pump with disastrous consequences to the ESP system's operation. To eliminate such problems the casinghead is directly connected (via a check valve) either to the flowline or, in special cases, to a separate gas line with proper pressure control.

For ESP installations, special wellheads are used to support the weight of the subsurface equipment and to maintain annular control. They must provide a positive **seal** not only around the tubing but also around the **cable**. Different solutions are available; the **Hercules** wellhead used in wells with low wellhead pressures is illustrated in Fig. 3.52 where the downhole power cable is fed directly through the tubing hanger and is sealed off by a rubber packing.

In other wellheads, the power cable is cut at the wellhead and its end is equipped with a power **connector**. The surface power cable, coming from the switchboard, also has a connector and the two are united in the wellhead, see

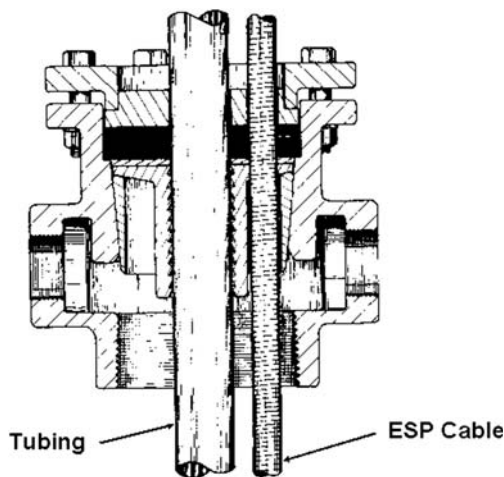


FIGURE 3.52 Hercules-type wellhead.

Fig. 3.53. This type of wellhead allows much higher wellhead pressure ratings than the previous models and is easier to use.

The electric connection of surface and downhole power cables is detailed in **Fig. 3.54**. The tubing hanger holds a **feed-through mandrel** (a.k.a. well penetrator) equipped with the proper seals to contain well pressure. The power cables are connected to the two ends of this device via their three-phase connectors.

3.8.2 Junction Box

The **power cable** coming from the well should be connected to a surface electric cable leading to the **switchboard**. As seen in **Fig. 3.55**, the two cables are joined in the **junction box**, also called a **vent box**. It is a ventilated weatherproof box performing the following three important functions:

1. It provides the electric **connection** between the downhole and the surface electric cables.
2. It **vents** any gas to the atmosphere that might reach this point due to **migration** of well gases up the ESP power cable. The venting of gas eliminates the danger of fire or explosion because gas is not allowed to travel in the cable to the switchboard.
3. It acts as an easily accessible **test point** to electrically check the downhole equipment.

As shown in the figure, breaking of the ESP cable's metal armor near the wellsite is a recommended **precaution** when gas migration in the cable is

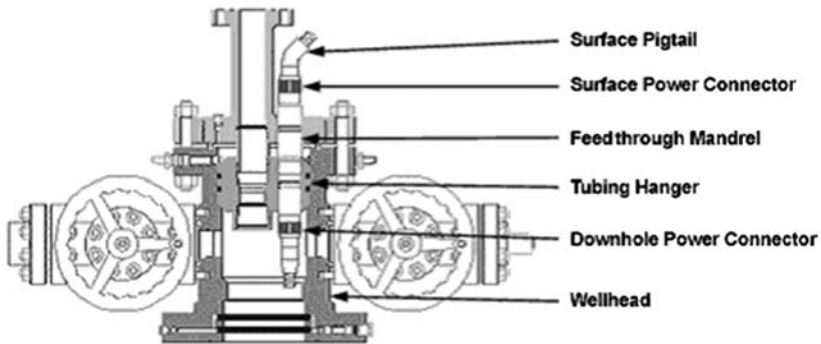


FIGURE 3.53 ESP wellhead with power connectors.

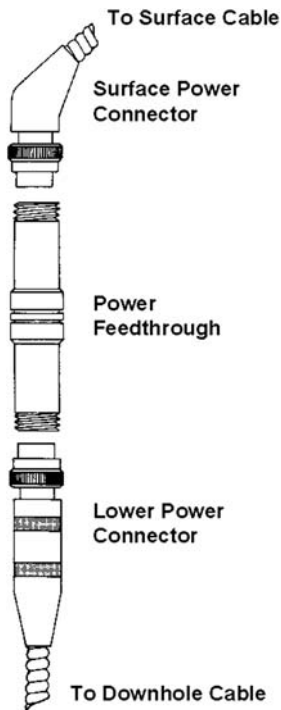


FIGURE 3.54 Cable connectors for an ESP wellhead.

suspected. The junction box should be located at a minimum distance of 15 ft from the wellhead and should be elevated to 2–3 ft above the ground. Connection of the cables requires no splicing or taping and is, therefore, more reliable.

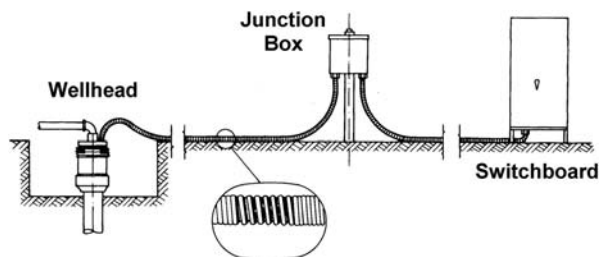


FIGURE 3.55 Surface arrangement of an electrical submersible pump installation.

3.8.3 Switchboard

The **switchboard** is the control center of a conventional ESP installation and acts as a **motor controller** and, consequently, controls the operation of the whole installation. It has the following functions:

- provides a controlled on/off **switching** of the ESP equipment to the power supply using high-capacity switch disconnectors or vacuum contactors,
- **protects** the surface and downhole equipment from a wide variety of problems,
- **monitors** and **records** the most important operating parameters.

Standard switchboards work under a constant electric frequency and vary in size, design, and power ratings (VSDs will be discussed in Chapter 4.) Originally, ESP switchboards contained magnetic **relays** but today's sophisticated units utilize **solid-state** circuitry and provide more advanced functions and an instantaneous response.

In addition to providing on/off switching from the electric network, the switchboard can protect the ESP equipment from problems occurring either downhole or on the surface. The most important downhole problems a switchboard can prevent are

- **overloading** of the motor due to a multitude of reasons such as high liquid gravity, undersized motor,
- **underloading** of the motor due to a **pump-off** condition or an oversized unit,
- unbalanced currents,
- excessive number of starts.

Several malfunctions of the surface power supply may cause problems and the switchboard is designed to protect against

- too high or too low input voltages
- voltage **unbalance**
- **transient** voltages (for example, lightning strikes),
- wrong phase rotation.

After automatic shutdown because of downhole or surface problems, the ESP unit is automatically restarted after an adjustable time delay.

The switchboard **monitors** the most important operational parameters such as line **currents** and **voltages**, **power factor**. For all parameters, **set points** can be input to trigger different alarms. The switchboard often contains a **recording ammeter** that continuously records the current taken by the motor, facilitating the diagnostic **analysis** of the system's operation. Displays of downhole measuring devices are also installed in the switchboard. Current models can **store** operational parameters in memory for later retrieval and can **communicate** by radio or wireline to central stations or the field's **SCADA** system.

3.8.4 Transformers

In most cases, the available surface voltage is not compatible with the required motor voltage and **transformers** must be used to provide the right voltage level on the surface. Power distribution in the oil field is at voltages of 6,000 V or higher, while ESP equipment operates at voltages between around 250 and 4,000 V. Transformers for ESP installations are **oil-filled**, **self-cooling** units and are available in three-phase standard, three-phase autotransformer configurations or a bank of three single-phase transformers can be used. At higher primary voltages, the use of three single-phase transformers is more economical than that of a three-phase transformer.

Transformers are selected on the basis of **voltage levels** and power **ratings**. The required surface voltage heavily depends on the setting depth of the ESP equipment because the voltage drop in the power cable increases with cable length. This voltage drop plus the selected motor voltage give the necessary surface voltage.

3.8.5 Miscellaneous Equipment

ESP cable is run and pulled on **cable reels** that provide the necessary protection against mechanical damage. When running or pulling cable, a cable **guide wheel** hung above the wellhead assists in the manipulation of the power cable.

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Chapter 4

Use of ESP Equipment in Special Conditions

4.1 INTRODUCTION

The conventional electrical submersible pump (ESP) installation described in detail in Chapter 3 cannot be applied to many types of oil wells if operating conditions differ from those ideal for the operation of the centrifugal pump. This is because the ESP, due to its operational principle, can be best used to lift **single-phase** liquids of **low** viscosity at relatively **large** flow rates. Because the operation of the whole ESP system is essentially determined by the performance of its pump, less-than-optimal conditions for the pump may very adversely affect the effectiveness of this artificial lifting technology.

During the long history of the application of ESP systems, different solutions to overcome the adverse conditions for centrifugal pumps have been developed and this chapter is devoted to the description of the methods available today. These solutions may involve the introduction of novel equipment or procedures, or both, all with the final objective of stretching the application ranges of ESP systems. The topics covered in this chapter include the description of the performance of ESPs when pumping highly viscous liquids, fluids with free gas, or abrasive solid contents. A special section is devoted to installations using variable-speed drive (VSD) units that enable the operator to eliminate the ESP system's greatest inherent weakness: the ESP's **narrow** application range in liquid rates.

4.2 PUMPING VISCOUS LIQUIDS

4.2.1 Introduction

The use of submergible centrifugal pumps in heavy oil wells, although not normally considered in practice, has a definite operational advantage. The reason is that high-viscosity heavy oils usually have very low bubblepoint pressures and there is practically no free gas present at the pump suction. This condition eliminates all detrimental problems caused by free gas entering the pump, thus making the use of ESP systems in heavy oil fields a viable approach.

As detailed earlier, ESP performance curves are obtained experimentally by using water as a test fluid. Therefore the design and analysis of ESP installations based on these curves can be properly accomplished only if water or a light crude is produced by the well. However, many times, ESP units are used to produce crude oils or emulsions with considerably **high viscosities**. Liquids with viscosity much greater than that of water cause increased frictional losses and disk friction inside the centrifugal pump's stages, resulting in lower developed heads, correspondingly decreased pump efficiencies, and high brake horsepower requirements. It should be mentioned that performance parameters at shut-off conditions (zero liquid rate) are independent of liquid viscosity. The main effects of increased liquid viscosity are the following:

- pump capacity rapidly drops,
- the head developed by the pump stage also decreases,
- the required power to drive the pump increases,
- the pump's efficiency is reduced.

Thus the standard performance curves obtained from tests made with water as a transport medium cannot be used with the required accuracy if viscous liquids are handled by the ESP installation. If under such conditions viscosity is not taken into consideration when selecting the equipment, the ESP unit would be extremely **overloaded**. Although the complete effect of liquid viscosity on centrifugal pump performance is not yet known, several empirical methods exist for converting standard pump performance curves, if measured ones are not available. The following sections will describe the most common solutions to this problem.

The traditional procedures of the conversion of water performance curves do not account for the effect of pump stage geometry and utilize uniform conversion factors for all geometries [1]. Also the increase in flowing temperature due to the heat generated by the pump is usually disregarded. Therefore, in cases when oils of greater viscosity have to be produced, the conventional procedures of viscosity correction detailed in the following should be treated with caution.

Special problems arise when crudes with high water cuts are produced where the viscosity of the mixture may be much greater than that of the crude oil alone. Such **emulsions** may behave like **non-Newtonian** fluids and the description of their flow performance may be very difficult. In such cases, complete performance testing of the ESP using the actual liquid is advised.

4.2.2 The Hydraulic Institute Model

For the conversion of centrifugal pump performance curves to viscosities greater than the viscosity of water, **Stepanoff** [2] presented the first calculation procedure, but the most frequently used model was developed by the

Hydraulic Institute. The empirical procedure suggested in their publication [3] is based on experimental data and provides a means to determine the centrifugal pump's performance for handling viscous liquids based on the pump's performance in water. The conversion involves the use of several diagrams and is valid in the following ranges of parameters, although it is universally used:

- pump capacities (liquid rates) in the range of 3,400–340,000 bpd,
- pumping heads between 6 and 600 ft,
- liquid viscosities from 4 to 3,000 cSt (40–15,000 SSU).

The conversion of water performance parameters is based on the determination of **correction factors** and the following formulas:

$$Q_{\text{visc}} = C_Q Q_w \quad (4.1)$$

$$H_{\text{visc}} = C_H H_w \quad (4.2)$$

$$\eta_{\text{visc}} = C_\eta \eta_w \quad (4.3)$$

where

Q_{visc} , H_{visc} , η_{visc} = rate, head, and efficiency for the viscous case, respectively

Q_w , H_w , η_w = rate, head, and efficiency for the water case, respectively

C_Q , C_H , C_η = rate, head, and efficiency correction factors, respectively.

The calculation of the correction factors involves visual reading of two diagrams, which is a time-consuming and inaccurate procedure. To improve pump sizing for viscous liquids, **Turzo et al.** [4] curve-fitted the original **Hydraulic Institute** diagrams and presented the numerical model detailed in the following.

The correction factors are the sole functions of the corrected liquid rate, Q^* , that depends on the viscosity of the liquid, the head, and the rate belonging to the pump's **best efficiency point** (BEP) on the water performance curves. The authors developed the following formulas to calculate Q^* :

$$Q^* = \exp\left(\frac{39.5276 + 26.5605 \ln(\nu) - y}{51.6565}\right) \quad (4.4)$$

$$y = -7.5946 + 6.6504 \ln(H_{\text{wBEP}}) + 12.8429 \ln(Q_{\text{wBEP}}) \quad (4.5)$$

where

Q^* = corrected liquid rate, 100 gpm

Q_{wBEP} = water rate at BEP, 100 gpm

H_{wBEP} = head at BEP, ft

ν = liquid kinematic viscosity, cSt.

Based on the value of Q^* , the correction factors in Eqs. (4.1)–(4.3) are determined from the equations given below. The **Hydraulic Institute** model uses two single correction factors for the determination of the liquid rate and the efficiency of the centrifugal pump in viscous service:

$$C_Q = 1.0 - 4.0327 \cdot 10^{-3} Q^* - 1.724 \cdot 10^{-4} (Q^*)^2 \quad (4.6)$$

$$C_\eta = 1.0 - 3.3075 \cdot 10^{-2} Q^* + 2.8875 \cdot 10^{-4} (Q^*)^2 \quad (4.7)$$

where

C_Q, C_η = rate and efficiency correction factors, respectively, –
 Q^* = corrected liquid rate, 100 gpm.

The handling of pump head is different and starts by selecting four points on the original water performance curve at 60%, 80%, 100%, and 120% of the water rate belonging to the BEP. These four points are corrected individually by using their respective correction factors, with the formulas

$$C_{H0.6} = 1.0 - 3.68 \cdot 10^{-3} Q^* - 4.36 \cdot 10^{-5} (Q^*)^2 \quad (4.8)$$

$$C_{H0.8} = 1.0 - 4.4723 \cdot 10^{-3} Q^* - 4.18 \cdot 10^{-5} (Q^*)^2 \quad (4.9)$$

$$C_{H1.0} = 1.0 - 7.00763 \cdot 10^{-3} Q^* - 1.41 \cdot 10^{-5} (Q^*)^2 \quad (4.10)$$

$$C_{H1.2} = 1.0 - 9.01 \cdot 10^{-3} Q^* + 1.31 \cdot 10^{-5} (Q^*)^2 \quad (4.11)$$

where

C_H = head correction factors, –
 Q^* = corrected liquid rate, 100 gpm.

At this point, using the calculated correction factors, the pump's performance curves can be plotted. Because viscosity at **shut-in** conditions does not affect neither the head nor the efficiency, the original shut-in head and efficiency can directly be plotted on the new curves. Thus pump head is known in five points, which makes it possible to estimate the head performance in a quite broad range of liquid flow rates.

The **brake horsepower** required to drive the pump at any flow rate can easily be calculated if the head, the liquid specific gravity, and the pump efficiency are known. The following generally applicable formula should be used:

$$BHP = 7.368 \cdot 10^{-6} \frac{Q H \gamma_l}{\eta} \quad (4.12)$$

where

BHP = brake horsepower required, HP
 Q = pump capacity, bpd

H = head developed by the pump, ft

γ_l = liquid specific gravity,—

η = pump efficiency, %.

Example 4.1

Calculate the head and efficiency curves of the pump whose water performance curves for one stage are represented by the data given in the following table, if a liquid with $\gamma_l = 0.9$ and a viscosity of $\nu = 88$ cSt is pumped.

Point	Q (bpd)	H (ft)	Eff. (%)
0.6 Q_{wBEP}	540	27.9	50.9
0.8 Q_{wBEP}	720	25.5	60.3
Q_{wBEP}	900	21.8	64.0
1.2 Q_{wBEP}	1,080	15.2	55.4

Solution

First the corrected liquid rate is calculated, based on the value of y given in Eq. (4.5), where Q_{wBEP} must be converted to 100 gpm (gallons per minute) as follows:

$$Q_{wBEP} = 900 \cdot 42/1,440 = 26.25 \text{ gpm} = 0.2625 \cdot 100 \text{ gpm}.$$

$$y = -7.5946 + 6.6504 \ln(21.8) + 12.8429 \ln(0.2625) = -4.276.$$

Now, using Eq. (4.4), Q^* is found:

$$Q^* = \exp[(39.5276 + 26.5605 \ln 88 + 4.276)/51.6565] = \exp 3.15 = 23.34.$$

The correction factors are found from the corrected rate, Q^* , calculated earlier, by using Eqs. (4.6)–(4.11):

$$C_Q = 1 - 4.0327 \cdot 10^{-3} \cdot 23.34 - 1.724 \cdot 10^{-4} \cdot 23.34^2 = 0.812.$$

$$C_\eta = 1 - 3.3075 \cdot 10^{-2} \cdot 23.34 + 2.8875 \cdot 10^{-4} \cdot 23.34^2 = 0.385.$$

$$C_{H0.6} = 1 - 3.68 \cdot 10^{-3} \cdot 23.34 - 4.36 \cdot 10^{-5} \cdot 23.34^2 = 0.890.$$

$$C_{H0.8} = 1 - 4.4723 \cdot 10^{-3} \cdot 23.34 - 4.18 \cdot 10^{-5} \cdot 23.34^2 = 0.873.$$

$$C_{H1.0} = 1 - 7.00763 \cdot 10^{-3} \cdot 23.34 + 1.41 \cdot 10^{-5} \cdot 23.34^2 = 0.844.$$

$$C_{H1.2} = 1 - 9.01 \cdot 10^{-2} \cdot 23.34 + 1.31 \cdot 10^{-4} \cdot 23.34^2 = 0.797.$$

The pumping rates valid for the viscous case are found from Eq. (4.1):

$$Q_{\text{visc}1} = 0.812\,540 = 438.5 \text{ bpd},$$

$$Q_{\text{visc}2} = 0.812\,720 = 584.6 \text{ bpd},$$

$$Q_{\text{visc}3} = 0.812\,900 = 730.8 \text{ bpd}, \text{ and}$$

$$Q_{\text{visc}4} = 0.812\,1,080 = 876.9 \text{ bpd}.$$

The heads belonging to these liquid rates are calculated from Eq. (4.2):

$$H_{\text{visc}1} = C_{H0.6} 27.9 = 0.89\,27.9 = 24.9,$$

$$H_{\text{visc}2} = C_{H0.8} 25.5 = 0.873\,25.5 = 22.3,$$

$$H_{\text{visc}3} = C_{H1.0} 21.8 = 0.844\,21.8 = 18.4, \text{ and}$$

$$H_{\text{visc}4} = C_{H1.2} 15.2 = 0.797\,15.2 = 12.2.$$

Finally, the pump efficiencies at these liquid rates, according to Eq. (4.3), are

$$\eta_{\text{visc}1} = 0.3854\,50.9 = 19.6,$$

$$\eta_{\text{visc}2} = 0.3854\,60.3 = 23.2,$$

$$\eta_{\text{visc}3} = 0.3854\,64.0 = 24.7, \text{ and}$$

$$\eta_{\text{visc}4} = 0.3854\,55.4 = 21.3.$$

The calculated and the original performance curves are presented in Fig. 4.1. As seen, the pump efficiency radically decreases and the head developed by the pump also drops if crude with a higher viscosity is pumped.

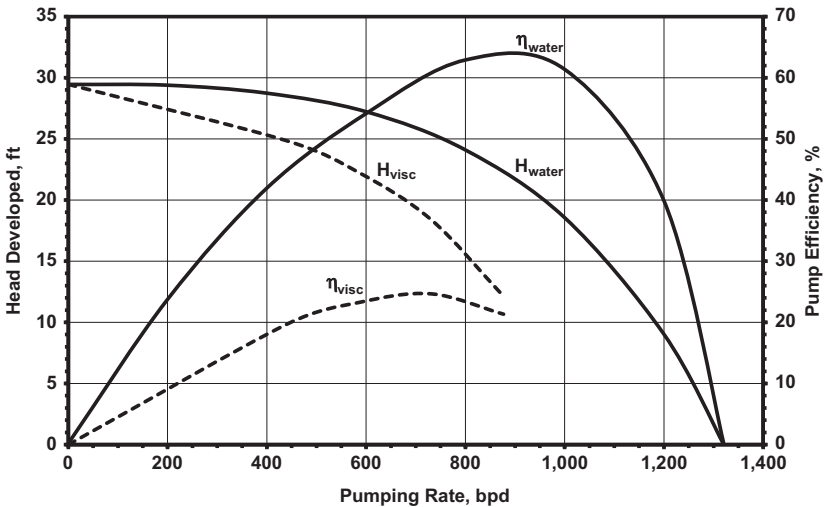


FIGURE 4.1 Pump performance curves for viscous service, Example 4.1.

4.2.3 Other Models

Some ESP equipment manufacturers developed viscosity corrections different from the **Hydraulic Institute** model and recommended the use of tabulated correction factors [5,6]. The final correction of the performance curves is performed similar to the model previously discussed and Eqs. (4.1)–(4.3) are used. An additional correction factor, C_{BHP} , is introduced for calculating the brake horsepower required for viscous service and is used in the following formula:

$$BHP_{visc} = C_{BHP} BHP_w \gamma_l \quad (4.13)$$

where

BHP_{visc} = the required brake horsepower for the viscous case, HP

BHP_w = the required brake horsepower for the water case, HP

C_{BHP} = brake horsepower correction factor,

γ_l = specific gravity of the produced liquid, —.

Table 4.1 presents the latest version [7] of the correction table that, according to one manufacturer, provides standard approximations for most cases. The table contains correction factors for pump capacity, head, efficiency, and brake horsepower, all in the function of liquid viscosity measured in Saybolt seconds universal (SSU) unit. To convert viscosities from the usual unit of centistokes (cSt) to SSU, use the following formula:

$$SSU = 2.273 \left(cSt + \sqrt{cSt^2 + 158.4} \right) \quad (4.14)$$

where

SSU = liquid viscosity

cSt = liquid viscosity.

Example 4.2

Calculate the pump performance curves for the case described in Example 4.1 by using the corrections given in Table 4.1. The brake horsepower values read at the four rates from the water performance curves are 0.22, 0.225, 0.22, and 0.21 HP.

Solution

First calculate the viscosity in SSU unit from Eq. (4.14):

$$\nu = 2.273 \left[88 + (88^2 + 158.4)^{0.5} \right] = 402 \text{ SSU.}$$

The correction factors are taken from Table 4.1 at a liquid viscosity of 400 SSU:

$$C_Q = 0.847; \quad C_H = 0.909; \quad C_\eta = 0.497; \quad C_{BHP} = 1.549.$$

TABLE 4.1 Viscosity Correction Factors According to Centrilift [7]

Viscosity (SSU)	Correction Factors			
	Capacity	Head	Efficiency	BHP
50	1.000	1.000	0.945	1.058
80	0.980	0.990	0.870	1.115
100	0.970	0.985	0.825	1.158
150	0.947	0.970	0.736	1.248
200	0.924	0.958	0.674	1.341
300	0.886	0.933	0.566	1.460
400	0.847	0.909	0.497	1.549
500	0.819	0.897	0.462	1.590
600	0.792	0.883	0.434	1.611
700	0.766	0.868	0.410	1.622
800	0.745	0.858	0.390	1.639
900	0.727	0.846	0.368	1.671
1,000	0.708	0.833	0.349	1.690
1,500	0.659	0.799	0.307	1.715
2,000	0.621	0.771	0.272	1.760
2,500	0.590	0.750	0.245	1.806
3,000	0.562	0.733	0.218	1.890
4,000	0.518	0.702	0.278	2.043
5,000	0.479	0.677	0.149	2.176

BHP, brake horsepower.

Using these correction factors, any point on the original pump performance curves can be converted to the conditions of producing the given viscous liquid. For example, the BEP is modified as follows, using Eqs. (4.1)–(4.3) and Eq. (4.13):

$$Q_{\text{visc}} = 0.847\ 900 = 762\ \text{bpd},$$

$$H_{\text{visc}} = 0.909\ 21.8 = 19.8\ \text{ft},$$

$$\eta_{\text{visc}} = 0.497\ 64 = 31.8\%,\ \text{and}$$

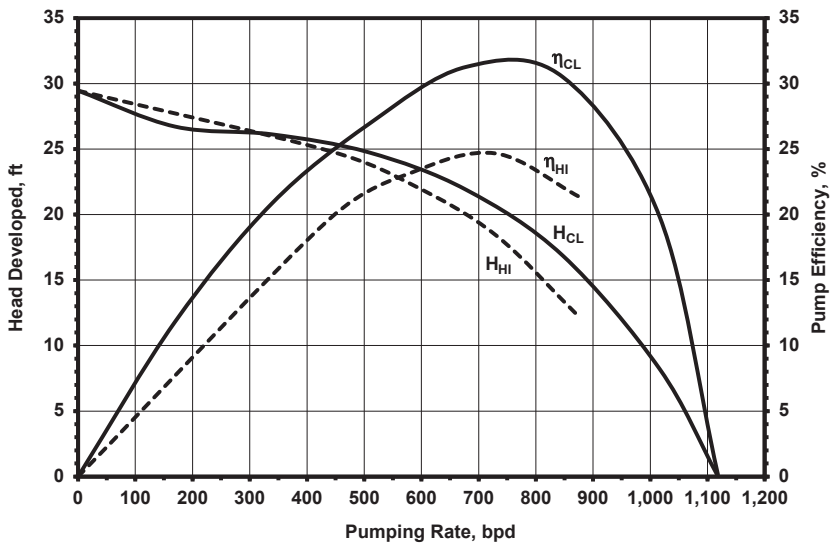


FIGURE 4.2 Comparison of calculated viscous performance curves for Example 4.2.

$$BHP_{\text{visc}} = 1.549 \times 0.22 = 0.34 \text{ HP.}$$

Fig. 4.2 presents the calculated performance curves in bold line, in which the results of Example 4.1 (utilizing the **Hydraulic Institute** model) are shown in dashed line. As seen, the calculated heads are very similar and the efficiency values are also in good agreement. Brake horsepower values are plotted in Fig. 4.3 in which the original water performance curve is shown along with results of corrections by the **Hydraulic Institute** and the **Centriflift** models.

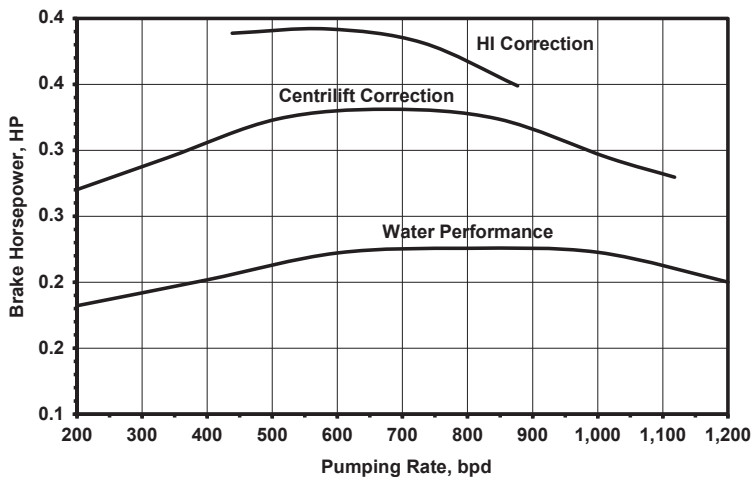


FIGURE 4.3 Comparison of brake horsepower curves for Example 4.2.

Several investigations [8,9] heavily criticized the Hydraulic Institute model based on measurements using actual ESPs with viscous liquids. Deviations were observed as high as 25% in liquid rate and about 35% in pump efficiency calculations. The reason for such high inaccuracies can be attributed to the model's **assumptions** and its database; the viscous correction factors were developed for **single-stage** volute pumps operating at lower rotational speeds. Therefore, the Hydraulic Institute correlations cannot be expected to give accurate results if applied to ESPs.

A major manufacturer used a different approach to solve the problem and set up an extensive database [10] from test measurements in a special flow loop for a wide range of conditions. Multistage ESPs of 4'–10' size running between 40 and 75 Hz were used, the maximum liquid rate was 65,000 bpd and the maximum liquid viscosity reached 4,000 cSt. Data were fitted by a three-dimensional technique to develop correlation charts for head and brake horsepower, both expressed in function of viscosity and pump speed. The authors concluded that pumping speed affects the viscous rate correction factor only.

4.3 LOW OR CHANGING PUMPING RATES

Application of ESP technology to unconventional oil production poses several problems that cannot be solved using standard equipment. Typical conditions could be the following or their combinations: low liquid rates, high gas and abrasive contents, and quickly changing inflow conditions. To overcome these problems, new technological solutions have been developed, as detailed in the following sections.

4.3.1 Low-Rate Pumps

Although ESP equipment is typically applied when high to extremely high liquid rates are desired, **low**-volume applications are also possible to compete with sucker-rod pumping technology. In deep low-productivity wells, sucker-rod pumping has many challenges that are associated with well deviation, high temperature, and frequent rod failures. Use of ESP systems under such conditions is made possible through the utilization of the latest technological developments in two main areas: pump **stage** design and **housing** strength improvement. The use of computational fluid dynamics (CFD)-designed stages with impellers of wide vane openings (as described in **Section 3.2.4**) improved pump performance when free gas and sand are present. Deep pump setting depths also necessitate modification of the pump housings to increase their **burst** pressure. All these improvements allowed production of the relatively low rates between 150 and 650 bpd from more than 10,000 ft deep wells [11].

The efficiency and head performance of low-rate ESPs is improved by a patented modification of the impeller [12]. As shown in **Fig. 4.4**, a series of

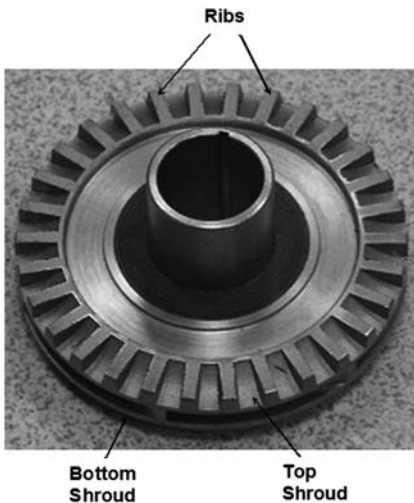


FIGURE 4.4 Impeller with “vortex crown.”

ribs are arranged on the perimeter of the impeller’s top shroud, called “**vortex crown**.” Rotation of the ribs along with the impeller creates an additional flow whose kinetic energy, after the fluid gets in the diffuser vanes, is **converted** to head and thus increases the head developed by the stage. An increase of 15%–25% can be achieved by this process over the head developed by a normal impeller. The other significant advantage of the vortex crown is that the rotating ribs very efficiently **disperse** the larger bubbles of free gas and the homogenized mixture is easier to pump [13].

A major Russian ESP manufacturer offers pump stages with vortex crowns made by powdered metal technology (see **Section 3.2.4**). Pumps are available for low production rates but most higher-capacity pumps may also have this feature that enhances the pump’s operation at high gas–liquid ratios. The manufacturer claims these pumps can handle up to 35% of free gas at suction conditions [14].

4.3.2 Wide-Operating-Range Pumps

Wells producing from unconventional reservoirs usually have abruptly decreasing flow rates so that original rates in the thousands of barrels’ range may drop to the hundreds in 1 or 2 years. The high initial rates demand the use of ESP equipment that must be replaced at a later date with lower capacity artificial lift equipment, often with sucker-rod pumps. This situation is inevitable if standard centrifugal pumps with limited liquid production rate ranges are used. The **FLEXpumpER** pump offered by a major manufacturer [15] can eliminate the change to other technologies because it has an extremely wide

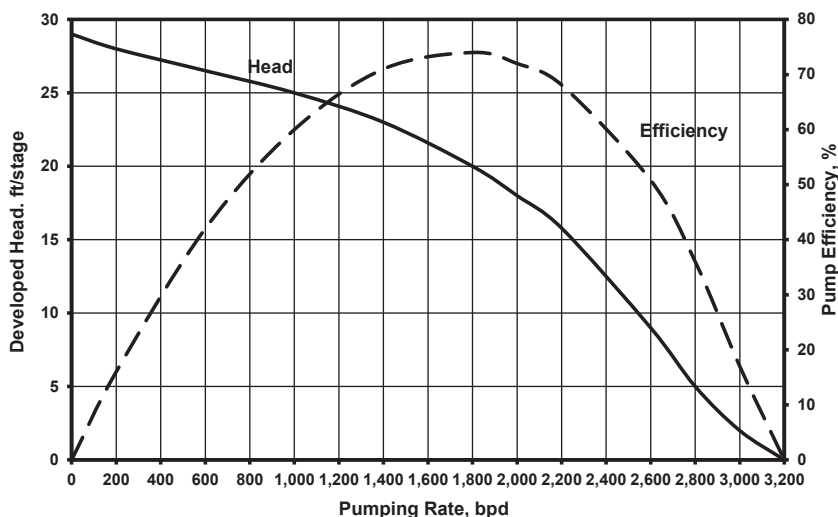


FIGURE 4.5 Performance curves of the Centrifliff FLEXPumpER stage.

range of operation between 50 and 2,900 bpd. It easily adapts to changing well conditions and greatly reduces workover costs by eliminating pump change-outs. As shown in Fig. 4.5, the mixed-flow 4" OD stage has an excellent efficiency at its BEP of 74%. Other beneficial features of the pump are improved gas handling and an improved high-temperature bearing system that works without failure even during a gas slug. Production of abrasives, often associated with unconventional wells, is also properly handled and erosive wear is significantly reduced.

Another wide-range pump is the **MVPER** [16] that has an operating range of 50–2,500 bpd and can be used with free-gas contents exceeding 70%. It was developed for the usual conditions of an unconventional well with high gas–liquid ratios and substantial abrasive production and a quickly declining liquid rate.

4.4 PRODUCTION OF GASSY WELLS

4.4.1 Introduction

The ESP is a **dynamic device** that imparts a high rotational velocity on the fluid entering the impeller, the amount of kinetic energy passed on being proportional to the density of the fluid pumped. Because of their great density, liquid particles receive a great amount of kinetic energy, which, after conversion in the pump stage, increases the flowing pressure. On the other hand, although being subjected to the same high rotational speed, free gas cannot produce the same amount of pressure increase because the kinetic energy

imparted on it by the impeller is significantly lower because of the much lower density of the gas phase. The centrifugal pump's stage, therefore, cannot move free gas with the same velocity as it can do with liquid. Because of these reasons the performance of centrifugal pumps always **deteriorates** if, along with the liquid, **free gas** also enters the pump suction.

In spite of natural separation in the annulus and/or the operation of a gas separator, gas may eventually reach the pump. Free gas in the ESP rapidly ruins the pump's efficiency and increased gas volumes may cause fluctuations of pump output causing **surges** in well production. Surging in the pump leads to cyclic changes in motor load and the current drawn by the motor starts to fluctuate accordingly forcing the surface motor controller to shut down the ESP unit. **Cycling**, i.e., frequent system shutdowns and restarts, eventually damages the motor and the whole installation's run life is severely reduced.

The presence of free gas at pump intake and discharge involves additional implications in the design of the ESP installation.

- The two-phase mixture flowing from the intake through the pump stages undergoes a continuous change in pressure that modifies fluid **properties** such as density and volume. The performance of the subsequent pump stages, therefore, will be different, if compared to a case where a single-phase liquid is pumped through all stages.
- In addition to the modified pump performance, the performance of the **well tubing** changes, too, because gas evolving in the tubing above the ESP unit decreases the average flowing density. This effect can considerably reduce the required pump discharge pressure and an appropriate correction in **total dynamic head (TDH)** calculations is needed. In more sophisticated solutions such as computer programs, tubing pressure is calculated from a vertical multiphase pressure drop model.

In ideal conditions, wells producing gassy fluids would be produced at pump intake pressures (PIPs) above the well fluid's **bubblepoint pressure** so there is no free gas present at the pump suction. This would require a sufficiently great **submergence** of the pump below the dynamic liquid level causing a high flowing bottomhole pressure severely limiting the well's production rate. This is the reason why, in the majority of cases, specific solutions (nonstandard installation types, gas separators, etc.) have to be considered when producing gassy fluids with ESP units.

4.4.2 Free-Gas Volume Calculations

The presence of free gas at the pump suction depends on the *PIP*, the thermodynamic properties of well fluids, and the well temperature at suction conditions. The case of direct gas production, usually the result of gas coning

around the well, should also be mentioned. Depending on the produced crude's bubblepoint pressure, two cases are possible:

1. for *PIP* values higher or equal to bubblepoint pressure, **no free gas** enters the pump,
2. at *PIP* values lower than the bubblepoint pressure, a progressively greater portion of **solution gas** evolves from the crude oil as free gas.

As the amount of free gas increases the volumetric fluid rate the centrifugal pump has to handle, the free gas as well as the total fluid volume at pump suction conditions must be calculated by using the procedures discussed in the following. It must be noted, however, that the actual amount of gas that gets into the pump is usually less than the free gas present at the pump suction. Depending on the characteristics of the actual installation, part of the free gas can leave the annulus because of **natural** gas separation.

The in situ free-gas volumetric rate present at pump suction conditions is found from the well's production gas—oil ratio (*GOR*) and the amount of solution gas, R_s , still in solution at the *PIP*, as given here:

$$q_g' = q_o(GOR - R_s)B_g \quad (4.15)$$

where

q_g' = in situ gas volumetric rate, ft³/day

q_o = oil volumetric rate, STB/day

GOR = production *GOR*, scf/STB

R_s = solution *GOR* at *PIP*, scf/STB

B_g = gas volume factor at *PIP*, ft³/scf.

Because in most cases no packer is set in ESP installations and the annulus is open at the wellhead the annular space acts as a natural gas separator. In the stagnant liquid column existing above the pump suction, gas bubbles can rise to the dynamic liquid level and the gas thus separated reaches the surface and enters the flowline. The **natural separation** of free gas from the produced liquid in the annulus is a highly discussed topic in the professional literature. Although a sufficiently accurate description of the separation process is not yet available the model proposed by **Alhanati** [17] can be used to approximate the amount of gas separated in the annulus.

Alhanati proposed the use of the following formula to estimate the efficiency of natural gas separation:

$$\eta_n = \frac{v_b}{v_b + v_{sl}} 100 \quad (4.16)$$

where

η_n = natural gas separation efficiency, —

v_{sl} = liquid superficial velocity, ft/s

v_b = terminal bubble rise velocity, ft/s.

The liquid superficial velocity is the in situ velocity of the liquid phase inside the annulus and is determined from

$$v_{sl} = 6.5 \cdot 10^{-5} \frac{q_l}{A} \left[\frac{B_o}{1 + WOR} + B_w \frac{WOR}{1 + WOR} \right] \quad (4.17)$$

where

v_{sl} = superficial liquid velocity, ft/s

q_l = liquid volumetric rate, STB/day

WOR = production water–oil ratio, —

B_o = oil volume factor at pump suction pressure, bbl/STB

B_w = water volume factor at pump suction pressure, bbl/STB

$A = 0.0055 (ID_c^2 - OD_t^2)$ annular area, ft²

ID_c = casing inside diameter, in

OD_t = tubing outside diameter, in.

The terminal velocity of the gas bubbles, v_b , is found from the following formula:

$$v_b = \sqrt{2} \sqrt[4]{\frac{\sigma g (\rho_l - \rho_g)}{\rho_l^2}} \quad (4.18)$$

where

v_b = terminal rise velocity of gas bubbles, ft/s

σ = interfacial tension, lb/s²

g = acceleration of gravity, ft/s²

ρ_l = liquid density, lb/ft³

ρ_g = gas density, lb/ft³.

The densities of the phases are also calculated at pump suction condition as follows:

$$\rho_g = \frac{0.0764 \gamma_g}{B_g} \quad (4.19)$$

where

ρ_g = gas density, lb/ft³

γ_g = specific gravity of the gas, —

B_g = gas volume factor, ft³/scf.

$$\rho_l = 62.4 \left[\frac{\gamma_o}{B_o} \frac{1}{1 + WOR} + \frac{\gamma_w}{B_w} \frac{WOR}{1 + WOR} \right] \quad (4.20)$$

where

ρ_l = liquid density, lb/ft³
 γ_o, γ_w = specific gravities of the oil and water, —
 WOR = production water-oil ratio, —
 B_o, B_w = volume factors of oil and water, bbl/STB.

Based on the calculated value of the efficiency of the natural separation process, the actual volume of the **gas ingested** by the pump is easily found using the total gas volumetric rate calculated in [Eq. \(4.15\)](#). After converting the gas rate to barrels per day unit, we get

$$q_{ing}' = \frac{q_g'}{5.61} \left(1 - \frac{\eta_n}{100}\right) \quad (4.21)$$

where

q_{ing}' = gas volume ingested by pump, bpd
 q_g' = in situ gas volumetric rate, ft³/day
 q_o = oil volumetric rate, STB/day
 η_n = natural gas separation efficiency, —.

The in situ **liquid** volumetric rate entering the pump is found from the water and oil rates:

$$q_l' = q_o(B_o + B_w WOR) \quad (4.22)$$

where

q_l' = liquid rate to pump, bpd
 q_o = oil volumetric rate, STB/day
 WOR = production water—oil ratio, —
 B_o = oil volume factor at pump suction pressure, bbl/STB
 B_w = water volume factor at pump suction pressure, bbl/STB.

The **total fluid** volume that the ESP must handle is calculated as the sum of the liquid and gas rates:

$$q_t' = q_l' + q_{ing}' \quad (4.23)$$

where

q_t' = total fluid rate to pump, bpd
 q_l' = liquid rate to pump, bpd
 q_{ing}' = gas volume ingested by pump, bpd.

For the calculation of many of the terms defined previously, several thermodynamic properties of the liquid and gas phases must be known at the pump suction pressure. However, measured thermodynamic parameters of the

produced fluids are seldom available and the use of general **oil field correlations** is recommended. In the following, selected correlations are described but the reader is warned that many other methods are also available and for accurate results one should select those which give the least amount of error for the given oil field.

The **Standing** bubblepoint pressure correlation [18] allows the calculation of solution GORs at the PIP:

$$R_s = \gamma_g \left(\frac{PIP}{18 \cdot 10^y} \right)^{1.205} \quad (4.24)$$

where

R_s = solution GOR, scf/bbl

$y = 0.00091 T - 0.0125 \text{ } ^\circ\text{API}$

T = suction temperature, $^\circ\text{F}$

API = American Petroleum Institute (API) gravity of oil, —

γ_g = specific gravity of the gas, —.

The gas volume factor is evaluated from the following formula:

$$B_g = 0.0283 \frac{Z T_a}{PIP} \quad (4.25)$$

where

B_g = gas volume factor, ft^3/scf

PIP = pump intake pressure, psia

T_a = intake temperature, $^\circ\text{R}$

Z = gas deviation factor, —.

The **deviation factor** of natural gas mixtures can be calculated by several methods, the simple **Papay** formula [19] follows:

$$Z = 1 - \frac{3.52 p_{pr}}{10^{0.9813 T_{pr}}} + \frac{0.274 p_{pr}^2}{10^{0.8157 T_{pr}}} \quad (4.26)$$

where

Z = gas deviation factor, —

p_{pr} = pseudoreduced pressure, —

T_{pr} = pseudoreduced temperature, —.

The terms $p_{pr} = PIP/p_{pc}$ and $T_{pr} = T_a/T_{pc}$ are the pseudoreduced pressure and temperature, respectively, where the critical pressure and temperature can be found from the **Hankinson–Thomas–Phillips** correlation [20]:

$$p_{pc} = 709.6 - 58.7 \gamma_g \quad (4.27)$$

$$T_{pc} = 170.5 + 307.3 \gamma_g \quad (4.28)$$

where

p_{pc} = pseudocritical pressure, psi
 T_{pc} = pseudocritical temperature, R
 γ_g = gas specific gravity, —.

The volume factor of oil can be obtained from the **Standing** [18] correlation:

$$B_o = 0.972 + 1.47 \cdot 10^{-4} F^{1.175} \quad (4.29)$$

where

B_o = oil volume factor, bbl/STB
 $F = R_s (\gamma_g/\gamma_o)^{0.5} + 1.25 T$
 R_s = solution GOR, scf/bbl
 γ_o, γ_g = oil, gas specific gravities, —
 T = suction temperature, °F.

Example 4.3

Calculate the gas, liquid, and the total fluid volumes at the suction of an ESP pump with the following well data by considering the natural separation of gas in the annulus.

Oil API Degree = 30	$PIP = 1,000$ psi
Oil Sp. Gr = 0.876	Oil Rate = 1,000 STB/day
Gas Sp Gr = 0.6	Production $GOR = 300$ scf/bbl
Suction Temp = 150 F	$WOR = 2$
Casing ID = 6.331 in	Tubing OD = 2.875 in

ID, inside diameter; OD, outside diameter; SP Gr, specific gravity.

Solution

First the solution GOR at the pump suction is found from Eq. (4.24):

$$y = 0.00091 \cdot 150 - 0.0125 \cdot 30 = -0.238.$$

$$R_s = 0.6 \left[1,000 / (18 \cdot 10^{-0.238}) \right]^{1.205} = 147 \text{ scf/bbl.}$$

Next the volume factor of the gas phase is calculated based on the thermodynamic parameters of the gas. Pseudocritical parameters are found from Eqs. (4.27) and (4.28):

$$p_{pc} = 709.6 - 58.7 \cdot 0.6 = 674.4 \text{ psi}$$

$$T_{pc} = 170.5 + 307.3 \cdot 0.6 = 354.9 \text{ R.}$$

The reduced state parameters are

$$p_{pr} = 1,000/674.4 = 1.483, \text{ and}$$

$$T_{pr} = (150 + 460)/354.9 = 1.719.$$

Deviation factor can now be calculated from Eq. (4.26):

$$Z = 1 - 3.52 \cdot 1.483/10^{0.9813 \cdot 1.719} + 0.274 \cdot 1.483^2/10^{0.8157 \cdot 1.719} = 0.871.$$

Volume factor of gas is evaluated from Eq. (4.25):

$$B_g = 0.0283 \cdot 0.871(150 + 460)/1,000 = 0.015 \text{ ft}^3/\text{scf.}$$

The free-gas volume can now be calculated from Eq. (4.15):

$$q_g' = 1,000 (300 - 147) \cdot 0.015 = 2,265 \text{ scf/d.}$$

The volume factor of the oil phase is found from Eq. (4.29):

$$F = 147(0.6/0.876)^{0.5} + 1.25 \cdot 150 = 309.$$

$$B_o = 0.972 + 1.47 \cdot 10^{-4} \cdot 309^{1.175} = 1.1 \text{ bbl/STB.}$$

The next task is the determination of the natural separation efficiency. First find the superficial liquid velocity in the annulus. The annular cross-sectional area is

$$A = 0.0055 (6.331^2 - 2.875^2) = 0.175 \text{ ft}^2.$$

The superficial liquid velocity from Eq. (4.17), assuming $B_w = 1$, is

$$v_{sl} = 6.5E - 5 \cdot 3,000/0.175 (1.1/3 + 2/3) = 1.15 \text{ ft/s.}$$

Gas and liquid densities from Eqs. (4.19) and (4.20), respectively, are

$$\rho_g = 0.0764 \cdot 0.6/0.015 = 3 \text{ lb/ft}^3.$$

$$\rho_l = 62.4 [0.876/(1.1 \cdot 3) + 2/3] = 58 \text{ lb/ft}^3.$$

The terminal bubble rise velocity can now be calculated from Eq. (4.18), by assuming $\sigma = 0.04 \text{ lb/s}^2$ and substituting $g = 32 \text{ ft/s}^2$

$$v_b = 2^{1/2} \{ [0.04 (58 - 3) 32] / 58^2 \}^{1/4} = 0.54 \text{ ft/s.}$$

The efficiency of natural gas separation in the annulus is found from Eq. (4.16):

$$\eta_n = 0.54/(0.54 + 1.15)100 = 32\%.$$

The gas volume ingested by the pump is calculated from Eq. (4.21):

$$q_{ing}' = 2,265/5.61(1 - 0.32) = 274 \text{ bpd.}$$

Liquid volumetric rate entering the pump, by assuming $B_w = 1.0$, is calculated from Eq. (4.22):

$$q_l' = 1,000(1.1 + 1.2) = 3,096 \text{ bpd.}$$

The total in situ fluid volume to be handled by the ESPs is found from Eq. (4.23) as

$$q_t' = 3,096 + 274 = 3,370 \text{ bpd.}$$

4.4.3 Pump Performance Degradation

4.4.3.1 Gas Interference in Centrifugal Pumps

Free gas entering the centrifugal pump's suction affects the performance of the pump stage; the greatest effect being degradation of its head performance curve as compared to single-phase liquid cases. All investigators agree that the problems associated with free gas originate from the performance of the **impeller** and that the diffuser plays a negligible role when pumping multi-phase mixtures. Free gas in the impeller reduces the effective area available for liquid flow, increases hydraulic losses in the stage, and decreases the head developed by the pump.

In the following, main conclusions of experimental studies and **CFD** simulations on the performance of centrifugal pump impellers are presented to illustrate the effects of free gas entering the pump suction [21–23].

Studies of the impeller's operation while pumping liquid with free-gas content showed that pump performance did not significantly change when the free-gas phase was evenly **dispersed** in the liquid. In this case the only effect on pump performance is that the free-gas volume entering the pump increases the total fluid volume the pump must handle (see Example 4.3). The operating point on the pump performance curve shifts to the right and the head developed by the pump consequently decreases. The head curve of the ESP for gassy fluids, therefore, **deteriorates** even for small amounts of free gas evenly dispersed in the liquid phase.

In actual situations, however, flow in the impeller is **inhomogeneous** as caused by the great difference between the densities of the two phases. Owing to their density being two to three orders of magnitude greater than that of gas, liquid particles receive a much greater kinetic energy than gas and move with a greater velocity inside the vane openings. Gas bubbles, on the other hand, are not subjected to high centrifugal forces and move more slowly. Centrifugal forces can thus **segregate** the gas and liquid particles more quickly than the

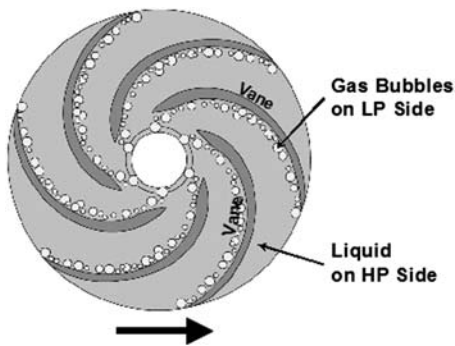


FIGURE 4.6 Distribution of gas bubbles between vanes in the impeller.

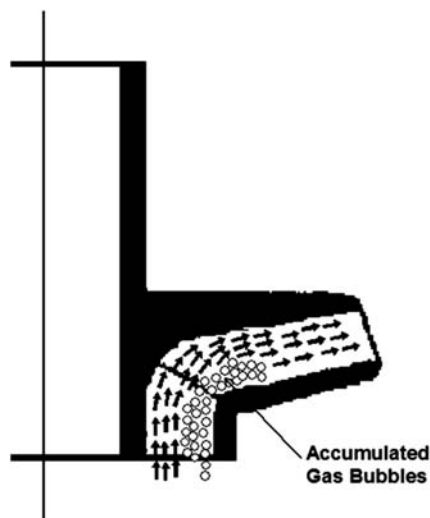
turbulence existing in the vane opening can **mix** them. This is why a homogeneous flow cannot be maintained and gas and liquid particles have a tendency to move inside the impeller independent of each other.

Gas, being the lighter phase, tends to move on the low-pressure sides of the impeller vanes, whereas liquid flows at the high-pressure sides, as shown in Fig. 4.6. Small gas bubbles are dragged by the liquid flow toward the diffuser; this is the situation when low amounts of gas enter the pump. The flow pattern is called **bubble flow** and dispersed fine bubbles are moved by the liquid without any slip between the phases. As free-gas volume at the pump suction increases and more small bubbles enter the impeller intake, bubble flow pattern is no longer possible because existing bubbles coalesce and larger bubbles are created. When the size of these large gas bubbles reaches a critical value, gas becomes stagnant at the impeller intake causing further accumulation of bubbles and formation of a **gas pocket**. Gas pockets cause an unstable operation of the pump stage called **surging** characterized by sudden discharge of liquid and gas slugs from the pump and leading to severe equipment failures. If these pockets are not transferred by the liquid flow toward the impeller discharge at a sufficient rate, they will grow in size and can finally completely block the liquid flow through the impeller and **gas lock** occurs. Fig. 4.7 presents the typical distribution of gas bubbles in the impeller **eye**; the stagnant bubbles block part of the area available for flow. In case of a **gas-locked** impeller, the bubbles occupy the total flow area and liquid production is stopped completely with dire consequences on the operation of the ESP system.

As described previously, most of the problems with free gas come from the **segregation** of the phases in the pump's impeller. Phase segregation is affected by a host of parameters:

- **Stage geometry.** The free-gas-handling ability of a centrifugal pump depends on its specific speed, N_s . The higher the pump's **specific speed**, the

FIGURE 4.7 Accumulation of gas bubbles in the impeller eye.



higher the amount of free gas it can manage. Thus, radial discharge pumps with low specific speeds are much more likely to have segregation problems just described than those with mixed-flow stages having much higher specific speeds. Axial pumps, on the other hand, are the best in handling free gas. Also, pump stage geometries with low net positive suction head (NPSH) values can handle greater free-gas volumes without gas locking.

- **Bubble size.** The size of gas bubbles has a great impact on the drag and the buoyant forces acting on the bubbles immersed in the liquid phase. **Drag forces** try to keep gas bubbles moving with the liquid, whereas **buoyancy forces** try to separate them from the liquid. Smaller bubbles tend to flow with the liquid phase and finely dispersed bubbles are much harder to separate.
- **Phase densities.** The individual densities of the liquid and gas phases, especially their ratio, determine the magnitude of **buoyancy** forces that cause the phases to separate. The closer the densities are to each other, the lower the danger of phase separation.
- **Liquid viscosity.** Liquid viscosity affects the drag forces acting on the gas bubbles and the greater forces occurring at higher liquid viscosities work against phase segregation.
- **Pump speed.** The pump's rotational speed has a dual effect because its increase builds up centrifugal forces trying to segregate the phases and also increases turbulence that reduces the size of the gas bubbles and disperses them in the liquid.

When pumping a multiphase fluid the centrifugal pump's head performance curve typically changes as shown in [Fig. 4.8](#), based on both

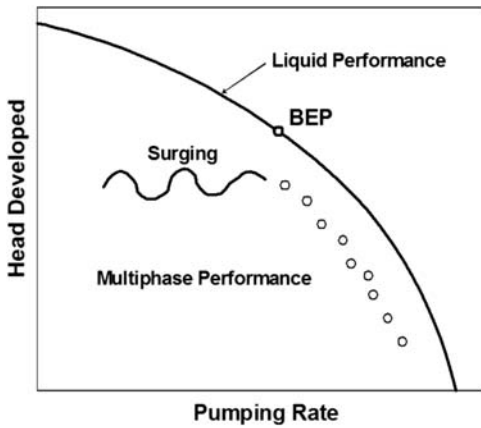


FIGURE 4.8 General trend of head performance degradation. *BEP*, best efficiency point.

experimental and theoretical studies [24–27]. Compared to single-phase liquid performance, the following observations can be made:

- There is a region of unstable pump operation where “**surging**” or “**heading**” occurs because of the cyclic changes in mixture density due to the irregular flow in the impeller. This usually happens to the left of the pump’s BEP.
- Depending on the gas content at suction conditions and on the suction pressure, surging can lead to “**gas locking**” when the impeller is fully blocked by gas and no pumping action takes place.
- Stable operating points (indicated by circles in Fig. 4.8) are expected at the right of the pump’s BEP with developed heads lower than those valid for single-phase liquid operation.

4.4.3.2 Performance Criteria

When investigating the ability of ESPs to handle gas, **Dunbar** [28] and **Lea et al.** [29] agree that pump **suction pressure** (a.k.a. *PIP*) plays a great role in determining the amount of gas that can be accommodated without a significant degradation in pump performance. The widely used **Turpin correlation** [30] relates pump performance to the in situ gas and liquid volumes and the *PIP*. According to **Turpin**, the limits of **stable** pump operation can be evaluated based on the value of the following group:

$$\Phi = \frac{2000 \frac{q_{ing}'}{q_l'}}{3 \text{ } PIP} \quad (4.30)$$

where

Φ = the Turpin function, 1/psi

q_{ing}' = gas volumetric rate ingested by pump, bpd

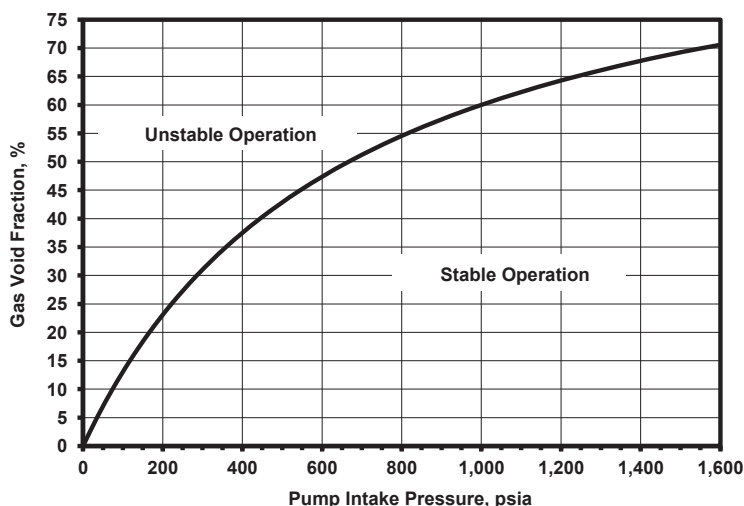


FIGURE 4.9 The Turpin correlation.

q_l' = liquid volumetric rate at suction conditions, bpd

PIP = pump intake pressure, psia.

Stable pump operation can be expected for values of $\Phi < 1.0$, whereas severe gas interference and deterioration of pump performance occurs when $\Phi > 1.0$.

The **Turpin correlation** is depicted in Fig. 4.9 in which instead of the gas–liquid ratio in Eq. (4.30), the **gas void fraction** (the percentage of free gas in the total fluid) is plotted versus the PIP . The areas of stable and unstable operation are separated by a curve representing Eq. (4.30) for $\Phi = 1.0$. As seen in the figure, the amount of free gas that can be handled by an ESP increases with increasing suction pressures (PIP).

Many researchers investigated the effects of other parameters, such as rotational speed, bubble size, and liquid flow rate, that may affect the stability of pump performance in multiphase flow but the mechanism causing surging is still unknown [31]. Of the many surging correlations proposed, mostly for its simple application, the use of the Turpin correlation is still advised.

Example 4.4

Using the **Turpin correlation** decide if the gas volume calculated in Example 4.3 can be handled by an ESP pump.

Solution

The in situ gas void fraction in percentages is easily found as

$$\text{void} = 274 / 3,096 = 8.8\%.$$

Plotting this value at the *PIP* of 1,000 psi in Fig. 4.9 shows that the point falls in the safe operating area; therefore, no gas handling problems are expected.

4.4.4 Possible Solutions

The many available solutions for fighting gas interference in ESP operations will be described in this section. The three main solutions [32] and the ideas behind them are

- not to allow free gas to enter the ESP system by utilizing the **natural separation** of the liquid and gas phases in the casing annulus,
- if natural separation is insufficient and free gas enters the ESP system **separate** and expel free gas into the annulus before it can get into the pump; rotary gas separators (RGSs) are used for this purpose,
- if gas cannot be separated upstream of the pump use special pumps that can **handle** free gas along with the liquid without severe degradation of pumping action.

The necessary installation types and specific pieces of equipment will be discussed in the next sections according to this classification [33,34].

4.4.4.1 Utilization of Natural Gas Separation

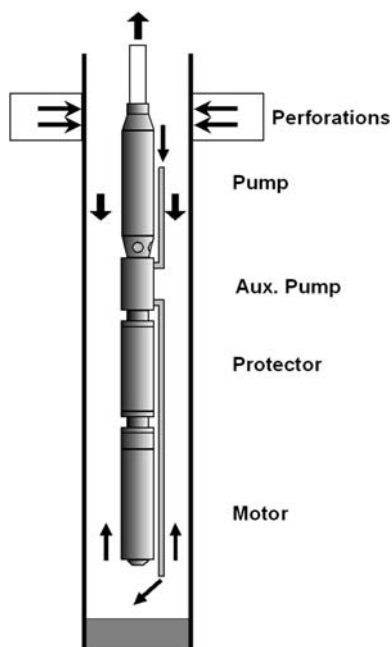
In theory the **reverse-flow gas separator** discussed in Section 3.5 also belongs to this category of gas avoidance in ESP systems. As detailed earlier, this separator has a very narrow application range; this is why solutions capable of removing considerable amounts of free gas from the wellstream are discussed in the following sections.

4.4.4.1.1 Pump Set Below the Perforations

The use of a standard ESP installation (with or without a gas separator), as depicted in Fig. 3.1, but run **below** the well perforations can improve the performance of the ESP system in gassy wells. By running the unit below the perforations, the **natural separation** of liquid and gas in the casing-tubing annulus can be fully utilized, like in a natural **gas anchor** used in sucker-rod pumping installations. Separation by gravity occurs if fluid downward velocity is lower than the rising velocity of the gas bubbles, usually at 0.5 ft/s. This can be ensured only if casing and equipment sizes allow, in which case free gas is automatically directed to the surface without entering the pump suction. In addition to this effect, the *PIP* increases because of the greater pump submergence causing the amount of free gas to decrease or even diminish. This way ESP's efficiency will not be affected.

The basic prerequisite of this solution is the existence of a **rathole** or **sump** in the well, i.e., the perforations being situated above the total well depth [35].

FIGURE 4.10 Electrical submersible pump installation using an auxiliary pump to cool the motor.



The serious drawback of placing the motor below the perforations is the excessive **heating** of the motor because of the lack of liquid flow along the ESP unit. To overcome this problem, one can use high-temperature motors that can be effectively cooled by the stagnant fluid and the formation. The other remedy is shown in [Fig. 4.10](#) in which a smaller **auxiliary pump**, situated between the protector and the ESP, redirects a part of the well's liquid production below the depth of the motor. Flow of the redirected liquid causes movement of well fluids along the motor in the upward direction and provides the cooling effect required for trouble-free operation.

4.4.4.1.2 Use of Motor Shrouds

Motor shrouds (short sections of pipe around the length of the ESP unit) have been successfully used to

- act as simple reverse-flow **gas separators** that, by changing the direction of flow, allow the buoyancy effect to decrease the amount of free gas that enters the pump,
- provide liquid flow along the ESP motor's length to ensure proper **cooling** of the unit.

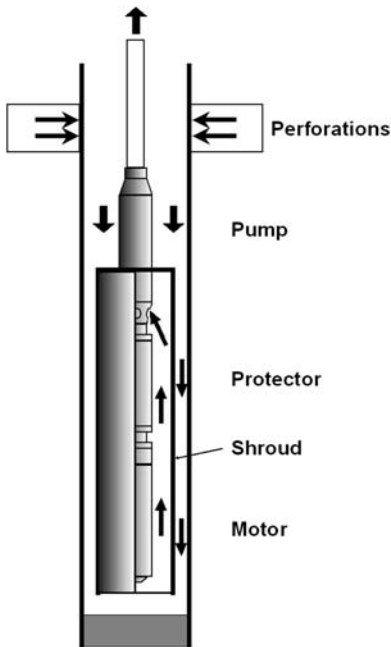


FIGURE 4.11 Typical shrouded electrical submersible pump installation.

The simplest “open-ended” shrouded installation is depicted in Fig. 4.11 in which the ESP unit is run below the perforations. The motor shroud (a pipe section closed at the top and placed around the ESP unit) is hanging from above the pump intake and forces well fluids to flow downward in the casing-shroud annulus. The annular space must have a sufficiently large cross-sectional area to ensure a low (preferably less than 0.5 ft/s) fluid velocity so that gas bubbles may **rise** and be vented up the casing annulus. The shroud also guarantees that produced fluids flow along the motor’s length thus providing proper **cooling**.

In case of greater gas production rates, in addition to the motor shroud, the use of a simple **reverse-flow gas separator** (see Section 3.5) is advised and the required installation is shown in Fig. 4.12. Here the shroud is installed just above the separator intake holes and **vent tubes** direct the separated gas into the casing-tubing annulus above the perforations.

Many times, a **dip tube** is connected to the bottom of the regular motor shroud, as seen in Fig. 4.13. The benefits of this design, as compared to the use of open-ended shrouds, are

- the natural separation of the free gas and liquid is greatly **improved** because of the increased annular cross-sectional area available for downward flow between the casing and the dip tube,

FIGURE 4.12 Shrouded electrical submersible pump installation with gas separator.

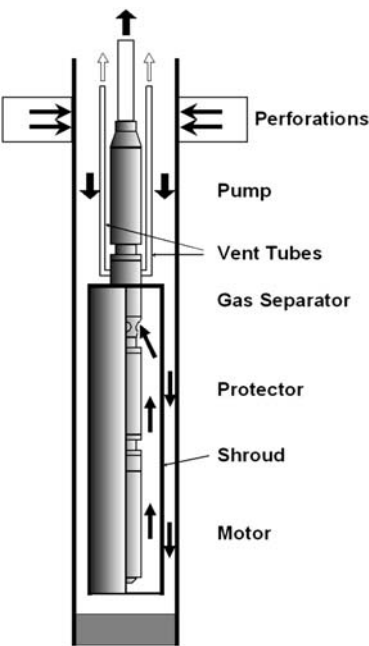
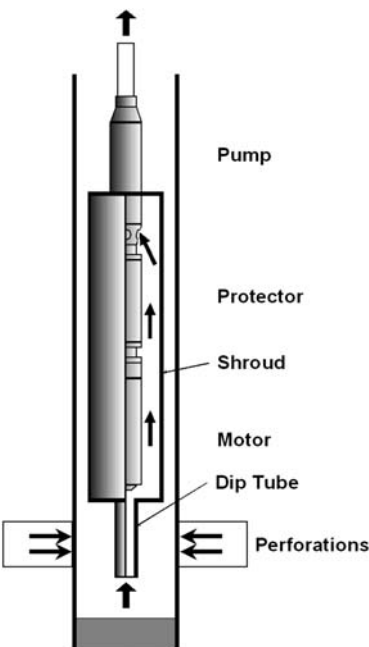


FIGURE 4.13 Shrouded electrical submersible pump installation with dip tube.



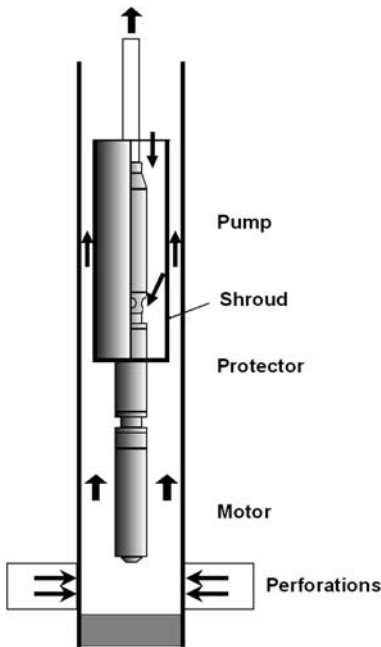


FIGURE 4.14 Electrical submersible pump installation with an inverted shroud.

- well fluids can be produced from a restricted section of a vertical or inclined hole where the ESP unit would not pass,
- this arrangement can also be used in **horizontal wells** with the ESP unit run in the vertical section and the dip tube reaching into the horizontal part of the well.

An **inverted shroud** means a motor shroud open at the top. The shroud is fixed below the pump intake and acts as a reverse-flow **gas separator**, as shown in Fig. 4.14. The ESP unit must be run above the perforations so that the inverted shroud forces well fluids to flow downward in the shroud-unit annular space. Proper design of the shroud diameter ensures that flow velocity here is less than 0.5 ft/s required for the gravitational separation of the gas from the liquid. The reverse-flow velocity in the annulus between the shroud and the ESP unit can be easily controlled by installing at the top of the shroud a **swage** of a different diameter than that of the shroud. Use of this type of ESP installation is advantageous in horizontal wells with severe **slugging** problems because a long inverted shroud acts as a fluid reservoir that keeps the pump primed in periods when large gas slugs are produced by the well.

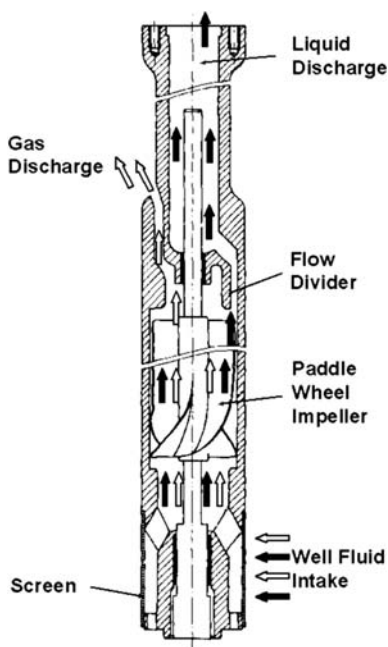
4.4.4.2 Rotary Gas Separators

RGSs work on the principle that a multiphase mixture, if spun at a high speed in a vessel, is separated to its constituent liquid and gas phases because of the different levels of **centrifugal force** acting on the liquid and gas particles. The rotational speed is provided by the separator's shaft, connected to the motor, and separation takes place in the body of the separator. Here liquid is forced to the inner wall of the separator while gas is concentrated near the shaft. A **flow divider** ensures that the separated phases move along different paths and a **crossover device** directs (1) gas into the casing annulus for venting to the surface and (2) liquid to the pump intake.

4.4.4.2.1 Available Types

The first RGS called the “**paddle-wheel**” type appeared in the early 1970s [36] and is shown in Fig. 4.15. It contains four to eight axial blades (vanes) that run parallelly along the length of the separator's shaft. Well fluid containing free gas is sucked in the separator body at the bottom and enters the chamber containing the rotating paddle-wheel impellers. The high centrifugal forces acting on the liquid particles force them to move toward the separator's wall, while gas collects near the shaft due to the much lower forces acting on it. The separated liquid and gas streams are split by the **flow divider** and after passing

FIGURE 4.15 Rotary gas separator of the paddle-wheel type.



the **crossover device**, gas leaves to the annulus while liquid reaches the suction of the ESP.

Although the paddle-wheel separator provided a superior performance in comparison to the **reverse-flow gas separator** discussed in **Section 3.5** and is still in use today, it has a severe operational weakness that limits its efficiency. This comes from the fact that the tips of the impellers, turning at a high speed, pick up part of the liquid from the inner wall of the separator body and mix it with the less dense fluid situated closer to the shaft. This **remixing** is inevitable because liquid velocity at the separator inside wall is close to zero. This is why separation efficiencies decrease in wells with higher free-gas contents. Another disadvantage of this design is the **abrasion** between the tips of the impellers and the separator wall, especially when pumping well fluids containing higher concentrations of sand.

The “**rotating chamber**” type of gas separator [37] eliminates the remixing of liquid and gas phases by isolating the rotating impellers from the stagnant liquid layer present on the inside wall of the separator body. As shown in **Fig. 4.16**, the four impellers are enclosed by a rotor **shroud**. Thus, four **separation chambers** are created where the fluid rotates as a solid body and shearing and turbulence effects responsible for remixing of the phases are minimized.

The multiphase mixture enters the separation chambers where centrifugal accelerations of up to **six times** the acceleration of gravity ensure the separation of the phases. A perfect separation would be possible if a sufficiently long **retention time** could be maintained, i.e., if the fluid stayed in the separator chambers for a long enough time. By maximizing the cross-sectional

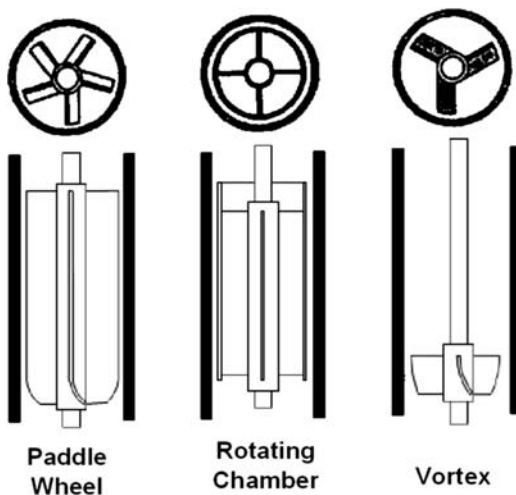


FIGURE 4.16 Different rotary gas separator systems.

area of the chambers, axial fluid velocity can be held at a low level and the retention time is increased accordingly.

At higher fluid rates, of course, fluid retention time in the separator decreases and the maximum efficiency of separation is inevitably limited. The rotating chamber gas separator is good for wells producing high liquid rates and/or highly viscous liquids because it involves the maximum acceleration possible for an effective gas—liquid separation.

The “**vortex**” separator, shown in Fig. 4.16, is a very simple device that has a single axial flow impeller as the only active member situated below the separation chamber. The fluid entering the separator is spun by this impeller inducing a **vortex** in the otherwise empty separator chamber. The vortex thus formed forces liquid to move to the separator wall but gas stays near the shaft. The separated phases are then divided and led to the annulus and the pump, just like in the other separator types. Centrifugal forces in the separator chamber are **lower** than in other rotary separators because the vortex generated in the separator body spins at a speed lower than that of the separator shaft. The efficiency lost this way is regained by the complete elimination of the **remixing** of the gas and liquid phases present in other devices. The vortex separator can be successfully used in wells producing **sand** but is not so effective in viscous fluids and emulsions.

4.4.4.2.2 The Role of Inducers

It has been long understood that the gas separator stages just described do not operate properly if there is not enough pressure pushing the multiphase mixture through them [38,39]. To provide the pressure necessary to overcome the hydraulic losses across the gas separator, **inducers** (a.k.a. **augers**) have been used upstream of the gas separator stage. These are axial-flow pumps with helical open vanes that develop the head required for expelling the separated gas into the well’s annulus.

The typical construction of a gas separator’s impeller assembly is presented in Fig. 4.17. The whole structure can be divided lengthwise into three parts with different shapes and functions; their cooperation makes the effective separation of the gas possible. The bottom section is the **inducer** that has two axial helical blades of a constant pitch disposed 180 degrees apart on the opposite sides of the device’s hub. The blades have a quite small blade angle (about 10 degrees) in respect to the axis’ normal plane and make two complete revolutions around the hub. Their leading edges are sharp to reduce turbulence in the fluids entering the inducer. The main function of the inducer is to force the gas—liquid mixture entering the separator through the rest of the unit. The pressure developed by the inducer must be sufficient to push the separated gas out from the unit and into the annulus.

The next component (in this case, integral to the separator’s impellers, but may also be a separate part) is a set of **guide vanes** that are helical blades with

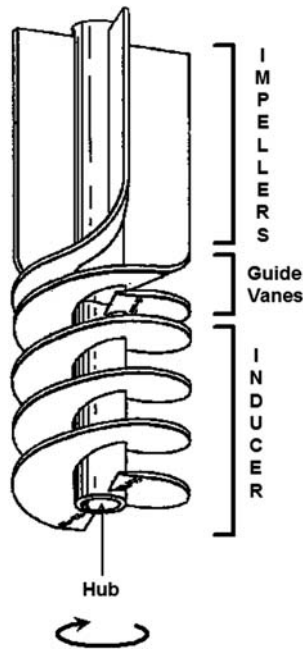


FIGURE 4.17 Typical construction of a gas separator's impeller assembly.

blade angles smoothly but rapidly increasing upward from the small angle of the inducer to 90 degrees (vertical) valid at separator impellers. The guide vanes take up only a short distance on the unit's hub and provide a smooth transition of flow without abrupt changes in direction or velocity. At their exit the tangential velocity of the fluid gets very close to the angular velocity of the separator impellers, minimizing flow disturbances. The guides also ensure that a minimum of pressure loss occurs as fluids move across them onto the separator's impellers.

The multiphase mixture discharged from the guide vanes enters the unit's main **impellers** that perform the gas—liquid separation using centrifugal force. The schematic figure in Fig. 4.17 depicts a paddle-wheel-type separator stage but other kinds of separators can also be used. This is the final and third component of the assembly that occupies a long portion of the hub's length. This is where separation of gas from the liquid phase takes place according to the processes described in the previous section.

In summary, modern-day RGSs typically include the following principal components:

1. fluid **intake** of low flow resistance,
2. the **inducer** that increases the pressure of the incoming multiphase mixture,

3. the **guide vanes** that modify the route of the fluid coming from the inducer into an axial direction,
4. a **separator** chamber where the actual phase separation takes place using any of the devices described in Fig. 4.16,
5. a **flow divider** that ensures that the separated phases move along different paths,
6. a **crossover device** directing gas into the casing annulus for venting to the surface and liquid to the pump intake.

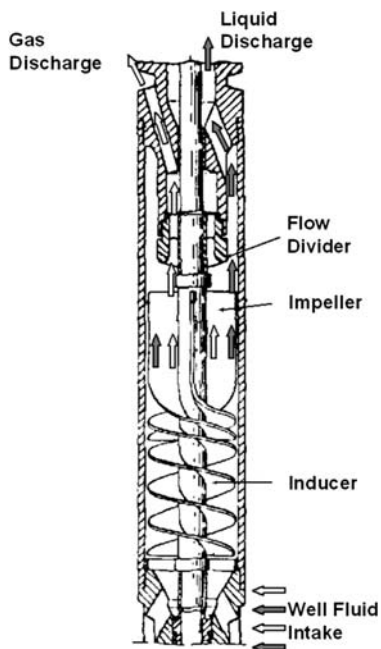
Construction of a typical RGS using paddle wheels is displayed in Fig. 4.18 [40]. As found from extensive laboratory and field experiments, the use of such rotary separators ensures ideal pump performance for in situ gas—liquid ratios as high as 0.6 [41].

4.4.4.2.3 Separation Efficiencies

In principle the separation efficiency of any RGS depends on

- the **retention time** (the time the fluid stays in the separation chamber),
- the magnitude of **turbulence** that occurs in the separator and causes remixing of the phases.

FIGURE 4.18 Rotary gas separator with inducer and paddle wheels.



Therefore the basic requirements for an efficient RGS are

- the **inducer** should develop the proper amount of head to transfer the fluids from the separator into the annulus,
- the **retention time** should correspond to the properties of the well fluids so that phase separation becomes as complete as possible, and
- there should be a minimum of turbulence caused in the separation chamber to reduce **remixing** of the separated phases.

Today's RGSs are very effective in separating free gas from the wellstream at suction conditions. The typical performance of a major manufacturer's product is illustrated in Fig. 4.19 [33], based on laboratory measurements. As shown, increasing liquid rates decrease the separation efficiency because the great mixture velocities entering the gas separator cause the gas phase to move along with the liquid.

Early investigations on the efficiency of RGSs were misleading because they treated the separator's operation independent of the natural gravitational separation taking place in the annulus. **Alhanati et al.** [42] were the first to recognize that the separation processes in the annulus and in the RGS are closely linked. The effectiveness of natural separation in the annulus affects the amount of free gas entering the separator, while the rest of the gas is leaving to the casinghead. The RGS, because it cannot achieve a perfect separation of the two phases, always transfers part of the incoming gas to the pump and part of the liquid into the annulus. The liquid expelled through the separator's gas discharge ports is **recirculated** to the intake and during its

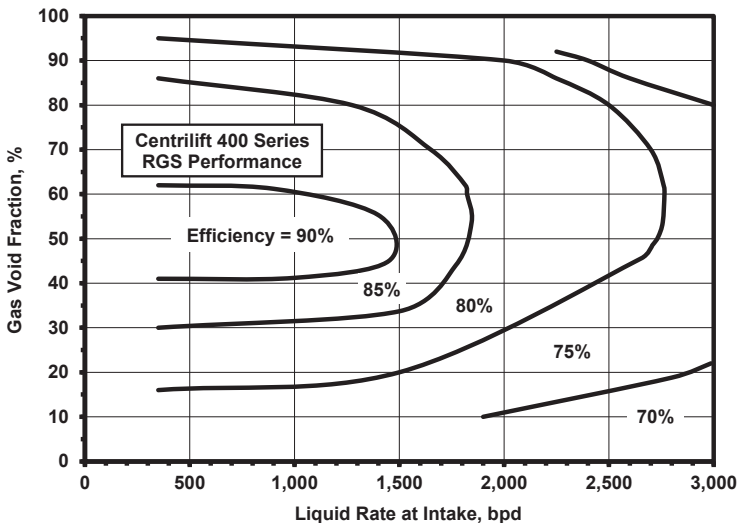


FIGURE 4.19 Separation efficiency of Centrilift 400 series gas separators [33].

downward travel can take some of the free-gas bubbles with it. The natural separation process in the annulus is thus **altered**, and this also changes the composition of the mixture entering the separator. Therefore the separation processes in the annulus and the RGS must be studied **simultaneously**, if meaningful performance data are desired.

The key for the proper description of the separation process lies in the simulation of the hydraulic processes inside the RGS. To expel fluids through the discharge ports, the pressure of the gassy mixture entering the separator has to be increased by the **inducer**. If the head generated by the inducer is sufficient to overcome the pressure losses across the discharge ports, the separator expels fluids into the annulus; otherwise no separation takes place in the RGS.

Laboratory investigations, later confirmed by field data, showed that RGSs operate in two performance regions depending on liquid flow rate. Starting at low rates, high separation efficiencies are attained with increasing liquid rates, but above a limiting rate, efficiency **suddenly** drops to zero. As shown for an example case [42] in Fig. 4.20, the composite separation efficiency of the RGS and the annulus is high at lower liquid rates and then suddenly drops to the level that characterizes the natural separation in the annulus. The reason for the sudden change lies in the **overloading** of the inducer that cannot generate enough head to overcome the pressure drop across the separator's discharge port; the separation process ceases. As gas is not expelled into the well's annulus from the separator, only natural separation in the casing-tubing annulus takes place at a low efficiency. This means that the maximum liquid rate a gas separator can handle is basically determined by the performance of the inducer used.

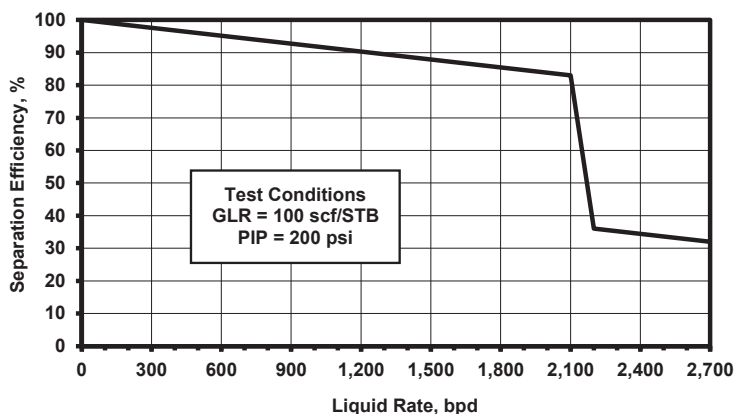


FIGURE 4.20 Separation efficiency of a rotary gas separator and the annulus [42]. *GLR*, gas–liquid ratio; *PIP*, pump intake pressure.

As shown previously, the efficient operation of the RGS heavily depends on the proper functioning of its **inducer**. Inducers are axial flow pumps with **helical blades** and low NPSH values (see [Section 4.4.4.2.2](#)); their most important geometrical properties being the diameter, pitch length, and pitch number. The performance of the RGS can be improved by optimizing the geometry of its inducer [\[38,43\]](#).

4.4.4.3 Gas Handling

The previous sections described the ways

- how free gas can be **prevented** to enter the ESP,
- how free gas that has entered the system can be **separated** before getting into the pump.

If none of these solutions can be applied or their application is not recommended for some reason, free gas **inevitably** reaches the ESP. In such cases the pump must, by using either some special piece of equipment or some modifications, be able to **handle** the gas, i.e., to move it along with the liquid phase. This section describes the various available techniques that make it possible to pump considerable amounts of free gas with an ESP.

4.4.4.3.1 Legacy Methods

4.4.4.3.1.1 Overstaged Pumps The earliest solution was to use more pump stages than normally required, so as to compensate for the smaller heads developed by the first few stages because of gas interference. The so-called “**overstaged**” pumps eliminate the overloading of the upper pump stages usually associated with free-gas production by supplementing the less-than-sufficient TDH developed by the original pump. The utilization of **oversized pumps** with flow capacities greater than the required liquid rate can also help handle greater amounts of free gas through the pump. Both solutions use **identical** pump stage types for making up the entire pump. Their common drawback is that different stages in the pump usually operate at different liquid rates that may be outside the recommended range of the given pump, inevitably leading to mechanical failures.

4.4.4.3.1.2 Tapered Pumps A successful and energy-efficient solution for handling free gas is the use of “**tapered**” pumps. These are ESPs made up of several (at least two) **different** stage designs with the capacities of the successive stages decreasing upward. The idea behind this solution is to allow the lower stages to compress gassy fluids and to continuously decrease the total volumetric flow rate through the pump. These stages do not significantly increase the pressure, only to the level so that free gas is compressed and part of it may go into solution with the oil. The density of the fluid discharged from the high-capacity stages increases and the volumetric rate decreases

accordingly. Therefore, upper stages receive a much-**reduced** fluid rate and can be used to develop the head necessary to lift the fluids to the surface.

The effectiveness of tapered pumps is the function of total fluid rate. At low to medium rates, the gas handling capacity of the ESP is greatly increased because higher-capacity stages with high specific speeds can be used at the bottom of the pump. But if high rates are desired, the upper stages inevitably have higher specific speeds and much lower improvements can be achieved by “tapering” the pump.

The design of tapered pumps should ensure that all stages in the pump operate inside their **optimum** capacity ranges and this requires the use of computer programs [44,45]. Such programs calculate for every stage the density and volume of the fluid as well as the pressure increase developed by the stage and other parameters. The accuracy of the tapered pump design heavily relies on **proper** well data and if well conditions differ considerably from the assumptions used for the design, some or all the stages may happen to operate outside their operating range, destroying all advantages and damaging the whole installation.

4.4.4.3.1.3 Stage Recirculation As radial discharge (pancake) pumps are most likely to get gas-locked, several modifications to stage design were attempted to improve gas tolerance. All designs attempt the **recirculation** of liquid in the pump stage [46] and try to

- break up the gas pockets usually created in regular stages,
- increase the homogenization of the fluid flowing through the stage.

Fig. 4.21 shows one version in which matching **holes** are drilled in the top shroud of the impeller and in the bottom shroud of the diffuser. The holes provide a path for the fluid to flow from the diffuser back into the impeller. The flow thus created greatly reduces gas **segregation** caused by centrifugal forces in the impeller. The circulating liquid flow can also eliminate the gas **pockets** that continuously grow in the eye of regular impellers and eventually lead to a gas-locked condition. The big disadvantage of such solutions is the inherent **reduction** of pumping efficiency by 20%–30% or more. The lower

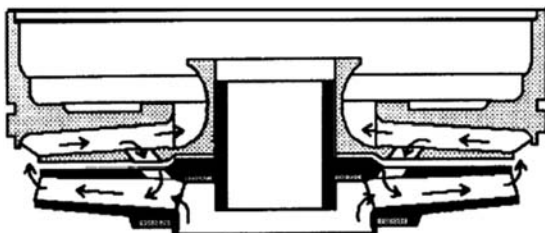


FIGURE 4.21 Pump stage with impeller recirculation.

efficiencies are caused by the reduction of the head developed by the stage because of the partial recirculation of the impeller's discharge to the intake.

4.4.4.3.2 Modern Solutions

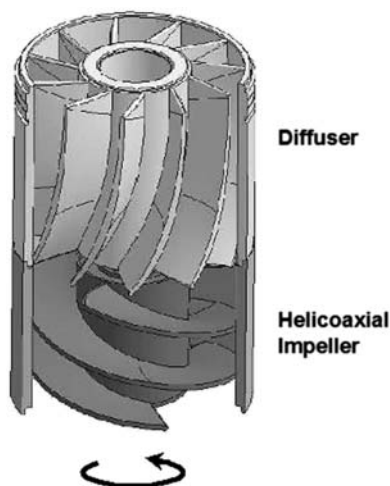
Special devices connected upstream of the pump can be used to improve the ESP's tolerance to free-gas production. A “**gas charger**,” a short lower-tandem pump with high-capacity stages, can be added below the main pump. It pumps the gassy fluids entering the pump suction and while doing so compresses the mixture so that the fluid is lifted easier by the main pump. Gas “**blenders**” disperse free gas in the liquid phase; the small gas bubbles thus created are carried with the liquid instead of resulting in a gas-lock situation.

A major ESP manufacturer's “**advanced gas-handling**” (AGH) unit [47] is a special multistage centrifugal pump that can handle up to 45% of free gas at its suction. The AGH does not separate gas from the liquid but conditions it so that it can be normally transferred by the pump. It significantly reduces the size of gas bubbles and **homogenizes** the gas–liquid mixture by changing the distribution of gas bubbles in the mixture. The AGH is normally installed upstream of the ESP; this way the pump receives a well-homogenized mixture that behaves very similarly to a single-phase fluid and poses no problem to the pump. At higher gas rates the AGH can be run above a gas separator and the combined effects of the two devices can provide ideal conditions for the submersible pump. An added advantage is that the gas content of the mixture in the tubing is increased as compared to cases when gas is separated and led into the annulus. The higher gas–liquid ratio causes a reduction in the required head at the pump discharge and may reduce the hydraulic power needed to produce the well.

Available models can be used in the liquid flow rate range between 500 and 35,000 bpd; they can handle a maximum of 45% free gas at suction conditions. Their operation requires quite substantial power; the smallest unit needs 13 HP and the biggest one needs 117 HP, both at 60 Hz frequency.

4.4.4.3.2.1 The Poseidon Stage A recent development from a leading manufacturer [48,49] makes use of a gas handler having special pump stages originally devised for transferring multiphase mixtures. The “**Poseidon**” centrifugal pump stage shown in Fig. 4.22 contains impellers with heli-coaxial vanes and diffusers providing a smooth axial flow of the produced fluids. This pump stage construction practically eliminates radial centrifugal forces and, therefore, greatly limits the **separation** of the flowing phases that is responsible for most of the problems free gas causes in radial and mixed flow pump stages. The stage, at the same time, does not develop much **head** to compress the flowing mixture so lifting of well fluids must be done by a conventional centrifugal pump connected downstream of the Poseidon stages.

FIGURE 4.22 Schematics of a “Poseidon” stage.



As shown in Fig. 4.23, the geometry of Poseidon impellers and diffusers greatly differs from conventional pump stage geometries by providing **axial** flow of the two-phase mixture. The main beneficial effects of Poseidon stages are

- the multiphase mixture is **homogenized** and the distribution of gas particles in the fluid is equalized,
- gas segregation present in other types of impellers is practically **eliminated** because of the low centrifugal forces developed by the axial flow impeller,
- the **size** of the gas bubbles is reduced by the stage,
- the stage effectively **mixes** the two phases.

The **Poseidon** gas handler is connected between the pump intake and the ESP and acts as a charger or **booster** pump. The homogeneous mixture of liquid and gas leaving the gas handler provides **ideal** conditions for the ESP to lift the fluid to the surface. The unit can handle wellstreams with up to 75% of free-gas content at the pump intake, effectively preventing gas locking of ESPs. Field experience in wells with high gas production rates showed [50,51] that using a few Poseidon stages (less than 20) in conjunction with conventional ESPs can prevent gas locking and the related equipment difficulties. Such installations can also handle single-phase gas slugs for shorter periods and can thus survive transient conditions.

Available models [52] work in the liquid flow rate range between 800 and 9,000 bpd and consume substantial powers between 16 and 200 HP at 60 Hz operation.



FIGURE 4.23 The impeller and diffuser of a Poseidon pump stage.

4.4.4.3.2.2 Special Pumps Special ESPs designed to produce well fluids with very high free-gas content are also available. Some of these are tapered pumps with an intake **inducer** (axial flow pump) at the bottom, specially developed, **low NPSH** pump stages in the middle, and standard stages at the top of the pump assembly. The intake inducer provides a positive pressure for the gas–liquid mixture to enter the upper stages, the low NPSH stages mix and compress the gassy fluid so that the **standard stages** at the top can increase the flowing pressure to the level required for lifting the liquid to the surface. Thanks to the special design of the low NPSH stages, **surging** and **gas locking** are eliminated, thus ensuring a long operational life of the ESP equipment even in gassy wells.

It is generally understood that pumps can handle gassy mixtures only if the segregation of the two phases is restricted and gas and liquid are efficiently mixed by the stage. Gas tolerance can thus be increased if pump stages that fulfill both these requirements are used like in a major manufacturer's multivane pump (**MVP**) stage [53,54]. As shown schematically in Fig. 4.24, the pump's impeller has **split vanes**, a larger inner one and a smaller outer one, both with different curvatures. The outer vanes are located in a leading offset position relative to the inner vanes. The stage is of the mixed-flow variety and has relatively large balance **holes** in its top shroud that can take up 45%–100% of the space between the vanes.

The main features of the MVP stage that allow pumping of high gas rates along with the liquid are

- the large balance holes permit liquid communication from the discharge side and increase mixing of the phases,
- the split vane design reduces the pressure differential between the high- and low-pressure sides of the impeller and keeps the mixture homogeneous,
- the two phases are further homogenized by the flow between the inner and outer vanes,
- the high vane exit angle (between 50 and 90 degrees) increases the liquid outflow velocity.

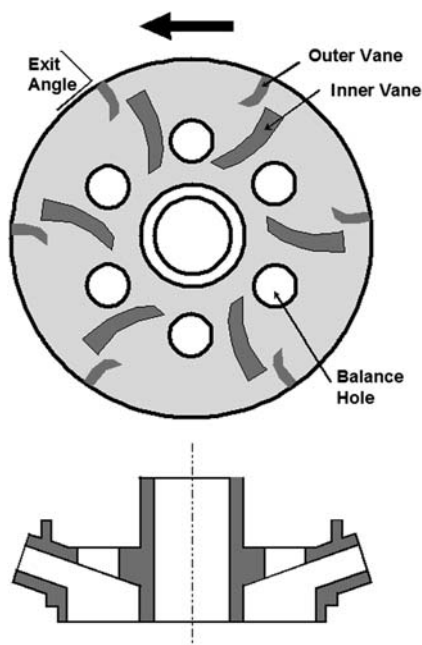


FIGURE 4.24 Impeller of the multivane pump.

Owing to the features described the MVP stage can handle free-gas contents exceeding 70% and can eliminate gas-lock situations [55]. It can be used alone or as a booster pump upstream of a conventional centrifugal pump because the MVP stages homogenize the mixture to such levels that pumping it by normal pumps poses no problems.

4.4.5 Conclusions

As it could be seen from the preceding discussions, the production of gassy wells with high-free-gas content is no longer outside the application range of the ESP system. By using the proper equipment, ESP installations can be used to lift oil wells with very high gas production rates. The possible solutions include the proper choice of pump stage types and the use of tapered pumps, RGSs, AGHs, and the combinations of these.

All devices used to mitigate the problems with free gas (RGSs, AGH units, Poseidon pumps) **consume** energy and their power requirement must be considered when selecting the motor to drive the ESP system. The power requirement of such devices is normally given by the manufacturer at some **fixed** electrical frequency, 60 or 50 Hz. If the ESP system is used at any other

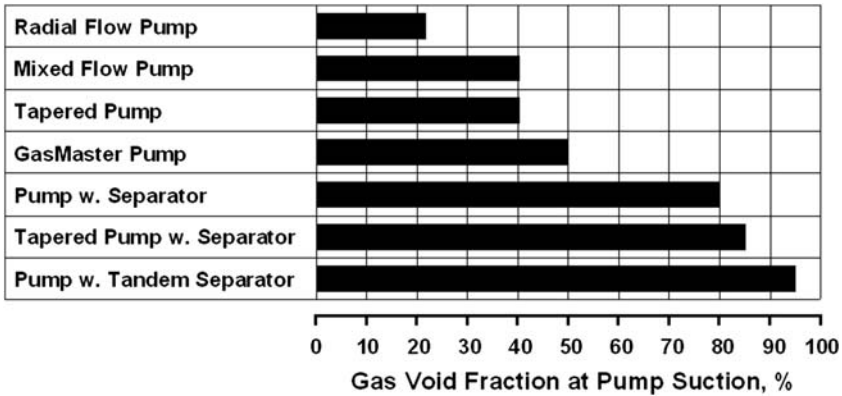


FIGURE 4.25 Comparison of free-gas-handling abilities of electrical submersible pump systems [56].

frequency, the required power is corrected by the next formula to find the actual power demand, based on the affinity laws:

$$BHP = BHP_{base} \left(\frac{f}{f_{base}} \right)^3 \quad (4.31)$$

where

BHP = required power at frequency f , HP

BHP_{base} = required power at frequency f_{base} , HP

f, f_{base} = actual and base frequencies, Hz.

Fig. 4.25, published by a major manufacturer, compares the application ranges of the different possible solutions of handling free gas in the function of the gas void fraction valid at pump suction conditions [56]. Radial flow ESPs can only be used for low void fractions, but pumps with mixed-flow stages perform about equally with tapered pumps. Special pumps (**GasMaster**) can handle up to 50% free gas in the total fluid stream. The use of RGSs very significantly increases the operational range of the ESP installation. The greatest relative amounts of free gas can be handled by a **tandem RGS**, composed of different types of separators: reverse-flow, or a version of the RGS.

4.5 PRODUCTION OF ABRASIVE SOLIDS

4.5.1 Conventional Pumps

4.5.1.1 Introduction

The ESP is a high-speed rotating device whose stationary and rotating parts are **lubricated** by the fluid pumped. As oil well fluids very often contain **solid**

particles, their abrasive action can easily damage the moving parts of the pump. Metal loss caused by **abrasion** and/or **erosion** at critical points in the pump stage or in its different bearings (radial and axial), even if other components of the pump are not attacked, may lead to the **catastrophic failure** of the pump.

In the following, sand problems and their solutions will be shown for the case of the ESP because the great majority of sand problems in the ESP system occur in the centrifugal pump. However, RGSs are also prone to sand damage and special separators must be used in sandy wells. In short, sand production severely shortens the life expectancy of the ESP system; this is why special technical solutions and the use of sophisticated materials are necessary when pumping solid-laden fluids.

Although solids other than **sand**, such as **iron sulfide**, **calcium carbonate**, may cause abrasion in centrifugal pumps, most of the abrasion problems originate from the production of sand (**quartz**, SiO_2) along with well fluids. The characteristic features of sand production from oil wells can be explained by the application of rock mechanics principles and can be summed up as follows:

- sand production usually starts at **high well rates**,
- the amount of sand produced increases after **water breakthrough** to the well occurs,
- sand production increases when well flow rates change.

ESP units are usually installed in wells with high fluid rates and low bottomhole pressures, are often the prime choice in fields under **waterflood** operations, and frequently involve cyclic operations. As these conditions exactly match the main causes of sand production in oil wells listed previously, ESP systems are particularly prone to sand problems.

4.5.1.2 *Characteristics of Abrasive Materials*

The effect of sand-laden fluids flowing through the ESP is the **removal** of metal particles from different parts of the pump. Damage can be classified as

- **erosion** occurring on a metal surface hit by abrasive material particles contained in the fluid,
- **abrasion** caused by an abrasive material and occurring between two metal surfaces because of mechanical wear.

The magnitude of damage caused by these types of wear depends on many factors: the **hardness** of the attacked metal and that of the solid particles and the **concentration**, **size**, **shape**, **toughness**, and particle **size distribution** of the solid particles [57].

The **hardness** of the abrasive particles as compared to that of the attacked metal has a direct effect on the damage made. As no abrasive will cut anything

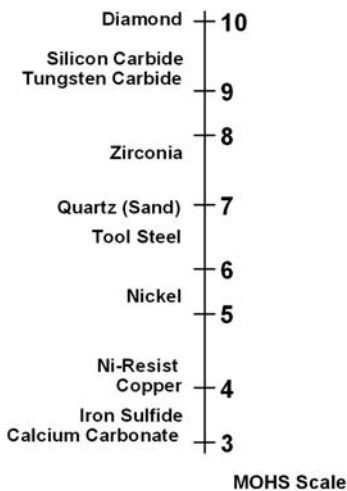


FIGURE 4.26 Approximate material hardness versus the Mohs scale.

harder than itself, metal surfaces in ESPs used for abrasive service must be harder than the abrasive contained in the wellstream. Therefore the proper choice of the base metal or the application of a hard metal coating to endangered surfaces is recommended.

Fig. 4.26 illustrates the relative hardness of several materials in ESP service. Also shown is the **Mohs scale** used for comparing the hardness of minerals, in which a material with a higher Mohs rating will scratch any other material with a lower rating. As seen, sand or **quartz** (SiO_2) is harder than Ni-Resist, the iron alloy most pump stages are made of, or regular steel or nickel. Sand, however, cannot damage the much harder exotic materials such as tungsten carbide or zirconia. The materials of other solids usually present in wellstreams (iron sulfate, calcium carbonate) are much softer than sand; hence, sand is considered the key abrasive substance in oil wells.

Particle **size** and **shape** of the abrasive material is also important and affects the damage done by abrasion or erosion. Abrasion is highest when the size of solid particles is comparable to the **clearances** used in ESPs; sizes between 50 and 250 micrometers are considered to be the most damaging. On the other hand, wear due to **erosion** is proportional to particle size and the square of particle velocity. Rough irregularly shaped solid particles do more damage than round smooth ones of the same size.

Solid **concentration** in the produced fluid has a prime effect on the damage done by erosion and abrasion in ESP equipment. When making estimates of the abrasive production of a well, one must consider that samples taken from the wellstream may contain particles with relatively **low** abrasive properties, such as salt crystals, corrosion products, scale. To eliminate these, samples

should be treated with concentrated **acid** to dissolve the nondestructive materials and leave a representative sample containing mostly sand. **Sand** concentration can be expressed in parts per million (ppm) or its equivalent unit of milligrams per liter; the usual classification of sand production is given in the following.

Concentration, mg/L	Definition
Less than 10	Light
11–50	Moderate
51–200	Heavy
More than 200	Severe

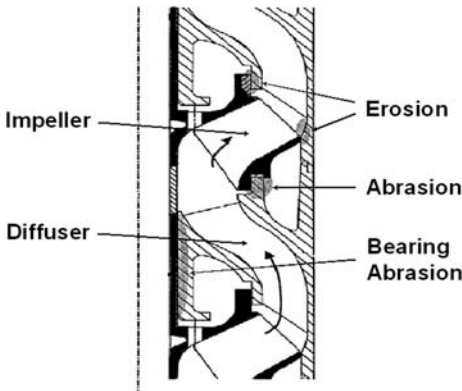
Considering the factors detailed previously, one can define an **aggressiveness index** [58] that shows the relative destructive power of the particular sand sample and allows the comparison of different wellstreams. It is expressed in percentages (between 0% and 100%) and represents the combined effects of particle size, shape, and sand content in the solids produced. Higher **aggressiveness indexes** mean more aggressive conditions; in the absence of a detailed laboratory analysis a value of 60% is usually assumed.

4.5.1.3 Sand Problem Areas

Production of abrasive solids along with well fluids has the greatest effect on the operation of the ESP where abrasive particles move with high local velocities. Abrasive damage caused in ESPs takes different forms and occurs in different parts of the pump [59]. Fig. 4.27 illustrates, in case of a fixed impeller pump, the possible wear points where sand damage can be classified as

- **erosion** in the pump stage
- abrasion in radial bearings (**radial wear**)
- abrasion in thrust washers and thrust bearings (**axial wear**).

FIGURE 4.27 Sand problem areas in a fixed impeller pump.



In addition to pumps, other components of the ESP system are also affected by abrasives in the well fluid. RGSs are especially **susceptible** to abrasive wear because of the great centrifugal forces acting on the solid particles [33]. The solids hitting the separator housing with great speed **remove** the layer of corrosion products from the metal surface that immediately **corrodes** again and the process repeats. This eventually may lead to a complete **cutting** of the separator housing, necessitating the use of special, abrasion-resistant materials.

According to their great importance the different types of abrasive damage occurring in the ESP will be detailed in the following.

4.5.1.3.1 Pump Erosion

Erosion in pump stages is caused by the abrasive solid particles striking the metal surfaces just like in sandblasting. The wear caused is greater for large and rough solid particles than for small and smooth ones. Also, wear is proportional to the square of flow velocity because the destructive potential of the solids is related to their kinetic energy.

Because of the cushioning effect of the liquid phase the suspended solid particles do most of the damage at points where a **change of flow direction** takes place: at the entrances to the diffuser and to the impeller. Another type of erosion occurs in the stagnant fluid areas in the diffuser (below the bottom shroud of the impeller) where trapped abrasive particles rotated by viscous drag forces cause **swirl erosion**. This problem can be mitigated by special swirl-suppressing **ribs**, as described in **Section 3.2.4**.

Although erosion of pump stages can be considerable, it **seldom** leads to failure because the pump usually fails for other reasons long before it is completely **eroded**.

4.5.1.3.2 Abrasion in Radial Bearings

Radial wear is caused by abrasion in the pump's radial bearings. In standard pumps, radial support of the pump shaft is provided by simple **journal bearings**, with the impeller hub acting as the journal and the diffuser bore being the bearing. The journal and bearing materials are identical to those of the stage, with well fluid lubricating the two parts. These bearings run almost for the total length of the shaft and have **fixed** radial clearances, the amount of clearance depending on pump design and machining tolerances. Clearances are usually small but **large enough** for the majority of sand particles to enter the space between the bearing and the journal. Larger particles, after entering the clearance space, are crushed and remove metal from the bearing surfaces, while small ones may be taken by the fluid flow without even touching the bearing surfaces. The amount of wear in radial bearings heavily increases with **increased flow rates** because of the increased amount of sand particles carried by the fluid.

The main effect of radial wear in an ESP is the **growing** of the clearances in bearings and sleeves. This brings about the **loss of radial stability** of the shaft, which starts to **rotate eccentrically** causing the side loads in the bearings to increase and these, in turn, further accelerate wear. Because of the pump shaft's **slenderness** and the high axial loads acting on it, the shaft starts to **buckle**, inducing severe **vibrations** along the shaft. These vibrations can completely destroy the pump in a very short time. As the pump shaft is directly connected to the protector, vibrations are transmitted to the shaft seals protecting the electrical motor from well fluids. Eventual failure of these seals leads to a complete system **breakdown**.

4.5.1.3.3 Abrasion in Thrust Washers

Axial wear is caused by abrasion in the pump stage between the impeller's thrust washers and the diffuser downthrust pads. Abrasive particles caught between the washers result in **worn washers** or even in **metal-to-metal rubbing** of the impeller on the diffuser. Because pumps with fixed impellers (a.k.a. **compression pumps**) completely eliminate the contact between impellers and diffusers, they are almost totally **protected** against the effects of abrasive fluid production.

In pumps with **floating impellers** the axial forces are absorbed by the thrust washers of the impellers that are free to "float" depending on flow conditions. The clearance, therefore, is not fixed but **varies** with the magnitude of the thrust and the viscosity of the fluid. In the recommended capacity range the pump is in the "**downthrust**" condition (see Fig. 3.4) and the clearance thus created is usually too small for the majority of sand particles to enter this space. However, if operating in the "**upthrust**" condition, the clearance between the washers greatly increases allowing large grains of sand to enter, and this can lead to completely **worn-out washers** and/or **abraded stages**.

4.5.1.4 Solutions

Over the years, manufacturers developed many modifications in pump design and introduced the use of different materials for fighting sand damage in ESP equipment. The common background for all designs is the use (on all influenced points in the pump stage) of materials **hard** enough to resist the harmful effects of abrasives. As the most aggressive abrasive material, almost always present in well fluids, is sand, all materials **harder** than sand (see Fig. 4.26) can be used at critical points in the pump stage.

Interestingly enough, soft materials such as **rubber**, can also be successfully used in journal bearings. In this case, because of the resilient nature of the material, sand particles entering the clearance between the rubber bearing and the metal journal, although hitting the rubber, do not remove any material because of the rubber's deflection. In addition to this, sand particles cannot imbed in the soft rubber part. All these result in the sand particles working

their way out of the bearing, thus greatly reducing abrasive wear on the metal journal.

As discussed in the previous section, the **severity** of abrasive damage in submersible pumps increases in the following order:

1. **erosion** in impellers and diffusers,
2. **axial wear** in thrust bearings and upthrust and downthrust washers in floater pumps,
3. **radial wear** in radial (journal) bearings.

Erosion wear in pump stages can be minimized by using **special metals** (Ni-Resist, an alloy containing 18% nickel) for manufacturing impellers and diffusers, instead of the less expensive grey iron, or by using hard surface coatings on endangered areas.

Axial abrasion is present in thrust bearings and the upthrust and downthrust washers of floater pumps. The wear of the washers can be reduced by increasing their surface areas and by the proper selection of their materials. In the ESP unit's main **thrust bearing**, situated in the protector, extremely hard materials, such as tungsten carbide or **ceramics** (usually zirconia), are used for thrust runners and shoes.

As **radial abrasion** is the most significant effect of sand damage in ESPs, the various ways of reducing it are detailed in the following.

4.5.1.4.1 Reduction of Radial Wear

The earliest solution [59] to decrease radial wear was the placement of special radial bearings at regular intervals in the submersible pump. Such bearings contain a special **resilient** (usually rubber) bushing pressed into the diffuser bore where the pump shaft turns, see Fig. 4.28. The rubber bearing is **fluted**, i.e., it has longitudinal grooves on its inside surface where sand particles are

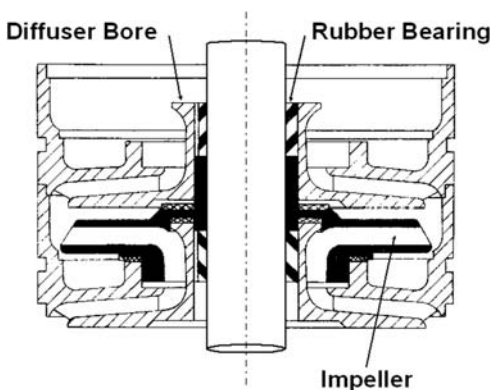
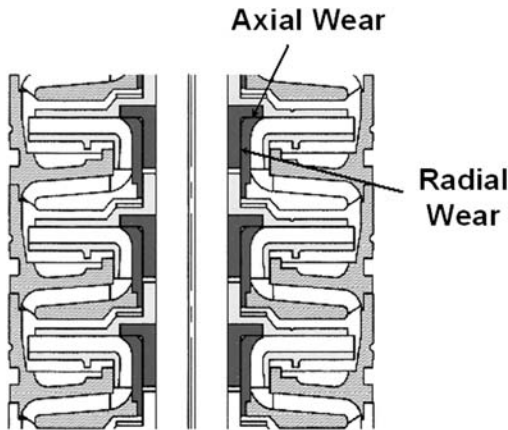


FIGURE 4.28 Construction of a resilient radial bearing.

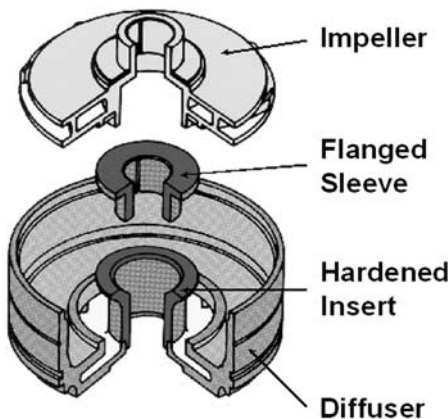
FIGURE 4.29 Floating impeller pump stages modified for abrasive service.



washed into and are continuously removed by the fluid pumped. By fitting these bearings in several stages instead of the standard diffuser bore–impeller hub type bearings, radial abrasion damage can be reduced. The shorter the distance between the special bearings (distributed evenly along the length of the pump shaft), the greater the radial stability of the pump shaft.

Hardening of wearing surfaces to decrease abrasion in radial and thrust bearings was also applied. Fig. 4.29 illustrates several stages of a floating impeller pump modified for abrasive service [60]. Radial and axial stabilization is ensured by the use of special **inserts** made of extremely hard materials so that radial and axial wear can be minimized. A detailed view of the pump stage is given in Fig. 4.30, where a hardened insert is fixed to the diffuser in

FIGURE 4.30 Modified pump stage used for abrasive service.



which a **flanged sleeve** of similar hardness is turned by the pump shaft. The radial and axial wear surfaces are, because of their **hardness**, highly resistant to abrasion damage.

Modular bearing assemblies made of **tungsten carbide** for radial stabilization as well as for upthrust and downthrust protection are available in a major manufacturer's abrasion-resistant pump [61]. Use of these bearings allowed expansion of the pump's thrust range by 70% and increased its operating range and run life.

The great disadvantage of having all pump stages of the abrasion-resistant version is the extremely **high cost**; this is why floater pumps with only a few abrasion-resistant stages (usually situated at the top and bottom of the pump) are also manufactured. Compression pumps with all or some of the pump stages being of the abrasion-resistant type are also available for producing well fluids of extremely high solid contents.

The use of **special** hard materials such as silicon carbide, tungsten carbide, or ceramics greatly increases the abrasion resistance of ESP parts. Early application of these materials in ESP bearings, however, proved to be unsuccessful because they are very **brittle** and are easily fractured if loaded at one point or on a line. Regular journal bearings are mounted into their housings by **press fitting**, and this technique inevitably causes line loadings and an eventual failure of the bearing if very brittle materials are used. This is why journal bearing designs had to be improved to facilitate the utilization of extremely hard materials [62].

The **compliant**-mounted journal bearing illustrated in Fig. 4.31 was developed to use extremely hard bearing materials (silicon carbide, tungsten carbide, ceramics, etc.) that normally cannot be press-fit [63]. The sleeve rotates with the shaft and the inner bushing is the stationary member of the bearing. This is made of a hard material, whereas the outer bushing covering is softer so it can be press-fit into the bearing carrier. There is a small gap between the two bushings that, in conjunction with the resilient rings, acts as a vibration and shock **dampener** and allows the bearing to find its best running

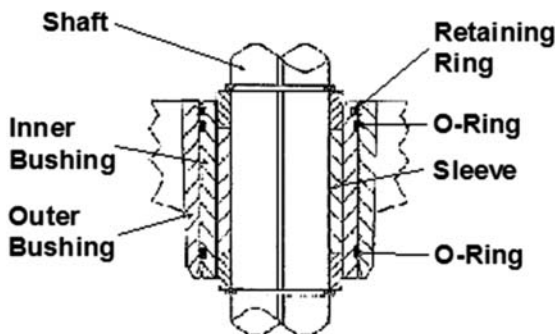


FIGURE 4.31 The structure of a compliant radial bearing.

position; point or line loading of the bearing is thus avoided. Therefore the compliant mounting of journal bearings eliminates the inevitable failures due to the low fracture strength of the extremely hard materials such as silicon carbide or tungsten carbide.

The usual material selections in compliant bearings are zirconia bearings and journals, or zirconia bearings with silicon carbide journals. **Zirconia** is a ceramic material of great hardness that is virtually unaffected by abrasives in the wellstream and can withstand temperatures up to 1,000°F. It has excellent lubrication properties and is not affected by the presence of H₂S or CO₂ gas.

A submersible pump specifically designed for abrasive service is illustrated in Fig. 4.32, in which the head and base bearings (at the two ends of the pump shaft) are of the **compliant** version. As shown in the figure, several similar bearings are distributed along the pump shaft, their **spacing**, because of their moderate cost, may be kept as **close** as possible. Closer spacing results in lower shaft deflections and lower vibrations and a consequent increase in **radial stability**. Radial instability being the prime cause of mechanical failure in ESPs, the run life of ESP equipment in wells producing abrasives can greatly be increased by using compliant bearings.

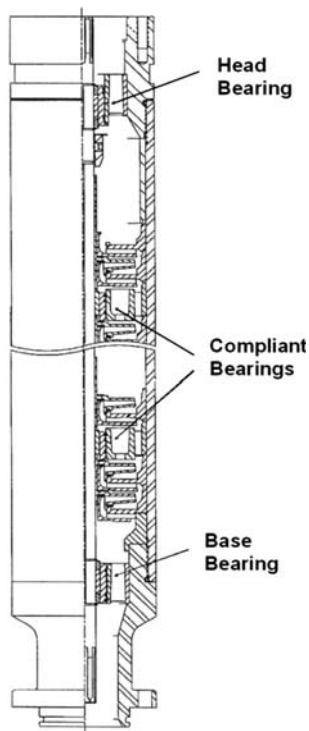


FIGURE 4.32 Abrasion-resistant pump.

4.5.1.5 Preventing Sand Accumulation Above the Pump

The solutions discussed so far reduce the harmful effects of the sand and abrasives that have entered the equipment during normal operations. However, every time the ESP unit is shut down, large amounts of abrasives suspended in well fluids **fall back** from the tubing string. Solid particles accumulating on the top stages of the submersible pump can hinder and may also completely prevent restarting of operations. Restart attempts of a sand-laden, plugged pump can have disastrous results including damaged pump stages, broken shafts, and motor burnouts.

Multilift's patented "SandGuard" [64,65] was developed to eliminate the problems sand fallback may cause in ESP installations after system shutdowns. The device, schematically depicted in Fig. 4.33, is installed just above the pump and takes care of the suspended sand falling out from well fluid during shutdowns. Its body contains an inner tube of the same inside diameter as the tubing used; the tube's inside is covered with a screen (wire mesh) to prevent passage of larger solid particles. The annulus thus formed communicates with the inner tube through several vent holes. At the top of the inner tube a check valve is installed that can completely close fluid flow into the tube.

During normal operation, fluid flow generated by the submersible pump opens the check valve and upward flow takes place in both the inner tube and the device's annulus. As soon as the pump is shut down, fluid draining from

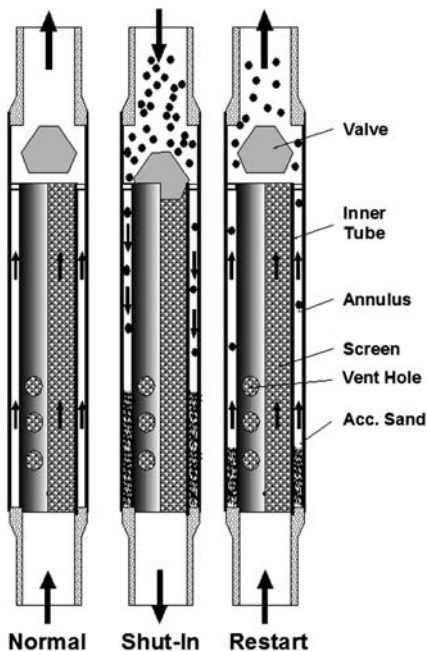


FIGURE 4.33 Schematic operation of the "SandGuard."

the tubing string closes the valve, flow is diverted into the device’s annulus, then through the vent holes into the pump, and finally back to the well bottom, see Fig. 4.33. Thus, sand particles suspended in well fluids will fall back and will accumulate in the device’s annular space because they are filtered out by the wire mesh covering the vents. When the pump is restarted, upward flow opens the check valve and produced fluids flow through the inner tube into the tubing string. The flow in the inner tube induces the flow in the device’s annulus through the vent holes; sand particles are thus carried upward and enter the tubing string to be lifted to the surface. This kind of flush operation occurs every time the ESP is restarted and keeps sand accumulated during shutdowns automatically produced to the surface.

4.5.1.6 Conclusions

Production of sand or other abrasive solid materials along with the well fluid severely shortens the **run life** of a regular ESP equipment, the most affected component being the centrifugal pump. The ESP industry, however, offers different solutions to combat the harmful effects of sand in the produced fluid. Thanks to special equipment components (mainly pumps), the efficiency of ESP installations in sand-producing wells can be maintained at levels competitive with other types of artificial lift.

As a guide to the **selection** of the proper ESP for abrasive service, Fig. 4.34 is presented [58]. The application range of the different pump types is shown in the function of the sand concentration and the **aggressiveness index** introduced previously. If a detailed sand analysis is not available, an aggressiveness index of 60% is assumed.

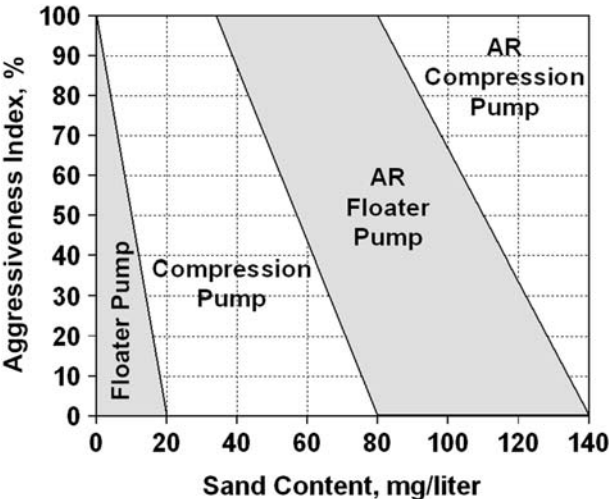


FIGURE 4.34 Pump selection chart for abrasive service. AR, abrasion-resistant.

Depending on the **severity** of abrasiveness, the following general rules apply to the selection of the proper ESP.

- Floating impeller pumps (**floaters** pumps) are generally not recommended if sand production is present.
- Under mildly abrasive conditions, **compression-type** pumps with fixed impellers and without any special features can be used.
- Under aggressive well conditions, abrasion-resistant (a.k.a. radially stabilized) **floating** pumps can be applied.
- Extremely aggressive conditions require the use of abrasion-resistant (a.k.a. radially stabilized) **compression-type** pumps.

Abrasion cannot be managed by the selection of abrasion-resistant pumps alone because trouble-free operations require obeying several basic **rules**. System **startup** in abrasive wells should be specially conducted to help the initial production of sand that has settled above the ESP unit during the shutdown period. Use of a “sacrificial” pump to clean out the well is also advised. Floating impeller pumps should always be operated in the “**down-thrust**” region, **screens** or **filters** must be used, etc.

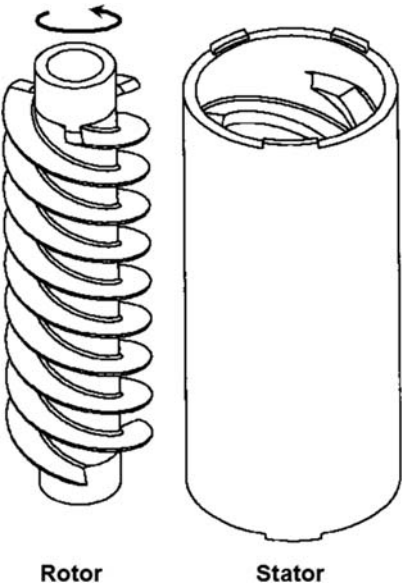
4.5.2 Special Pumps for Abrasive Handling

The use of new materials and technologies in pump stage manufacturing resulted in special pumps developed for demanding applications. The “**WR2**” (Wear Resistant Wide Range) pump stage from Borets [66] is designed for harsh environments including high gas and abrasive contents. The mixed-flow pump stage is manufactured from special materials having a hardness close to that of tungsten carbide using **powdered metal** metallurgy and thus has extreme abrasion resistance. Because of its geometry, the pump can reach efficiencies up to 75% and handles a maximum free-gas volume of 55% without the need of a gas separator. Its extended operating range (three models cover the liquid rate range from 125 to 1,900 bpd) makes it a good choice for wells with changing inflow conditions. The pumps can be used at extremely high pumping speeds of up to 6,000 RPM and can be driven by permanent magnet motors (PMMs).

A completely new kind of helicoaxial pump [67] called “**V-Pump**” was developed for producing liquids with high concentrations of abrasives, viscous and/or two-phase fluids. The pump’s operational mechanism is different from that of the usual centrifugal pumps, although it can be driven by conventional ESP motors in a specified direction only. Its main components are shown in Fig. 4.35:

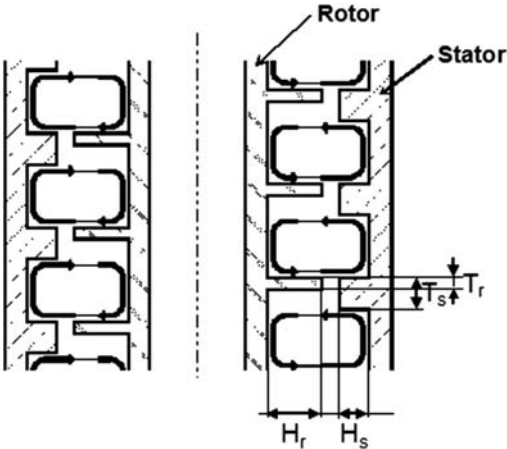
- the **rotor**, having three helical axial **blades** on the **outside** of its hub, phased by 120 degrees around the perimeter,
- the **stator** that has similar three helical axial **vanes** on its **inside**, progressing in the opposite direction to that of the rotor.

FIGURE 4.35 Main components of the V-Pump.



The pitch of the rotor’s helical blades and that of the stator’s vanes is identical, and the angle of the blades/vanes is about 70 degrees in respect to the axis’ normal plane. A vertical section of the pump given in Fig. 4.36 shows some additional important geometrical conditions. The radial height of rotor blades, H_r , is greater than the radial height of the stator vanes, H_s , their usual

FIGURE 4.36 Fluid flow inside a V-Pump.



ratio is 4.2:1. The thickness of the blades/vanes is also different, and the stator vane thickness, T_s , is greater than the rotor blade thickness, T_r , to increase the pump's resistance to erosion caused by abrasives.

There is a radial **gap** between the tips of the rotor's and the stator's vanes. This gap is normally much **greater** than the usual working clearance in rotary machinery and it has a very important function in the pump's operation. As the rotor is turned by a motor driving the pump, the produced fluid moves upward in the rotor **channels** on a spiral path. Fluid movement in the rotor channels creates radial and tangential flow velocity components as indicated by the arrows in Fig. 4.36, and these provide the necessary **liquid seal** across the radial gap between the vanes of the rotor and the stator. This sealing action was so **effective** that increasing the gap from the normal 1.28 mm up to 5 mm does not significantly affect the operation and the efficiency of the pump. If used for pumping highly viscous liquids, liquids with high-free-gas content, or fluids containing high concentrations of abrasive materials, this gap can be increased in the range of 5–10 mm. Experimental data have proved that the pump operates without significant decrease in efficiency even at such extremely high clearances between its rotor and stator.

Currently one manufacturer offers this pump [68] in two sizes: 400 series for $5\frac{1}{2}'$ casing size and 538 series for 7' casing. The performance curves of the 538 series pump are given in Fig. 4.37 at 60 Hz operation; the pump has a relatively wide operating range from 900 to 4,500 bpd. It can be used with any make of ESP equipment and is recommended for harsh production conditions: high-free-gas contents up to 75%, highly abrasive environments, and pumping highly viscous liquids.

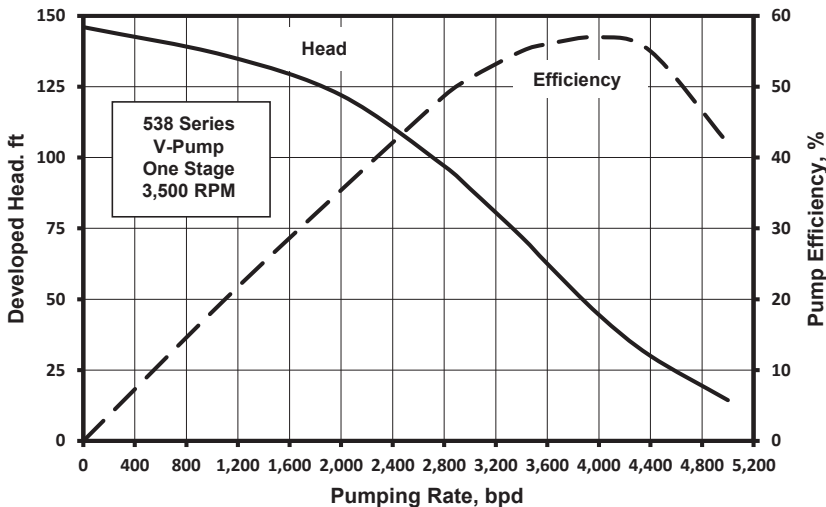


FIGURE 4.37 Performance curves of a V-Pump.

The total number of V-Pump installations by the end of 2016 was 68, with 22 units still running. The main conclusions from flow loop and field tests are the following, according to [69].

- The pump works exceptionally well in **abrasive** conditions without much erosion damage. Sand contents of up to 26% are easily handled.
- High-**viscosity** liquids (up to 13,000 cP) can be pumped, actually the name “V-Pump” came from the pump’s tolerance to viscosity.
- The pump does not **gas-lock** and can develop positive head even when most of the produced fluid is gas.

4.6 HIGH WELL TEMPERATURES

4.6.1 Temperature Problems and Solutions

Well temperature has always been a **limiting factor** in the application of ESP units; standard ESP equipment can be applied to a maximum ambient temperature of approximately 240°F. Above this limit, the performance of average ESP components quickly **deteriorates** and an eventual failure can be expected. As the depths of today’s oil wells are increasingly higher where extreme well temperatures are common, the latest developments aim to provide new **materials** and designs for ensuring proper operation of ESP installations at higher temperatures.

The main **effects** of high temperatures on the performance of submersible equipment are diverse and can be classified as in the following:

- Elastomers used in ESP equipment for sealing, etc. are **weakened** and their service life is **shortened** accordingly, decreasing run lives.
- The **dielectric properties** of insulation materials deteriorate and motor or cable insulations may lose their dielectric strength causing burnouts. According to the **Arrhenius rule**, the dielectric life of insulators is reduced by one-half for each 10°C (18°F) of temperature rise.
- The electrical resistance of the conductor in the submersible cable increases causing an increased **power loss** in the cable.
- The **viscosity** of motor oil decreases and the load-carrying capacity of the main thrust bearing, usually situated in the protector section, decreases.
- Owing to the different thermal expansion of dissimilar metals, mechanical failures in rotating machinery can occur.
- **Scale** formation on the inside and/or outside surfaces of ESP equipment becomes more pronounced.

With regard to **operating temperature**, the most critical component of the ESP unit is the submersible **motor** because it not only operates under the ambient well temperature but also the large amount of heat generated during

operation raises its inside temperature. Overheating of ESP motors, in addition to extreme well temperatures, may be caused by

- motor **overload**,
- **insufficient** cooling caused by a low liquid flow velocity past the motor, see Fig. 3.25,
- production of a fluid with unexpectedly low **specific heat**, providing an insufficient cooling effect,
- **scale deposition** on the outside of the motor severely reducing the amount of heat dissipated to the well fluid,
- the presence of **harmonics** in input power voltage, usually associated with those VSDs providing quasi-sinusoidal waveforms.

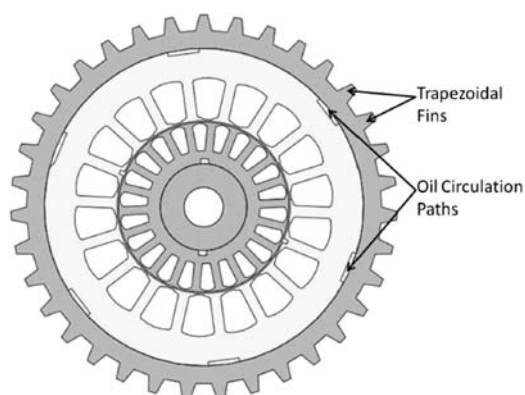
To limit the temperature rise and to increase the **service life** of standard ESP motors, the general solution is motor **derating** when a motor with a higher than necessary rating is chosen. As discussed in **Section 3.3.1.4.2**, by decreasing the horsepower load (the ratio of actual power to the rated power) the heat generated in the motor can be decreased so that the operating temperature falls below the motor's rated temperature (see Fig. 3.27). Liquid **flow velocity** past the motor should also be checked and if it is not sufficient, it can be increased by using a larger-diameter motor or a shroud attached to the outside of the motor.

The modified **features** of motors developed especially for high-temperature service usually include the following.

- Stator windings are insulated by **epoxy** in the stator slots, instead of the conventional varnish coating, to increase the dissipation of heat to the outside of the motor. Due to the high heat **conductivity** of epoxy, motor winding temperatures of about 80°F or less can be experienced as compared to simple varnished motors [70].
- Rotating clearances are **increased** to provide for the larger thermal expansion of the different metal parts occurring at higher temperatures.
- Special **outside coatings** on ESP components reduce the severity of deposition of scale or other precipitants and prevent the reduction of the cooling effect of the wellstream.

A complex project to decrease the internal temperature of ESP motors [71] revealed that the **largest thermal resistances** against heat transfer occur because of conduction in the stator-housing **gap** and forced convection from the motor **housing** to the wellstream. The gap between the motor housing and the stator laminations is created during manufacturing when the stator laminations are press-fit into the motor housing. Although the size of this gap is only a few thousands of an inch, the motor oil filling this space very much reduces the amount of heat that is radially transmitted into the wellstream. Similarly, the effectiveness of forced heat convection on the outer surface of the motor is quite low.

FIGURE 4.38 Cross section of modified motor [71]. Copyright 2014, Society of Petroleum Engineers Inc., SPE. Reproduced with permission of SPE. Further reproduction prohibited without permission.



Based on CFD modeling and experimental work, two basic modifications [72] were proposed on the motor, as illustrated in Fig. 4.38:

- Six rectangular **slots** with a combined total cross-sectional area of 2–4 sq in are cut at the perimeter of the stator laminations. A separate centrifugal pump forces motor oil to flow through these slots; the type of heat transfer in the stator-housing gap thus becomes forced **convection**. Compared to conduction that exists in conventional motors across the stationary oil occupying the stator-housing gap, the heat generated in the motor is transferred with a much higher efficiency to the outside of the motor by forced convection.
- Trapezoidal **fins** (36 pieces in all) running axially along the outside of the motor housing are created. These very substantially increase the surface **area** of the motor's outside and improve the effectiveness of heat transfer from the motor to the wellstream. Thus the heat generated inside the motor is easily transferred to the surrounding wellstream resulting in a greater cooling effect.

Experiments with modified motors proved that the proposed changes very effectively reduce the temperature rise in operating ESP motors. While conventional motors exhibited temperature rises of 80–100°C, the improved motors had only 60°C increases. The average temperature reduction of 30% can significantly increase the motors' reliability and run life in high-temperature applications.

In addition to the changes in motors, high-temperature applications necessitate several modifications in the protector, the pump, and the cable. The rotating members (protector, pump) should have increased inside **clearances** just like the submersible motor. The seal material providing the best performance at high temperatures is **ethylene propylene diene monomer (EPDM)**,

and all the elastomers in motors, pumps, and protectors must be made of this material. In high-temperature electrical cables the insulation and the jacket are also made of **EPDM** materials.

The improvements discussed so far have highly increased the temperature limits of ESP applications. Today, all manufacturers offer ESP equipment for operating temperatures of 400°F; reliable operation of ESP motors at 500°F in a steam-flooded field was also reported [73].

4.6.2 Geothermal, Steam-Assisted Gravity Drainage Applications

The use of ESP equipment is getting increasingly popular outside the application in producing conventional oil wells. The most important fields of these novel applications are **geothermal** energy production, tertiary oil well production by steam-assisted gravity drainage, and steam flooding. All such uses require specialized ESP systems that are capable of withstanding exceptionally high operating **temperatures**, in some cases up to 500°F. Other conditions are also extremely harsh in those wells, including (1) exceptionally high liquid flow rates, (2) great well depths, (3) high salinity, and (4) high concentrations of abrasive materials.

Because all conditions just described tend to decrease the reliability and shorten the run life of ESP equipment, manufacturers took great efforts to come up with solutions. Today, most major manufacturers offer ESP components that can work under the harsh conditions of unconventional applications. In the following the main features of the REDA “**HotlineSA3**” high-temperature ESP system [74] are described, as an example, but other manufacturers also have similar high-temperature equipment [75].

The worst operating conditions in high-temperature applications exist at the ESP motor whose temperature gets much higher than that of the other equipment components because of the heat generated by the motor. To increase the life of the motor under such conditions an **integrated** motor-protector unit was developed that includes the following modifications.

- The **integrated** motor-protector system is sealed and prefilled with a special oil, eliminating field servicing at installation.
- At the top of the unit, four shaft **seals** with **metal** bellows are provided to prevent motor oil contamination.
- A high-load-capacity **ceramic** thrust bearing takes up axial loads.
- Connection of the motor lead extension cable is through a special high-temperature plug-in pothead with dual seals.
- The motor section contains special **insulation** of the stator.
- Below the motor a downhole **monitoring** system exists with a motor internal temperature transducer and fluid pressure and temperature gauges.
- At the bottom of the integrated motor-protector unit a **compensator** with metal bellows is installed to take up expansion/contraction of motor oil.

All these important changes made it possible to increase the allowed motor internal temperature to 572°F and the allowed wellbore temperature to 482°F. The 562 series motor produces 257 HP power at 60 Hz, and the ESP system's maximum fluid capacity is 54,000 bpd. In addition to improvements in the motor, other components of the ESP system were also revised. Pumps are today designed using CFD techniques and have wide vane openings to improve gas and abrasives handling.

4.7 VARIABLE-FREQUENCY OPERATION

In the previous chapters, it was often pointed out that the **conventional** ESP system running at a **constant** speed is very **inflexible**, when compared to other types of artificial lift. The reason is that operating an ESP outside its quite narrow recommended liquid rate range not only reduces the system's efficiency but also can lead to early equipment **failures**. This is why the proper knowledge of well inflow parameters is an absolute prerequisite for a proper design and why the use of improperly assumed well performance data results in bad designs. But even in cases with accurate data and a perfect installation design, the inevitable changes in well inflow parameters (formation pressure, fluid rates, etc.) with time can lead to quick deterioration of the operating conditions and an eventual system breakdown.

These problems, along with some other characteristic disadvantages of conventional ESP installations, are eliminated if the submersible pump were driven with widely **variable** speeds. Because the submersible motor drives the centrifugal pump directly, pump speed is simply controlled by changing the motor's rotational speed. For a given motor construction, however, motor speed is a direct function of the **frequency** of the alternating current (AC), so a proper regulation of the power supply's frequency achieves the necessary effect. This is the background of the application of variable-frequency power operations in artificial lifting by ESP systems, the two versions of which are **VSDs** and **variable-frequency generators** (VFGs).

4.7.1 Pump Performance at Variable Speeds

As the submersible motor's speed is directly proportional to the frequency of the AC supply, the **affinity laws** describing the operation of the ESP and introduced in **Chapter 2** can be expressed in the function of the electrical frequency:

$$Q_2 = Q_1 \left(\frac{f_2}{f_1} \right) \quad (4.32)$$

$$H_2 = H_1 \left(\frac{f_2}{f_1} \right)^2 \quad (4.33)$$

$$BHP_2 = BHP_1 \left(\frac{f_2}{f_1} \right)^3 \quad (4.34)$$

where

f_2, f_1 = AC frequencies, Hz

Q_1, Q_2 = pumping rates at f_1 and f_2 Hz, bpd

H_1, H_2 = developed heads at f_1 and f_2 Hz, ft

BHP_1, BHP_2 = required brake horsepower at f_1 and f_2 Hz, HP.

These formulas enable one to construct pump performance curves for any electrical frequency if the standard performance curve at $f = 60$ Hz is known. Fig. 4.39 displays the head performance curves of one stage of an SN3600 submersible pump at different frequencies. The three dashed lines represent the loci of

1. the **lower** limits of the recommended liquid rates,
2. the **BEPs**,
3. the **upper** limits of the recommended liquid rates.

The effect of frequency increase is easy to see that the higher the pump's speed, the broader the liquid rate range is where the pump can be applied; also, the BEPs move to higher rates. Decreasing pump speeds, on the other hand, cause liquid rate ranges to shrink. If this pump is driven by a 60-Hz electrical supply, its recommended range is between the rates of 2,400 and 4,400 bpd. It

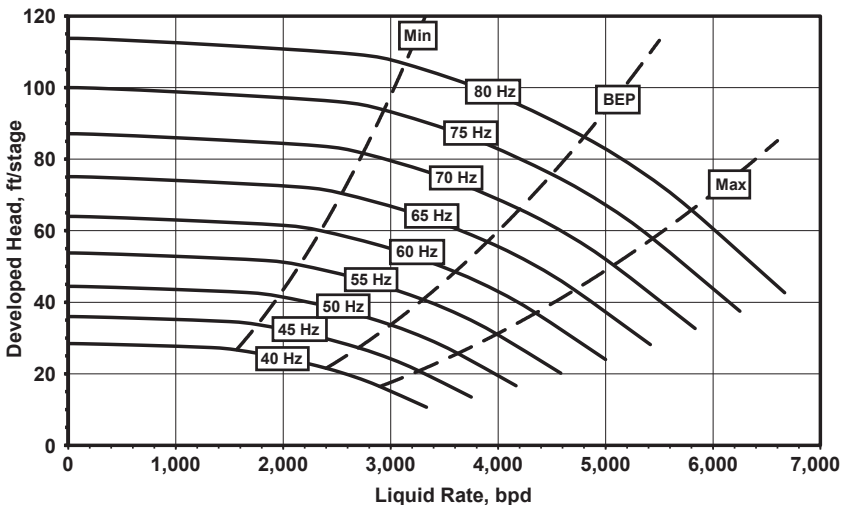


FIGURE 4.39 Head performance curves of an SN3600 pump at different frequencies. *BEP*, best efficiency point.

can easily be seen that by regulating the electrical frequency from 40 to 80 Hz, the pump can cover a much wider range of flow rates, approximately from 1,600 to 5,800 bpd. This extended operational range gives the much-needed **flexibility** for the designer/operator and makes it possible to easily compensate for any uncertainties or changes that may occur in the well's inflow parameters.

The brake horsepower required to drive the pump also changes with the electrical frequency and the motor speed. Fig. 4.40 presents the required powers in function of the pumping rate and the electrical frequency. As seen, increase in pumping speed very substantially increases the power needed to drive the pump.

The **efficiency** of the same pump driven at different speeds is shown in Fig. 4.41. If compared to the constant-frequency (60 Hz) operation, much **higher** efficiencies can be attained at lower or higher liquid rates by changing the driving frequency. The pump's **optimum** efficiency (values close to the BEP) can thus be maintained over the whole extended operational range. This means lower energy requirements for those pumping rates outside of the recommended range are available at 60 Hz operation.

There are two solutions available for using variable-frequency power in ESP systems:

- VSDs utilize standard three-phase electrical power from a utility source and control their output frequency electronically,

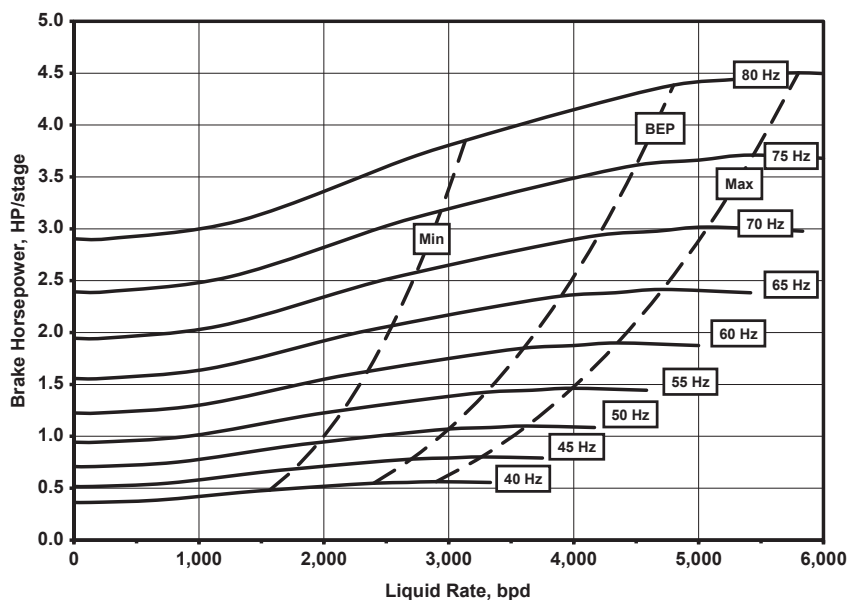


FIGURE 4.40 Brake horsepower performance curves of an SN3600 pump at different frequencies. BEP, best efficiency point.

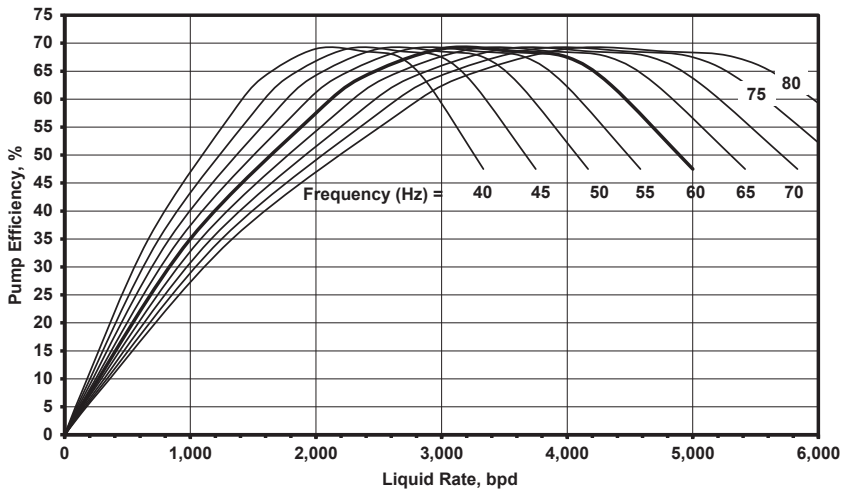


FIGURE 4.41 Pump efficiencies of an SN3600 pump at different frequencies.

- VFGs generate electrical power on the wellsite using some kind of an engine and they control the output frequency by regulating the speed of the engine.

4.7.2 Variable-Speed Drives

VSDs have many advantageous features and have been used on an always-increasing number of wells since their introduction in 1977 [76]. The different available VSD types and their operational features will be described in this section.

The VSDs presented in the following can be used with normal induction motors (IMs) or with PMMs. The main difference is that IMs are simply connected to the VSD but PMMs also necessitate the use of their special motor controllers, discussed in **Section 3.3.2.2**. Normally the PMM controller is an integral part of the VSD made for PMM applications; such devices are simply called VSDs.

In oil fields with ESP installations, electrical power supply may come

- directly from a **utility** company,
- from the field's power **generators**, or
- from a **dedicated** generator at the wellsite.

In either case, primary supply voltage is usually quite high and the required surface voltages should be individually adjusted on each well. If a **VSD unit** is also used, the general arrangement at the wellsite follows the schematic presented in Fig. 4.42. In this power flow, the VSD unit provides the required

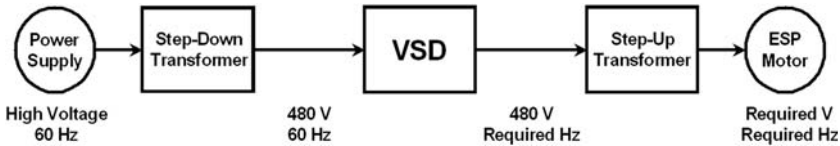


FIGURE 4.42 Typical electrical power arrangement of an ESP well. VSD, variable-speed drive.

frequency and step-down and step-up transformers ensure that the required voltage is available at the wellhead.

4.7.2.1 Constructional Details

To optimize the operating conditions of electrical motors, the use of AC power supplies with variable frequencies is a widely used method in many industries. Such power supplies come in two versions: voltage or current source inverters. **Voltage source inverters** are commonly used in the ESP industry; they control the VSD's output **voltage** and the current fluctuates according to the load on the unit.

The VSD unit's task is to convert the input frequency (usually 60 Hz) into any frequency in its operating range. All VSDs [77] contain the following three basic components (see Fig. 4.43):

1. the **rectifier** section converts the 60 Hz AC voltage and current into a direct current (DC) voltage and current,
2. the DC **control** section provides a smooth DC waveform to the next section, and
3. the **inverter** section converts the DC voltage back to an AC voltage at a selected frequency.

4.7.2.1.1 The Rectifier

The simplest rectifiers use diodes to convert AC to DC by allowing current flow in one direction only. The originally used **six-diode bridge** configuration (see Fig. 4.43) creates a pulsating DC output presented in Fig. 4.44. As seen, the DC voltage has six pulses for a complete cycle; hence, the name of this

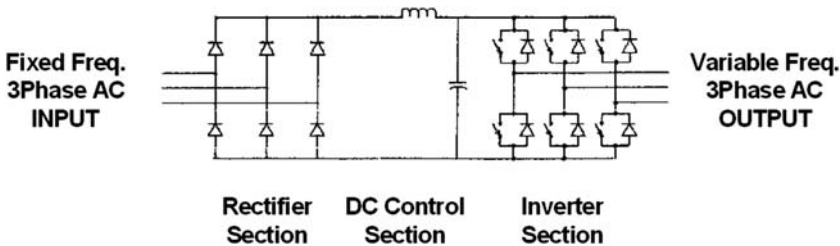


FIGURE 4.43 Main components of a variable-speed drive unit. AC, alternating current; DC, direct current.

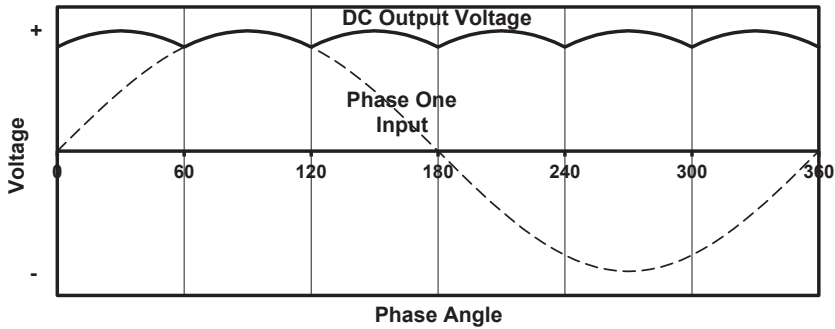


FIGURE 4.44 Output direct current (DC) waveform from a “6-pulse” rectifier.

rectifier type is “**six-pulse**.” There are rectifiers producing more pulses such as 12, 18, 24 and the higher the pulse number, the **smoother** the DC output. Smoother DC waveforms have the advantage of minimizing the amount of **harmonics**, i.e., deviations from a pure sine wave. The average total harmonic distortion (THD) for different pulse numbers changes as follows:

Rectifier Type	Average THD (%)
6-Pulse	25
12-Pulse	11
18-Pulse	5

As seen, an 18-pulse rectifier basically eliminates all problems associated with harmonics and provides an efficient use of electrical power. Some manufacturers use silicon controlled rectifiers but the use of diodes provides better performance: much lower amount of harmonics and uniform, high power factor at all speeds.

4.7.2.1.2 The Direct Current Control Section

The function of this section (also called the DC bus) is to make the DC waveform as **smooth** as possible so that the next section (the inverter) receives a smooth DC voltage. The components used for this purpose are **capacitors** and **inductors**. Capacitors store energy and smooth out the pulses in the DC waveform whenever the rectifier voltage drops. Inductors, on the other hand, restrict the amount of current that charges the capacitors and thus increase the duration of current flow. The effect of large-enough inductors can even make the current continuous, thereby effectively reducing the amount of harmonic distortions.

4.7.2.1.3 The Inverter

The inverter section of the VSD converts the DC voltage produced by the control section into an AC voltage at the selected frequency. It contains

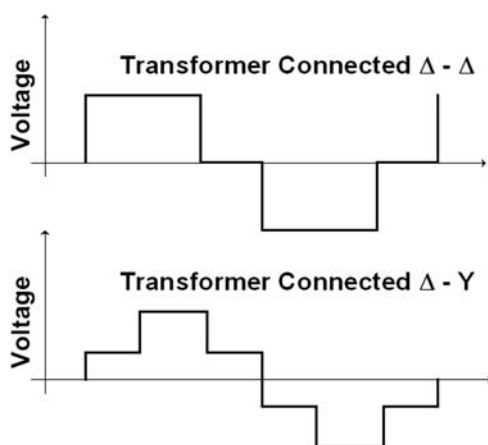
switchable power electronic devices, normally bipolar Darlington transistors, but the latest units use insulated gate bipolar transistors (IGBTs) with very high switching speeds. These devices contain a controllable switch and the output AC frequency is adjusted by the **speed** of switching. The output voltage is controlled according to the desired frequency so that the ratio of voltage over frequency is kept **constant**; this ensures that the motor connected to the VSD unit develops a constant torque at all frequencies.

4.7.2.2 Available Variable-Speed Drive Types

4.7.2.2.1 “Six-Step” Variable-Speed Drive

The first VSD units manufactured in the late 1970s used the **Darlington** transistors in their inverters to create the AC waveform and produced a very crude approximation of a sine wave. They control the output frequency only and the voltage necessary for the ESP unit must be reached using step-up transformers. The output waveform depends on the configuration of the transformer, as shown in Fig. 4.45, and the drive is named for the characteristic “**six-step**” shape of the Δ -Y connection.

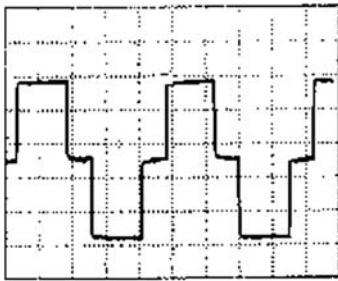
FIGURE 4.45 Voltage outputs from a “six-step” variable-speed drive unit.



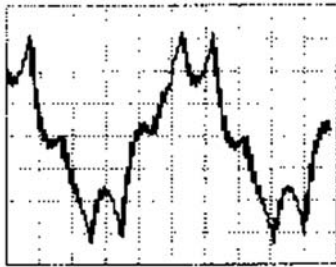
Although used for more than 35 years, “**six-step**” VSDs are almost completely replaced by more advanced units. Their main advantages are the low investment and maintenance costs; the biggest disadvantage is the very **crude** AC voltage waveform. As shown in Fig. 4.46, the output voltage is only a crude approximation of a sine wave, the current being more representative but it contains a large amount of harmonics.

4.7.2.2.2 Pulsed Width Modulation

VSDs operating on the **pulsed width modulation** (PWM) principle are widely used in other industries because of their lower costs. They use inexpensive



Voltage



Current

FIGURE 4.46 Waveforms of a “six-step” variable-speed drive unit.

rectifiers with diodes and inverters with IGBTs. The output voltage of these VSDs takes the form of a series of voltage **pulses**, each of the same magnitude but having a different controlled **width**. The output **frequency** is controlled by adjusting the **number** of pulses per cycle (called the carrier frequency) and the output **voltage** is controlled by the **width** of the pulses.

Fig. 4.47 shows two cases with an identical carrier frequency and thus with the same output frequency but with different maximum output voltages on the sine waveforms. Thinner pulses (Fig. 4.47A) result in a sine waveform with a lower maximum value, whereas wider pulses (Fig. 4.47B) give higher maximum voltage output. The ratio of the sine wave’s maximum voltage and the voltage of the generated pulses is called the **modulation**, which is about 50% in the first and almost 100% in the second case.

The output voltage and current of a typical PWM unit are shown in Fig. 4.48 depicting the characteristic **chopped-up** voltage waveform and the quite smooth current that very closely approximates a perfect sine wave. If compared to the output of older VSDs (see Fig. 4.46), units working on the PWM principle provide much better waveforms, but they have many disadvantages when used with cables longer than about 100 ft. This is why a sine wave **filter** is usually added to filter out harmonics and to prevent other electrical problems.

FIGURE 4.47 (A,B) Voltage output at different modulations.

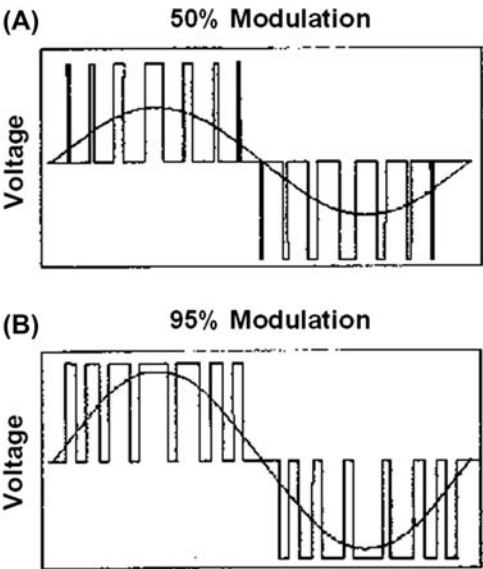
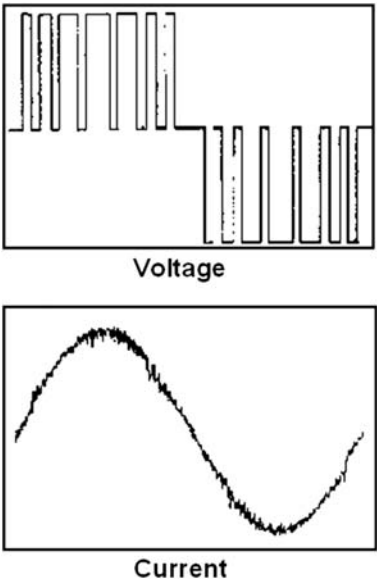


FIGURE 4.48 Outputs from a pulsed width modulation type variable-speed drive.



The “**Electrospeed**” VSD unit of a major manufacturer [78] uses filtered pulse width modulation output that closely imitates a sine wave, thanks to the electronic filters used. In case of filter failure, the filter is disconnected and the unit switches to the six-step operational mode without shutting down the motor.

4.7.2.2.3 Sine Wave Generators

All electrical problems caused by harmonics are eliminated if the waveform of the VSD’s output voltage is a pure **sine** wave. As shown in Fig. 4.49, such units produce near-perfect voltage and current waveforms [79], providing optimum conditions to the power source and the ESP motor at the same time. The VSD units manufactured according to the patented variable sine wave generation principle [80] incorporate electronic components performing functions similar to a sine wave **filter** and produce a cleaner sine wave without the use of an additional filter.

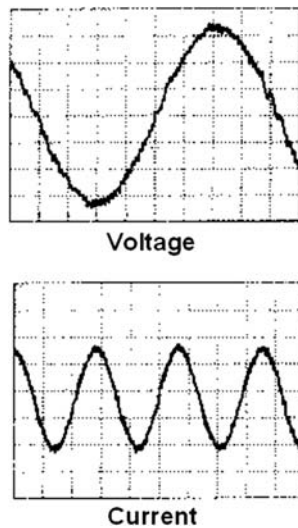


FIGURE 4.49 Outputs from a sine wave generator.

4.7.2.2.4 Vector Control Devices

The “**Vector Control**” technology improves the operation of the drives as well as provides a smooth sine wave output and an enhanced speed or torque control. As discussed in Section 3.3.2.2.2, the operational principle of this drive is based on the behavior of the two components of motor current: the **magnetizing** one and the **torque**-producing one. Magnetizing (or inductive) current produces the magnetic flux in the stator and the rotor, whereas the other current (resistive) produces the torque at the motor’s shaft. Magnetizing

current always **lags** the voltage by 90 degrees, and the torque-producing current is always in phase with the voltage; the motor's **power factor** is defined by these two current components, see Fig. 2.21.

The “Vector Control” drives use microprocessors to control the magnetic flux vector in the motor, which rotates in phase with the current creating it. After decomposing the stator current into its two components the magnetizing current is held at the **minimum** required to induce a magnetic field, whereas torque-producing current is independently adjusted. This feature helps maintain the system's power factor at a **maximum** that very significantly improves the power efficiency of the ESP system. This type of VSD also uses the PWM technology but with a **feedback** loop from the motor; motor position and speed is continuously monitored to control the motor's behavior. Sensorless controllers use indirect (software) techniques to estimate the position and the speed of the rotor. The greatest advantage of such VSDs is the superior torque control provided by controlling the inverter section in response to actual load conditions at the motor.

The “**SpeedStar SWD**” offered by a major manufacturer [81] employs six IGBTs in its inverter and has a constant power factor ($PF = 0.99$) regardless of speed. In addition to the traditional features the unit has a rocking-start function to start **stuck** pumps, when the motor is rotated back and forth to free it from abrasives. The company claims, based on a huge number of cases, that VSDs with sine wave output can extend ESP system life by 50%, as compared to nonfiltered PWM drives [82].

The latest development is the **Active Front End (AFE)** VSD technology that simplifies the construction of the drive and provides excellent speed control [83]. The active rectifier component of the drive uses IGBTs that detect harmonics and create signals with the same amplitude but **opposite** phase; harmonic frequencies are therefore **cancelled** out. THD values are thus very much reduced without the need of higher-pulse (18, 24, and greater) rectifiers, harmonic filters, or phase-shift transformers that other technologies require. The AFE VSD is less complex, smaller, lighter, and cheaper than other units and meets the requirements of the **IEEE 519** standard. It also provides a **constant** power factor of $PF = 0.99$ that has several benefits: significant power savings, lower utility fees, elimination of power factor penalties, etc. The VSDs offered by a leading manufacturer provide all the additional features other units also have [84].

4.7.2.3 Operational Characteristics

The AC electrical power available from utility companies takes the form of an almost perfect **sine** wave. This is also the waveform required for achieving optimum operating conditions for IMs used in ESP units. If a VSD unit is inserted between the power source and the ESP motor, then it is inevitable that the VSD unit's presence affects the operation of both the power source and the ESP unit. Because the DC control section isolates the input and output sides of

the VSD, the **rectifier** affects the power source side only, whereas the **inverter** affects the output side only. Both effects are mainly due to harmonic **distortion**, i.e., deviation of the actual waveforms from a pure sine wave.

Harmonics are voltage or current loads that are due to the deviation from a pure sine wave and have frequencies that are integral multiples of the **fundamental** frequency. In a 60-Hz system, these may be 120, 180 Hz, etc. and are termed the second-order, third-order, etc. harmonics. The combined amount of harmonics is measured by the **THD**, which is defined as the ratio of the sums of the total harmonic amplitudes and the fundamental amplitude in a cycle.

The harmonics affecting the power source are produced by the VSD's **rectifier** that constantly turns the diodes on and off to rectify the AC power. The amount of harmonics can be reduced by rectifiers using more pulses than the “**six-pulse**” rectifier. In this case, DC waveforms smoother than those given in Fig. 4.44 are produced. The most adverse effects of these harmonics on the power system are (1) low power factors, (2) increased load, and (3) overheating of cables and other equipment. Of the VSD types, “**six-step**” units have the worst THD values in the range of 25%–70%, while sine wave generators produce a THD of almost zero [85].

On the output side of the VSD unit, the **converter** produces the harmonics reaching the ESP system. Harmonic distortion of the electrical voltage and current has different effects:

- High THD values of the voltage waveform mainly affect the electrical cable.
- High THD of the current results in
 - overheating of the motor
 - higher currents
 - lower motor speeds.

“**Six-step**” units, again, generate high THD values and the best solution, as before, is the use of sine wave generators.

4.7.3 Variable-Frequency Generators

In oil fields with no or unreliable electrical power supply and/or in mobile applications, the use of electrical generators on the wellsite is common. A newly developed system [86] by a major manufacturer called the **VFG** provides a cost-effective supply of variable-frequency power to ESP installations.

The **VFG** consists of the following major parts:

- a Diesel **engine** providing the necessary continuous mechanical power in a wide operational speed range,
- a three-phase AC **generator** with a widely varying output voltage and frequency range,
- a special electronic motor **controller**.

The unit's power source is the diesel engine that supplies mechanical power at variable speeds regulated by the VFG unit's controller. The engine is directly connected to the AC generator and turns the generator's rotor. Three-phase electricity is induced in the generator's stator the frequency of which is proportional to the diesel engine's speed. The generated three-phase electrical power is used to drive the ESP motor. All components of the VFG unit are mounted on a self-contained skid unit and require significantly less space than a conventional VSD system, thanks to the elimination of several bulky parts such as transformers and filters.

One of the greatest advantages of the described system lies in its output waveform, which is a **perfect** sine wave. Thus, higher harmonics inevitably produced by the operation of VSDs are completely **eliminated** at any output frequency or power. Harmonics, as detailed previously, are responsible for power losses as well as for overheating of the ESP motor and reduce the operating life of the motor lead extension, the electrical cable, etc. Pure sinusoidal power, in contrast, provides ideal operating conditions to the motor and other components and may significantly extend their run lives.

A case study [87] reports a motor current reduction of 4.5%, as compared to the application of a VSD unit in a well producing about 19,000 bpd liquid. The VFG unit had a frequency range from 30 to 60 Hz and was driven by an 855-HP diesel engine running at speeds between 900 and 1,800 RPM.

4.7.4 Interaction of VSD/VFG and ESP Units

4.7.4.1 Variable-Frequency Operation of Electrical Motors

Before the discussion of the interaction of variable-frequency power units with the ESP system, the behavior of ESP motors in a variable AC frequency environment should be presented. The following discussion applies both to induction-type **asynchronous** and to permanent magnet **synchronous** motors because their stators (1) are basically identical and (2) behave similarly under power.

First investigate the normal (nameplate) conditions of the electrical motor. To develop the maximum power the motor maintains the maximum possible magnetic **flux** in its stator windings at design conditions: design frequency and rated voltage. The rated current running through the motor's stator windings makes the stator's iron core magnetically "**saturated**" and the windings become as strong magnets as possible. Increasing the voltage at the terminals above the nameplate value would increase motor current but the strength of the magnetic field cannot increase above the saturated value valid at the rated voltage. This is the reason why motor current must be held **constant** at its nameplate value to keep the stator in a saturated state in all conditions. Limiting the motor current is necessary for several reasons. Motors are rated for temperature rise that is caused by the current, so increasing the current

above the nameplate value would overheat and damage the motor. Also, a properly designed motor's nameplate current ensures full saturation of the stator windings and optimum operating conditions.

The motor drives the submersible pump and it should provide the energy required by the **pump** to produce the required liquid rate to the surface. Fluid lifting requires that a given **torque** be provided for the pump to operate and that the torque must be available always, independent of the motor's speed or the frequency of the electricity. The torque developed by the motor at nameplate conditions, T_{np} , can easily be calculated from the motor's mechanical power and its rotational speed:

$$T_{np} = \frac{5252 P_{np}}{N} \quad (4.35)$$

where

T_{np} = motor nameplate torque, ft-lbs

P_{np} = nameplate mechanical power, HP

N = motor rated speed, RPM.

Now if the motor is used at different electrical frequencies, the torque developed by it must be kept **constant** for all frequencies to drive the centrifugal pump. However, as the frequency is changed, the motor develops different **mechanical** powers that can be found by solving the previous formula for power, P_m , based on the value of T_{np} for all frequencies used. From the calculated mechanical powers the motor's input **electrical** power requirement, P_e , is found after correction for the mechanical efficiency of the motor. This electrical power is expressed in kilowatts and is found as given here:

$$P_e = 0.746 \frac{T_{np} N}{5,252 \eta_m} \quad (4.36)$$

where

P_e = motor electrical power, kW

T_{np} = motor torque, ft-lbs

N = motor speed, RPM

η_m = motor mechanical efficiency, —.

In this formula, motor speed varies proportionally to the input frequency as discussed in **Chapter 2** and can be found from the motor's full-load speed at the nameplate frequency using the next formula:

$$N = N_{np} \left(\frac{f}{f_{np}} \right) \quad (4.37)$$

where

f, f_{np} = AC frequencies, Hz

N, N_{np} = motor speeds at f and f_{np} Hz, RPM.

Next, by assuming that the nameplate current, I_{np} , flows through the motor at **all** frequencies for the reasons discussed earlier, the required motor **voltages** can be found by solving the three-phase electrical power formula (Eq. 2.30):

$$U = \frac{1,000 P_e}{\sqrt{3} I_{np} \cos(\varphi)} \quad (4.38)$$

where

U = required motor voltage, V

P_e = motor electrical power, kW

I_{np} = motor nameplate current, A

$\cos(\varphi)$ = motor full-load power factor, —.

The calculated values represent the voltages providing full **saturation** of the motor's iron core at different driving frequencies and ensure that the motor operates properly. To fully understand these basic principles a sample case is presented using an **induction**-type electrical motor.

Example 4.5

Investigate the operation of an IM, having the parameters given, at different electrical frequencies.

Nameplate power = 84 HP

Nameplate voltage = 968 V

Nameplate current = 55.5 A

Design frequency = 60 Hz

Speed = 3,500 SPM

Power factor = 0.82

Efficiency = 0.83

Solution

For illustration, calculations are detailed for the frequency of 30 Hz only.

The motor speed at 30 Hz operation is found from Eq. (4.37), using the nameplate frequency and speed:

$$N = 3,500 \cdot 30 / 60 = 1,750 \text{ RPM.}$$

The torque of the motor for all frequencies is set equal to that calculated for the nameplate conditions from Eq. (4.35):

$$T_{np} = 5,252 \cdot 84 / 3,500 = 126 \text{ ft} - \text{lbs.}$$

The mechanical power developed by the motor at this frequency is calculated by solving the universally valid Eq. (4.35) for power as follows:

$$P_m = T_{np} N / 5,252 = 126 \text{ 1,750} / 5,252 = 41.9 \text{ HP.}$$

Motor electrical power is found from Eq. (4.36):

$$P_e = 0.746 \text{ 126 1,750} / 5,252 / 0.83 = 37.7 \text{ kW.}$$

The voltage required for the motor to develop this power is, according to Eq. (4.38),

$$U = 1,000 \text{ 37.7} / 1.732 / 55.5 / 0.82 = 478 \text{ V.}$$

Finally, the ratio of the required voltage and the electrical frequency is calculated:

$$U/f = 478/30 = 16 \text{ V/Hz.}$$

Calculation results for other electrical frequencies are presented in Table 4.2.

It can be observed from the calculated results that the required motor voltages divided by the electrical frequencies produce a **constant** value for an **asynchronous** IM. However, similar calculations performed for **synchronous** PMMs produce the same final results. The general conclusion is, therefore, that the **ratio** of motor voltage to driving frequency must be held constant if electrical motors at different electrical frequencies are used. Fulfilling this requirement guarantees that the motor will develop the same **torque** at all

TABLE 4.2 Calculation Results for Example 4.5

Frequency (Hz)	Speed (RPM)	Torque (ft-lbs)	P_m (HP)	P_e (kW)	U (V)	U/f
20	1,167	126	28	25	319	16
30	1,750	126	42	38	479	16
40	2,333	126	56	50	639	16
50	2,917	126	70	63	798	16
60	3,500	126	84	75	968	16
70	4,083	126	98	88	1,117	16
80	4,667	126	112	101	1,277	16
90	5,250	126	126	113	1,437	16
100	5,833	126	140	126	1,596	16

frequencies, which is a basic requirement of driving a centrifugal pump in an oil well. This is the reason why VSD units' output frequencies and voltages satisfy the following relationship:

$$U_2 = U_1 \left(\frac{f_2}{f_1} \right) \quad (4.39)$$

where

f_1, f_2 = AC frequencies, Hz

U_1, U_2 = output voltages at f_1 and f_2 Hz, V, respectively.

The **constant** volts/frequency ratio ensures constant magnetic flux **saturation** in the electrical motor when the ESP motor becomes a **constant-torque, variable-speed** device. It follows from the constant ratio of the input voltage and frequency that motor power is also **linearly** proportional to input frequency [88,89]. This behavior can be observed also from the calculation results in Table 4.2 and can be expressed as given in the following:

$$HP_2 = HP_1 \left(\frac{f_2}{f_1} \right) \quad (4.40)$$

where

f_2, f_1 = AC frequencies, Hz

HP_1, HP_2 = motor powers available at f_1 and f_2 Hz, HP, respectively.

4.7.4.2 Interaction of ESP Motors and VSD/VFG Units

If the power developed by an ESP motor and the power required by the centrifugal pump, both valid at different frequencies, are compared then it is easy to see that motor power is **linear**, while the required pump power is a **cubic** function of the driving frequency [compare Eqs. (4.40) and (4.34)]. This means that the ESP and the motor, if operated at variable electrical frequencies, do not properly match with each other [88–90].

The cooperation of an ESP and an electrical motor at variable frequencies is illustrated by the schematic drawing presented in Fig. 4.50. As motor power changes linearly and pump power according to a cubic function (both as functions of the AC frequency), the motor is overloaded above a given frequency ($f_{\max}$) and cannot be used with the given pump [91]. The allowed maximum frequency can be found from the following formula:

$$f_{\max} = f_1 \sqrt{\frac{HP_1}{BHP_1}} \quad (4.41)$$

where

$f_{\max}$ = maximum allowed frequency, Hz

f_1 = AC frequency, Hz

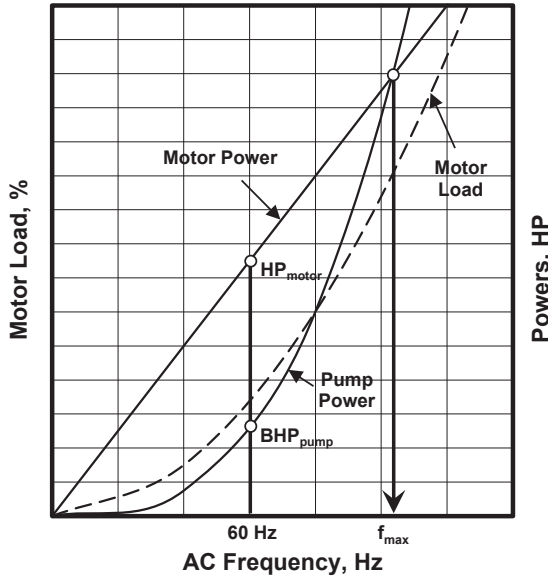


FIGURE 4.50 Pump and motor powers at variable frequencies. AC, alternating current.

HP_I = motor power available at f_I frequency, HP

BHP_I = required pump brake horsepower, HP.

Alternatively, if the required pump brake horsepower, BHP_I , at a given frequency, usually at $f_I = 60$ Hz, is known along with the maximum desired frequency, $f_{\max}$, then the minimum motor rating at 60 Hz should be chosen as

$$HP_{60} = BHP_{60} \left(\frac{f_{\max}}{60} \right)^2 \quad (4.42)$$

where

$f_I, f_{\max}$ = AC frequencies, Hz

BHP_I, BHP_{60} = required pump powers at f_I and 60 Hz, HP, respectively

HP_I, HP_{60} = motor nameplate powers at f_I and 60 Hz, HP.

The **loading** of the electrical motor is shown in a dashed line in Fig. 4.50 and can be calculated at any frequency from the available motor power and the required pump power, both at 60 Hz, as follows:

$$\text{Load} = 100 \frac{BHP_{60}}{HP_{60}} \left(\frac{f}{60} \right)^2 \quad (4.43)$$

where

Load = motor load, %

f = AC frequency, Hz

BHP_{60} = required pump power at 60 Hz, HP

HP_{60} = motor nameplate power at 60 Hz, HP.

As shown in Fig. 4.50, the motor is very lightly loaded at low frequencies, a very unfavorable condition for IMs whose efficiencies diminish under light loads. Hence, to assure favorable motor efficiencies, the operational frequency range should be carefully selected.

When checking the **strength** of the pump shaft, the actual driving frequency must be considered and the shaft rating, given by the manufacturer at 60 Hz operation, must be corrected as given here:

$$HP_{shaft} = HP_{shaft60} \left(\frac{f}{60} \right) \quad (4.44)$$

where

HP_{shaft} = pump shaft rating at frequency f , HP

$HP_{shaft60}$ = pump shaft rating at 60 Hz, HP

f = AC frequency, Hz.

Alternatively, the maximum allowed frequency, $f_{\max}$, which does not overload the pump shaft, is found from

$$f_{\max} = 60 \sqrt{\frac{HP_{shaft60}}{BHP_{60}}} \quad (4.45)$$

where

$f_{\max}$ = maximum allowed frequency, Hz

$HP_{shaft60}$ = pump shaft rating at 60 Hz, HP

BHP_{60} = required pump power at 60 Hz, HP.

Example 4.6

Select the proper ESP motor for an installation using 100 stages of an **SN3600** pump running at a frequency of 49 Hz and producing 3,000 bpd of liquid. At this frequency and the required liquid rate the **SN3600** pump develops 33 ft/stage and requires 1.1 BHP/stage.

Solution

The 100-stage pump requires $100 \times 1.1 = 110$ HP at 49 Hz, which corrected to 60 Hz, using Eq. (4.34), equals

$$BHP_{60} = 110 (60/49)^3 = 202 \text{ HP.}$$

Assume that an RGS is also required and it needs 5 HP at 60 Hz, as found from manufacturer's data. As the separator is also a centrifugal device, its power requirement must be corrected the same way:

$$BHP_{sep\ 49} = 5(49/60)^3 = 3 \text{ HP.}$$

The total power requirement for pumping at the operational frequency of 49 Hz, therefore, equals $110 + 3 = 113 \text{ HP}$.

Assuming that a motor with this power is used, its required power at 60 Hz is found from Eq. (4.40) describing the performance of the ESP motor:

$$HP_{60} = 113 (60/49) = 139 \text{ HP.}$$

Using Table D.2 from **Appendix D**, a Dominator 562 motor is selected with the following parameters:

$$\begin{aligned} \text{Power} &= 150 \text{ HP} \\ \text{Voltage} &= 3,190 \text{ V} \\ \text{Amperage} &= 28.5 \text{ A} \end{aligned}$$

For checking the maximum allowed frequency, Eq. (4.41) is used:

$$f_{\max} = 60 (150/202)^{0.5} = 52 \text{ Hz.}$$

Note that although this value is greater than the required 49 Hz, the VSD unit should never be run above this frequency because the motor then will be overloaded. Choosing a bigger-capacity motor would eliminate this problem.

Actual loading of the motor is found from Eq. (4.43) in which the power required for driving the separator is also included:

$$\text{Load} = 100\ 207/150 (49/60)^2 = 92\%.$$

The pump shaft strength is checked next. Using the shaft limit of 256 HP, found from Table C.1 in **Appendix C**, the shaft capacity at 49 Hz is found from Eq. (4.44):

$$HP_{shaft} = 256 (49/60) = 209 \text{ HP, greater than the required pump HP.}$$

Finally, the motor voltage required to achieve full saturation at 49 Hz is calculated from Eq. (4.37):

$$U_{\text{motor}} = 3,190 (49/60) = 2,605 \text{ Volts.}$$

Calculation results, including the motor loading, are displayed in Fig. 4.51 in the function of AC frequency.

4.7.5 Benefits of Using Variable-Speed Drive/Variable-Frequency Generator Units

The VSD/VFG units perform three distinct functions when connected to an ESP system: (1) by changing the electrical frequency they change the pumping

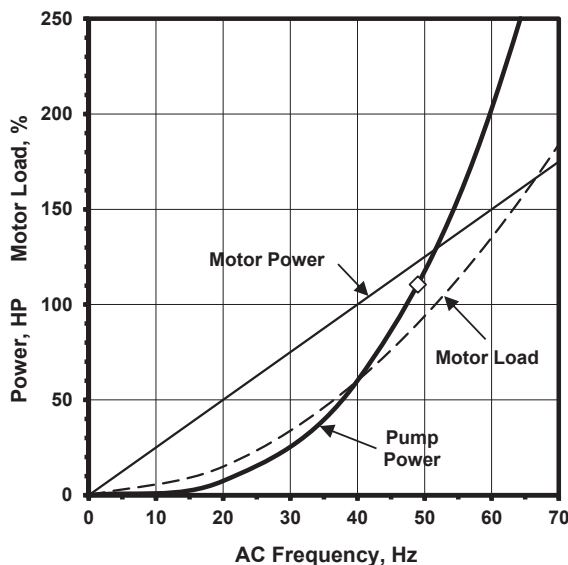


FIGURE 4.51 Variation of motor and pump powers for Example 4.6. AC, alternating current.

rate, (2) they **protect** the downhole equipment from network transients, and (3) allow for a “**soft start**” of the ESP unit. The benefits of these functions are detailed in the following.

- The use of variable frequencies has several advantages if compared to conventional ESP installations with constant motor speeds. The most important general improvement is the wide **flexibility** of the variable frequency ESP system that permits perfect **matching** of the lift capacity of the ESP system and the well’s productivity. This means that, in contrast to fixed-frequency units, great uncertainties or extensive changes in well inflow conditions can easily be accommodated during installation design or later. Thus an installation designed with unreliable well data must not be **choked** or **cycled** to produce the desired liquid rate. Choking involves wasting of electrical power, while cycling, on the other hand, decreases the run life of the ESP equipment. With a variable-frequency unit, these problems are eliminated.
- The VSD also **protects** the downhole motor from power system **disturbances** (voltage transients, lightning strikes, etc.) because the ESP motor is automatically **isolated** from the power source. Although VSDs can be shut down or damaged by such electrical disturbances, their repair is much easier than motor repairs involving pulling of the equipment from the well.
- Variable-frequency units facilitate “**soft starting**” of the ESP installation, which has three major benefits. First, soft starts reduce the startup power

requirements compared to an “**across-the-line**” start. At the same time, they significantly decrease the load on the shafts of the ESP unit; this is especially advantageous when pumping gassy wells or wells producing abrasives. Finally, the reduced startup currents during a soft start can extend the operational **life** of ESP motors.

Additional **benefits** of both kinds of variable-frequency units (VSDs and VFGs) can be summarized in the following [92–95].

- Liquid rates from wells with **small-diameter** casing strings can be increased by running the ESP unit at higher speeds [89].
- The efficiency of the ESP system increases if a high-capacity pump is slowed down to produce a lower rate because the efficiency of larger pumps is higher even at lower speeds.
- The **life** of ESP equipment is increased compared to that of an overloaded unit running at a constant speed because of the lower heating effect and the lower thrust loads.
- Variable-speed ESP units can perform **well testing** operations and help select the optimum equipment for the given well.
- Closed-loop **control** of the installation is possible when motor frequency is varied according to well inflow rate, sensed by downhole instrumentation [94].
- Gas-lock problems are eliminated if the pump is slowed down before a complete **gas lock** develops.

The advantages of **Variable Frequency Generators** (VFGs) over VSD units are given in the following.

- Investment costs can be lower because of the elimination of several system components such as transformers and filters.
- Maintenance requirements and costs of the VFG units can be lower because of their simplicity.
- Because of the **perfect** sine waveform output, heat loading of the motor and the cable is reduced, increasing the reliability and run life of ESP equipment.
- Under similar well conditions, motor current is reduced because no harmonic distortion occurs.
- The VFG unit usually needs less space than a VSD unit; application in offshore conditions is easier.

4.8 OTHER PROBLEMS

Several problems generally occurring in oil wells, such as corrosion, scale, must also be taken care of in ESP operations. Of special concern are **corrosion** problems associated with sour gases when H_2S and/or CO_2 are produced along

with natural gas. Conventional metals used in ESP housing manufacture are prone to sulfide stress corrosion cracking (hydrogen embrittlement) that leads to early failure; use of stainless steel housings is recommended to extend the run life of equipment installed in aggressive environments.

Inorganic **scale** formation in pump stages and on motor outside surfaces causes operational problems: impellers and diffusers may be **plugged** and motors can **overheat**. Inhibitor treatments can handle these problems. Scale deposition on motor housings creates a heat insulation and severely restricts the heat flow from the inside of the motor to wellbore fluids. To eliminate motor overheating due to scaling, a special **paint** is applied to the motor housing that prevents scale precipitation.

The use of ESP equipment in **deviated** and/or horizontal wells can be limited by the maximum buildup rate in the well's trajectory; for a reliable installation, a maximum buildup rate of 6 degrees per 100 ft is allowed. As more horizontal wells with tighter radii are completed, operators need improved ESP systems that can handle this problem. A major manufacturer, first in the industry, developed the "CENesis" tight radius system that can be used in wells with buildup rates as high as 25 degrees per 100 ft [96]. The system uses, instead of the traditional flanged connections of ESP components, a boltless method of connection. The patented connection is twice as strong as the bolted one and can reduce the time of running operations by more than 50%. Other advantages include the possibility to set the pump deeper and to increase the PIP, which decreases the free-gas content at the pump intake.

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Chapter 5

Design of ESP Installations

5.1 INTRODUCTION

Designing an electrical submersible pump (ESP) installation for a well that produces liquid only is a quite straightforward procedure because the operating conditions for the submersible pump are **ideal** and simple hand calculations suffice to describe the pressure conditions in the well. In any other situation, however, the use of a computer is almost a must because of the increased calculation demand. Two production scenarios stand out from those many possible cases: gassy wells and the use of variable speed drive (VSD) units.

Increasing amounts of gas production complicate the selection of ESP equipment in several ways and hand calculations for such cases can only be approximate. Unknown well inflow conditions can be accommodated if a VSD is used to provide the power supply to the ESP motor.

In this chapter, installation designs are detailed for the most common situations encountered in the field. Although hand calculations are presented, the procedures described form the basis of all computerized designs available; therefore a basic understanding of these helps the engineer understand the logic of more advanced computer program packages. The basic design scenarios covered in this chapter are

- pumping a single-phase **liquid** without considerations of motor slip,
- pumping a single-phase **liquid** with proper consideration of the motor's slip,
- producing free **gas** along with the liquid,
- using a **VSD unit** to drive the ESP equipment, and
- using a **permanent magnet motor** (PMM) to drive the submersible pump.

5.2 DATA REQUIREMENTS

For a proper design of an ESP installation, the knowledge of many different data is necessary. Perhaps the most important among them is the reliable information on the well's **productivity** so that the desired fluid rate from the well can be established. The fluid rate is always an input parameter in the design of ESP installations because the selection of the submersible pump (the heart of the system) can only be accomplished in the knowledge of the desired rate. Just remember that different pumps have different recommended application ranges.

Necessary input data can be grouped as given here.

1. Well physical data
 - Casing and liner sizes, weights, and setting depths
 - Tubing size, type, weight, and thread
 - Total well depth
 - Depth of perforations or open hole interval
 - Well deviation survey data
2. Well performance data
 - Tubinghead pressure at the desired rate
 - Casinghead pressure
 - Desired liquid production rate
 - Static bottomhole pressure or static liquid level
 - Flowing bottomhole pressure (*FBHP*) or dynamic liquid level
 - Productivity data (*PI* or $q_{\max}$ for the Vogel model)
 - Producing gas/oil ratio (*GOR*)
 - Producing water-cut or water/oil ratio
 - Bottomhole temperature at desired liquid rate
3. Fluid properties
 - Specific or API gravity of produced oil
 - Specific gravity of water
 - Specific gravity of produced gas
 - Maximum kill fluid density
 - Bubblepoint pressure
 - Viscosity of produced oil
 - pVT data of produced fluids (volume factors, solution *GOR*, etc.).
4. Surface power supply parameters
 - Primary voltage available at the wellsite
 - Frequency of the power supply
 - Available power supply capacity
5. Unusual operating conditions
 - Production of abrasives, especially sand
 - Paraffin deposition
 - Emulsion formation
 - Type and severity of corrosion
 - Extremely high well temperatures

5.3 CONVENTIONAL DESIGN

This section describes the design of a conventional ESP installation with an induction-type electric motor, having the following main features.

- The well produces single-phase liquid only, the **liquid** being either (1) clean oil, (2) water-cut oil, or (3) almost completely water.

- The amount of free gas at the pump suction, if any, is very **low** so that it can be removed easily by a simple **reverse-flow** gas separator. Therefore, no deterioration of submersible pump performance takes place.
- The **viscosity** of the produced liquid is low so that no correction of the standard pump performance curves is necessary.
- The ESP unit is fed with an AC current with a **constant** frequency.

Provided that the simultaneous flow of oil and water does not result in the formation of an emulsion, the water-cut oil can be considered as a single liquid phase. Such flow conditions, as discussed in previous chapters, are **ideal** for the operation of the centrifugal pump and allow the application of a very straightforward design procedure [1–6], the main phases of which are detailed in the following sections.

5.3.1 Well Inflow Calculations

The basic requirement for a proper installation design is the exact information on the well's inflow performance because the selection of the ESP cannot be made without the knowledge of the **liquid rate** and the corresponding **dynamic liquid level**. These parameters are interrelated and can be calculated from the well's **inflow performance curve**. As detailed in **Chapter 2**, either the constant **productivity index (PI)** or the **Vogel** model can be applied, depending on whether free gas is present at the perforations or not. In the first case the value of the *PI* and in the second the value $q_{\max}$ (the well's absolute open flow potential) must be available, preferably from a recent **well test** survey.

The well's *FBHP* is easily calculated from the previous data and the next formula must be used if the constant *PI* equation describes the well inflow:

$$FBHP = SBHP - \frac{q}{PI} \quad (5.1)$$

where

FBHP = flowing bottomhole pressure, psi

SBHP = static bottomhole pressure, psi

q = liquid rate, STB/day, and

PI = productivity index, STB/day/psi.

If well inflow follows the **Vogel** model, then *FBHP* is found from Fig. 2.2.

Based on the *FBHP* at the perforations, the pressure at the pump's **suction** (pump intake pressure, *PIP*) is found from the following formula, using the fluid gradient valid in the annulus below the pump setting depth:

$$PIP = FBHP - (L_{\text{perf}} - L_{\text{set}})grad_l \quad (5.2)$$

where

PIP = pump intake pressure, psi

FBHP = flowing bottomhole pressure, psi

L_{perf} = TVD of perforations, ft
 L_{set} = TVD of pump setting, ft, and
 grad_l = fluid gradient in the annulus, psi/ft.

The knowledge of the *PIP* allows one to check the existence of free gas at the pump's suction. For this purpose, first the solution *GOR* at the pump intake is calculated. In lack of measured pVT data, the bubblepoint pressure correlation of **Standing** [7] can be used:

$$R_s = \gamma_g \left(\frac{PIP}{18 \cdot 10^6} \right)^{1.205} \quad (5.3)$$

where

R_s = solution gas/oil ratio, scf/bbl
 $y = 0.00091 T - 0.0125^\circ \text{API}$
 PIP = pump intake pressure, psi
 T = suction temperature, $^\circ\text{F}$
 γ_g = gas specific gravity, —, and
 API = oil API gravity, —.

By comparing this value to the well's producing *GOR*, it can be established whether there is a free gas phase entering the submersible pump. ESPs can usually handle small amounts of free gas, and the **Turpin** correlation [8] shown in Fig. 4.9, can be used to determine if this is the case. If not, then the installation must be designed with the procedure developed for wells with high gas/liquid ratios (GLRs), to be described in a subsequent section.

Finally, the "in situ" **liquid volumetric rate** to be handled by the ESP is found from the oil and water flow rates:

$$q'_l = q_o B_o + q_w B_w \quad (5.4)$$

where

q'_l = liquid rate to pump, bpd
 q_o = oil volumetric rate, STB/day
 q_w = water volumetric rate, STB/day
 B_o = oil volume factor at pump suction pressure, bbl/STB, and
 B_w = water volume factor at pump suction pressure, bbl/STB.

In the previous formula, the volume factor of water is usually taken as unity, and that of the oil can be obtained from the **Standing** [7] correlation:

$$B_o = 0.972 + 1.47 \cdot 10^{-4} F^{1.175} \quad (5.5)$$

where

B_o = oil volume factor, bbl/STB
 $F = R_s (\gamma_g/\gamma_o)^{0.5} + 1.25 T$

R_s = solution gas/oil ratio, scf/bbl
 γ_o, γ_g = oil, gas specific gravities, —, and
 T = suction temperature, °F.

In addition to these calculations and based on the actual inflow performance data the well's **maximum** possible flow rate can also be established. This enables one to investigate the possibility of other producing scenarios.

5.3.2 Total Dynamic Head Calculations

To determine the required number of pump stages in a later phase of the design procedure, the total **head** to be overcome by the ESP has to be determined. This is called the **total dynamic head, TDH**, and is the sum of the following components, all expressed in length units:

- the **wellhead pressure** at the given liquid production rate,
- the **net** hydrostatic pressure acting on the pump, and
- the **frictional** pressure drop that occurs in the tubing string at the given liquid rate.

The second term equals the true vertical depth (TVD) of the **dynamic liquid level**, the depth where the fluid level in the casing annulus stabilizes while producing the desired liquid rate. It is easily found from the value of the *PIP*, assuming that an **oil** column exists in the annulus above the pump setting depth:

$$L_{\text{dyn}} = \frac{L_{\text{set}} \text{ grad}_o + \text{CHP} - \text{PIP}}{\text{grad}_o - \text{grad}_g} \quad (5.6)$$

where

L_{dyn} = dynamic liquid level, ft
 L_{set} = TVD of pump setting, ft
 PIP = pump intake pressure, psi
 CHP = casinghead pressure, psi, and
 $\text{grad}_o, \text{grad}_g$ = oil and gas gradients in the annulus, psi/ft.

It should be noted here that the pump is usually set below this depth to provide the necessary **submergence** required for the proper pump operation. A rule of thumb value is at least **500 ft** of fluid over the pump, the other limit being the depth of the sandface (perforations). The reason is that in conventional installations the motor must be set above the depth of the sandface to ensure the liquid velocity required for proper **cooling**.

The frictional head loss in the tubing string can be estimated from Fig. A.1 in **Appendix A**, where losses for 1,000 ft of pipe length, Δh_{fr} , are plotted versus liquid rate. Alternatively, frictional head loss can be calculated from the **Hazen–Williams formula** given in Eq. (2.15).

The **total** frictional head loss in the tubing string is found as

$$\Delta H_{fr} = \Delta h_{fr} \frac{L_t}{1,000} \quad (5.7)$$

where

ΔH_{fr} = frictional head loss in tubing, ft

Δh_{fr} = frictional head loss in 1,000 ft of tubing, ft/1,000 ft, and

L_t = measured tubing length to the pump setting depth, ft.

If not done before the start of the installation design, the **size** of the existing or predicted **tubing** string should be checked for **extensive** frictional pressure drop. The generally accepted practice in the ESP industry is to limit the frictional head loss in the tubing string to 100 ft/1,000 ft, as indicated by the horizontal dashed line in Fig. A.1. When the head loss found at the desired liquid production rate is greater than this value, selection of a **bigger** tubing size is recommended. If this rule is followed, the frictional energy loss in the tubing string consumes **less** than 10% of the total energy requirement of the ESP system [9,10], and this ensures an **economic** installation design.

Now, the *TDH* is calculated in head (ft) units as follows:

$$TDH = \frac{2.31}{\gamma_l} WHP + L_{dyn} + \Delta H_{fr} \quad (5.8)$$

where

TDH = total dynamic head, ft

WHP = producing wellhead pressure, psi

L_{dyn} = TVD of the dynamic liquid level, ft

ΔH_{fr} = frictional head loss in the tubing string, ft, and

γ_l = specific gravity of the produced liquid, —.

5.3.3 Selection of the Pump

The selection of the proper ESP pump for operating in a given well involves

- the determination of the pump **series** (outside diameter (OD)) to be used,
- the selection of the required **pump type**,
- the calculation of the required **number of stages**, and
- the checking of the mechanical **strength** of the selected pump.

5.3.3.1 Pump Series

As with all ESP components, the most important selection requirement is that the chosen pump should **fit** in the casing string of the well. Centrifugal pumps are manufactured in different ODs, their series numbers usually represent their ODs, e.g., a 338 Series pump's OD is 3.38", 400 Series means 4.00" OD, etc. To

facilitate pump selection, Table B.1 in **Appendix B** contains the drift diameters of several API casing sizes; ODs of ESPs are published by the manufacturers.

When selecting the pump series to be used, **economic** considerations command the choice of pumps with the **largest** OD that can be run in the given casing size. This is because pumps with greater ODs have higher efficiencies and thus lower power requirements. On the other hand, ESP components (pumps, motors, etc.) with identical technical parameters but different ODs, due to manufacturing complications, are less expensive if they have a **larger** OD.

5.3.3.2 Pump Type

After the pump series has been determined, the proper **type** is selected from the available options. Pump types differ in the design of the stages such as the shape and number of vanes and the height, angle, and length of vanes. All these factors have an impact on the liquid flow rate and the head developed by the pump stage.

Pump type selection is based on the comparison of the well's **desired** liquid production rate and the **recommended** liquid capacity ranges of the pump types available. The ESP type selected should

- have the required liquid rate within its **optimum capacity range**, and
- have the highest pump **efficiency**, i.e., the rate belonging to its best efficiency point (BEP) should fall close to the desired rate.

The selection process is facilitated by tables like the one given in **Appendix C** containing data of different series pumps available from a leading manufacturer.

If several pump types meet these requirements, the final selection may be based on the detailed comparison of the different pumps' performance at the desired rate and based on their published performance curves as follows.

- Life expectancy of pumps is increased if the desired rate falls to the right of the BEP point.
- Compare the head developed and the horsepower required and choose the pump with the highest head and lowest power requirement.
- If the accuracy of the well's inflow parameters is **unreliable**, choose a pump with a "**steep**" H–Q (head capacity vs. liquid rate) curve. Such pumps will produce a liquid rate near the desired volume even if the actual TDH differs from the design value.
- When pumping gassy fluids, a steep **H–Q curve** is advantageous because changes in the dynamic head due to gas production have a lower effect on the pump's capacity.

5.3.3.3 Number of Stages, Pump Power

As discussed in **Section 3.2**, the individual stages in a multistage centrifugal pump develop identical heads that are **additive**, so a pump with a given

number of stages develops a total head equal to the sum of the heads of all stages. This allows the determination of the necessary number of stages for a given case where the required total head represented by the TDH is known.

Because the pump type is already known at this step of the design process, one has to use the **performance curves** of the selected type, made available by the manufacturer. As previously detailed, pump performance curves contain, in the function of the pumping rate, the following measured operational parameters:

- the **head** developed,
- the **efficiency**, and
- the **power** required, all for a given number (usually one) of stages.

To determine the required **number** of stages, one has to read the head developed by one stage at the desired liquid production rate and use the following formula:

$$Stages = \frac{TDH}{head/stage} \quad (5.9)$$

where

$Stages$ = required number of stages, —

TDH = total dynamic head, ft, and

$head/stage$ = head developed by one stage of the selected pump, ft.

This formula is used if the operating frequency is identical to the base frequency of the pump's head performance curves. If the pump is operated at a frequency different from the value used for the head performance curves, the formula must be modified as follows:

$$Stages = \frac{TDH}{head/stage_{base}} \left(\frac{f_{base}}{f} \right)^2 \quad (5.10)$$

where

$Stages$ = required number of stages, —

TDH = total dynamic head, ft

f, f_{base} = actual and base frequencies, Hz, and

$head/stage_{base}$ = head per stage at frequency f_{base} , ft.

Because it would be too costly to fabricate ESPs with every possible number of stages, pumps are manufactured in different **housings** containing discrete numbers of stages. One should therefore consult the manufacturer's catalog for **standard** housings and to select that combination of housings that makes up the required number of stages. Here there are some more choices for the designer: if the combination of housings gives a greater number of stages than needed, it may be accepted or **dummy** stages must be inserted in the pump to exactly match the stage number calculated from Eq. (5.10).

At this point the **power** required to drive the selected ESP can be calculated. It is found from the pump performance curves, where the power needed to drive a given number of stages (usually one) is plotted in the function of the pumping rate at a specified base electric frequency. After reading the necessary power per stage at the desired liquid rate from the performance curve, one finds the total power needed at any electrical frequency from the following formula:

$$BHP_{pump} = \frac{BHP}{stage} Stages \gamma_l \left(\frac{f}{f_{base}} \right)^3 \quad (5.11)$$

where

BHP_{pump} = required pump power, HP

$BHP/stage$ = power to drive one stage at f_{base} , HP/stage

$Stages$ = the selected number of pump stages, —

f, f_{base} = actual and base (usually 60 Hz) frequencies, Hz, and

γ_l = specific gravity of the produced liquid, —.

It must be noted that in both calculations the pump performance curves must be read at the correct liquid rates valid at the different driving frequencies. This involves the use of the **affinity law** formula for pumping rate, as illustrated in the following example.

Example 5.1

Find the required number of stages and the pump power for an operation at 50 Hz if the desired liquid rate is 3,000 bpd, the system's TDH is 2,000 ft, and the pump performance curves are known at 60 Hz. The liquid pumped has a specific gravity of $\gamma_l = 0.96$.

Solution

First calculate the liquid rate at 60 Hz pump operation that corresponds to the desired value at 50 Hz, using Eq. (4.32):

$$q_{60} = 3,000 \cdot 60/50 = 3,600 \text{ bpd}$$

Read from its head-rate performance curve valid at 60 Hz operation the pump was found to develop 25 ft per stage at this rate.

The necessary number of stages is found from Eq. (5.10) as follows:

$$Stages = 2,000/25(60/50)^2 = 115 \text{ stages.}$$

To find the required pump power, first the power requirement per stage is read from the BHP-rate performance curve of the pump at 2,600 bpd and 60 Hz. Let us assume a value of 0.75 HP/stage.

Required pump power at 50 Hz operation is found from Eq. (5.11) as follows:

$$BHP_{pump} = 0.75 \cdot 115 \cdot 0.96 \cdot (50/60)^3 = 48 \text{ HP.}$$

5.3.3.4 Checking the Pump's Mechanical Strength

The pump selected by the previous procedure must be checked for mechanical strength of the pump **shaft** and the pump **housing** to ensure that the loadings on the shaft and/or the housing do not exceed the limits specified by the manufacturer.

The strength of pump shafts is given by the manufacturer (see Table C.1 in **Appendix C**) by specifying the maximum allowed **power**, $HP_{shaft\ base}$, the shaft can transmit at a given frequency (usually 60 Hz). Using this value, the allowed shaft power at any electrical frequency is found from

$$HP_{shaft} = HP_{shaft\ base} \left(\frac{f}{f_{base}} \right) \quad (5.12)$$

where

HP_{shaft} = pump shaft rating at frequency f , HP

$HP_{shaft\ base}$ = pump shaft rating at f_{base} , HP, and

f, f_{base} = actual and base (usually 60 Hz) frequencies, Hz.

The calculated maximum **allowed** power on the pump shaft, HP_{shaft} , must be less than the required pump power BHP_{pump} previously calculated so that the shaft is not overloaded by the pumping action.

Manufacturers specify the mechanical strength of pump **housings** by publishing the allowed internal **burst pressure** for their pumps. This pressure should be compared with the maximum possible internal pressure that occurs when the pump develops the **maximum** head. Most manufacturers use the “**shut-in**” head valid at zero flow rate, others use the highest head found in the recommended liquid rate range. Again, the performance curves of the selected pump must be used to find the maximum head per stage value at a base frequency, from which the maximum internal pressure at the actual electrical frequency is found as follows:

$$p_{max} = \left(\frac{head}{stage} \right)_{max} Stages\ grad_l \left(\frac{f}{f_{base}} \right)^2 \quad (5.13)$$

where

p_{max} = maximum pump internal pressure, psi

$(head/stage)_{max}$ = shut-in head of one stage at f_{base} , ft

$Stages$ = the selected number of pump stages, —

f, f_{base} = actual and base (usually 60 Hz) frequencies, Hz

$grad_l$ = fluid gradient, psi/ft.

For safe operation, the calculated **maximum** pressure must be less than the burst pressure **rating** of the pump housing.

5.3.4 Selection of the Protector

The protector or seal section of the ESP unit performs several **vital** functions for the proper operation of the installation the most important of them being

absorbing the axial **thrust** developed in the pump. This is the reason why protectors are selected primarily on the basis of the calculated thrust **load** developed by the pump.

In addition to thrust **load capacity**, several other features must be considered when selecting the proper protector for a particular application:

- the right size (**series**) is to be chosen,
- the protector shaft should be capable to transfer the required **power**, and
- the protector's oil **expansion** capacity should be sufficient.

The available **sizes** of protectors are compatible with motor and pump series and the proper OD is selected to match the ODs of the selected motor and the pump.

Most ESPs belong to the **floating** impeller class where all axial forces developing in the stages are absorbed by the thrust washers of the stages. The only thrust occurring in such pumps is the **downt thrust** on the pump shaft due to the shaft being exposed to a large pressure differential. Because this load is directly transferred to the protector, the load on the protector's thrust bearing is found by multiplying the cross-sectional area of the pump shaft by the maximum pressure developing inside the pump, found from Eq. (5.13):

$$F_{TB} = 0.785 \, p_{\max} \, d^2 \quad (5.14)$$

where

F_{TB} = maximum load on thrust bearing, lb

$p_{\max}$ = maximum internal pressure in the pump, psi, and

d = pump shaft diameter, in.

In **fixed** impeller pumps (a.k.a. **compression** pumps), in contrast to those with floating impellers, high axial loads develop in the pump stages because the impellers are axially fixed to the shaft. Thrust forces developing on the impellers, therefore, are directly transferred to the protector's thrust bearing. The calculation of these forces, however, requires that measured thrust values for the given stage be available. To increase the safety of installation design for compression pumps, manufacturers usually publish thrust values at **shut-in** conditions in lb/stage units for 50 or 60 Hz operation. Thus, for compression pumps the maximum load on the protector's thrust bearing at those base frequencies is found as

$$F_{TB} = \left(\frac{\text{thrust}}{\text{stage}} \right)_{\max} \text{ Stages} \quad (5.15)$$

where

F_{TB} = maximum load on thrust bearing, lb

$(\text{thrust/stage})_{\max}$ = shut-in thrust developed by one stage, lb, and

Stages = the selected number of pump stages, —.

Should the pump be driven by a motor run at a different frequency than 60 or 50 Hz, the load just calculated is corrected as follows:

$$F_{TB \text{ corr}} = F_{TB} \left(\frac{f}{f_{\text{base}}} \right)^2 \quad (5.16)$$

where

$F_{TB \text{ corr}}$ = thrust bearing load at frequency f , lb

F_{TB} = thrust bearing load at base frequency f_{base} , lb, and

f, f_{base} = actual and base electric frequencies, Hz.

As discussed in Section 3.4, the capacity of the protector's thrust bearing relies on the operating **temperature** because the viscosity of motor oil decreases at elevated temperatures making the oil film on the bearing surfaces to become thinner. Therefore, performance diagrams similar to the one shown in Fig. 3.44 have to be used and a protector with a load capacity greater than the load just calculated is selected.

The shafts of different protectors have different mechanical strengths, and manufacturers usually specify the maximum **allowed** power on their protector shafts at a base electrical frequency, usually 60 Hz. Allowed shaft power at any other frequency is found from the next formula, derived from the fact that torque is the limiting factor on the operation of shafts:

$$BHP_{\text{shaft}} = BHP_{\text{base}} \left(\frac{f}{f_{\text{base}}} \right) \quad (5.17)$$

where

BHP_{shaft} = allowed shaft power at frequency f , HP

BHP_{base} = allowed shaft power at frequency f_{base} , HP, and

f, f_{base} = actual and base (usually 60 Hz) frequencies, Hz.

The protector selected based on thrust bearing load must be checked by comparing its shaft rating calculated from Eq. (5.17) to the power needed to drive the pump. For proper operation, a protector with a greater shaft rating than the pump power calculated from Eq. (5.11) must be chosen.

ESP protectors or seal sections are available in different **designs** with different oil expansion capacities. Selecting the right design depends on well deviation, temperature, and the density of the produced liquid. In deviated wells, **bag-type** protectors should be used because the operation of **labyrinth-type** protectors is not satisfactory in wells with inclinations from the vertical of more than 30 degrees.

To increase the protection of motors in hostile environments, to save on pulling costs, or to increase the run time of the ESP system, **tandem protectors**, i.e., two or more sections stacked in series, can be used. Such protectors may contain both bag- and labyrinth-type chambers.

The protector's operation consumes a definite **power**, which should be provided by the ESP motor. Most ESP equipment manufacturers (except **Centrilift**) include this power in the pump horsepower requirement of their published pump performance curves. For such cases the total horsepower needed to drive the ESP unit equals the pump power calculated from Eq. (5.11).

5.3.5 Motor Selection

When selecting the submersible motor to match the ESP pump already chosen, one should determine

- the proper motor **series** (OD),
- the required motor **power**, and
- the right combination of motor **voltage** and **amperage**.

Motor series denotes, just like with the pump, the OD of the motor. Again, the motor series with the largest OD that can be run in the well casing should be preferred. Many times, motors with ODs different from the ESP pump are used, mainly because larger diameter motors are less expensive.

As discussed in Section 3.3, ESP motors are cooled by well fluids flowing past the motor's outside surface and the recommended **minimum** flow velocity required for proper cooling is 1 ft/s. To prevent the erosion of the housing, however, velocities must be limited to 12 ft/s for **clean** fluids and to 7 ft/s for fluids containing abrasives. The formula given in the following allows the calculation of flow velocities in the annulus formed by the casing string and the ESP:

$$v_l = 0.0119 \frac{q'_l}{ID_c^2 - OD_m^2} \quad (5.18)$$

where

v_l = liquid velocity around motor, ft/s

q'_l = "in-situ" liquid flow rate, bpd

ID_c = inside diameter of the casing string, in, and

OD_m = outside diameter of the motor, in.

The criterion for selecting the motor with the proper power capacity is the **total** power requirement of the ESP system, BHP_{system} , that consists of the powers needed to run (1) the **pump**, (2) the **protector**, and (3) the gas **separator** (if used). If no rotary gas separator is used, like in the conventional design case just discussed, and the power to drive the protector is included in the pump horsepower, the ESP system's total power demand equals the pump power, BHP_{pump} , calculated from Eq. (5.11). Because ESP motors in every series are manufactured with several discrete power ratings, one should choose a motor with a nameplate rating just above the required system power, BHP_{system} .

Manufacturers publish motor performance data at a stated electrical frequency, usually 60 Hz and nameplate motor powers change with the driving frequency as given here:

$$HP_{motor} = HP_{base} \left(\frac{f}{f_{base}} \right) \quad (5.19)$$

where

HP_{motor} = motor nameplate power at frequency f , HP

HP_{base} = motor nameplate power at frequency f_{base} , HP, and

f, f_{base} = actual and base (usually 60 Hz) frequencies, Hz.

Because induction motors are controlled to keep the **ratio** of the voltage and frequency **constant**, nameplate voltage at a given frequency is calculated from the published motor voltage valid at a base frequency (see Section 4.7.4.1) using the next formula:

$$U_{np} = U_{base} \left(\frac{f}{f_{base}} \right) \quad (5.20)$$

where

U_{np} = motor nameplate voltage at frequency f , HP

U_{base} = motor nameplate voltage at frequency f_{base} , HP, and

f, f_{base} = actual and base (usually 60 Hz) frequencies, Hz.

Motor nameplate current is kept at the value published by the manufacturer, and the actual current **demand** of the motor is calculated from the nameplate data and the system's power requirement as follows:

$$I = I_{np} \frac{BHP_{system}}{HP_{np}} \quad (5.21)$$

where:

I = motor current demand, amps

I_{np} = nameplate current of the motor, amps

HP_{np} = nameplate power of the motor, HP, and

BHP_{system} = power requirement of the ESP system, HP.

Technical data on ESP motors are published by manufacturers; Tables D.1 and D.2 in **Appendix D** contain sample data for motors of two series from a leading company. As seen in the table, most of the motors with a given power rating come in several versions differing from each other by the combination of their required **voltage** and **amperage**. Because electric power is defined as the product of voltage and amperage, motors with higher voltage values require less current, and vice versa. This feature gives the designer a great flexibility to achieve an **optimum** selection of motors for various conditions with the objective of maximizing the **economy** of fluid lifting.

The prime factor in selecting the motor voltage is the running **depth** of the ESP unit because it heavily affects the **voltage drop** in the electric cable. Deeper installations require longer cables and involve higher voltage drops across the cable; this leads to greater amounts of **wasted** electrical energy and lower system efficiencies. In such conditions, selecting a motor with a higher voltage rating means that a much lower current flows through the cable causing a lower voltage drop; consequently, the amount of wasted power is reduced. In addition, the use of higher voltages is advantageous in **high-temperature** wells because higher voltage motors need less current to develop the same horsepower; thus the operating temperature of the electric cable is reduced, and cable life is increased.

Based on the considerations just discussed and an analysis of the power flow in the ESP system, Powers [10] concludes that in most situations the motor with the **highest** voltage requirement is the proper choice. This not only reduces the total power consumption but many times allows the use of a smaller and less expensive power cable.

Example 5.2

Select the proper 456 series motor for the conditions in Example 5.1 and find its main parameters based on motor data at 60 Hz operation, given in Table D.1 in Appendix D.

Solution

The required motor power at 50 Hz was calculated as 48 HP in the previous example. The corresponding motor power at 60 Hz is found from Eq. (5.19):

$$HP_{60} = 48 \cdot 60/50 = 58 \text{ HP.}$$

From Table D.1 the motor with the next higher power rating is chosen, its nameplate data are as follows:

$$NPHP_{60} = 72 \text{ HP, } NPV_{60} = 2,288 \text{ Volts, } NP \text{ Amps}_{60} = 20 \text{ Amps.}$$

The motor's nameplate power at 50 Hz is expressed from Eq. (5.19):

$$NPHP_{50} = 72 \cdot 50/60 = 60 \text{ HP.}$$

Motor nameplate voltage at 50 Hz operation is found from Eq. (5.20):

$$U_{np} = 2,288 \cdot 50/60 = 1,907 \text{ Volts.}$$

Current demand of the motor is calculated from Eq. (5.21):

$$I = 20 \cdot 48/60 = 16 \text{ Amps.}$$

5.3.6 Selection of the Power Cable

The electric power cable is a **vital** part and usually is the most **expensive** item of the ESP system. Its proper selection is not only a technical task but requires,

as it will be shown later, **economic** considerations as well. Cable for the ESP industry is manufactured in a wide variety of types, sizes, etc. When designing an ESP installation, the right cable is selected by determining its required (1) **length**, (2) **type**, and (3) **size**, with proper considerations of all operational conditions.

5.3.6.1 Cable Length

The length of the power cable is determined from the running depth of the motor to which a sufficient length needed for the safe connection of surface equipment (about 100 ft) is added. The **main** cable (usually a round configuration) should reach down to the top of the ESP unit from where a motor lead **extension** (MLE: usually a flat cable section) runs to the **pothead** of the motor.

5.3.6.2 Cable Type

ESP cables are manufactured in a wide variety of types, i.e., with different insulating and conductor materials, constructions, and armors. The proper choice primarily depends on well conditions, most importantly on well **temperature**, and the composition, gas content, and corrosiveness of the well fluid. The main considerations [11,12] on the proper selection of the individual parts of the cable can be summed up as follows:

- The material of choice for cable conductors is **copper**, especially in deeper wells because aluminum has a lower conductivity.
- The two most common materials used for the **insulation** of the individual conductors are **polypropylene** and **EPDM** (ethylene propylene diene monomer); their temperature limits are 205 and 400°F, respectively. Polypropylene is susceptible to degradation by light hydrocarbons, CO₂, and hydrocarbon gases. EPDM materials have a much wider temperature range and are less sensitive to gases.
- **Jackets** are made from Nitril or EPDM materials, the latter being used at higher temperatures.
- **Braids** or tapes provide additional strength and protection to cable components, see Fig. 3.46.
- Metal **armor** provides mechanical protection to the cable during running and pulling and the choices are galvanized or stainless steel and Monel.

5.3.6.3 Cable Size

The main considerations for selecting the size of the power cable are (1) the physical **dimensions** and (2) the **voltage drop** along the cable. The physical size should be chosen so that the cable fits in the annulus between the casing string and the ESP unit. The voltage drop occurring in the cable is a function of the conductor size and the motor current and is a direct indicator of the energy losses occurring along the cable.

If the effect of conductor size on the operation of the whole ESP system is investigated, it is easy to see that smaller conductors mean greater energy losses and proportionally greater operational costs. However, at the same time, cable investment cost is lower than for a bigger conductor size. Bigger cable sizes, on the other hand, result in lower operating expenses but higher investment costs. Therefore, the selection of the optimum cable size should involve not only technical but **economic** considerations as well.

Before the application of optimization methods, one of the following **rules of thumb** was used to select the size of the power cable [10]:

- The cable with a voltage drop of less than 30 V per 1,000 ft of cable length was chosen. Consult Fig. E.1 in **Appendix E** where this limit is shown on the cable voltage drop diagram.
- The cable with a total voltage drop less the 5% of the motor nameplate voltage was selected.

Optimization procedures [13,14] are based on finding the least value of total operating **costs** over the expected life of the cable. The total operating cost is the sum of the **capital** and **operating** expenses and these, as described previously, vary with cable size. Because an increase of the conductor size involves increased capital costs but decreased operating costs, a cable providing the minimum of total costs can surely be found. It is easy to see that, contrary to the rules previously used, the smallest possible size may not be the best selection. Calculation of the cost items is described in the following.

The monthly payment necessary to **pay back** the capital outlay for the cable over its expected life, considering the **time value of money**, is found from the formula used for calculating partial payments of loans. In the following formula, the multiplier of the investment cost is the **capital recovery factor** that defines the variation of the capital cost's contribution to the total cost:

$$PB = CI \frac{(1+p)^N}{(1+p)^N - 1} p \quad (5.22)$$

where

PB = monthly payback for cable, \$/month

CI = capital investment for cable, \$

p = monthly interest rate, %, and

N = expected life of the cable, months.

In the formula, the monthly interest rate, p , is calculated as

$$p = \frac{P}{1,200} \quad (5.23)$$

where

p = monthly interest rate, %, and

P = prime interest rate, %/year.

The monthly **operating** expenses are equal to the **cost** of the electric power **wasted** along the cable. The calculation model of **Vandevier** [13] is detailed in the following, who used the next simplifications when calculating this cost item:

- Instead of using an average along the cable length, cable temperature is taken to be equal to the well's bottomhole temperature.
- The cable's inductive reactance is neglected because of its low value for any ESP cable, as described in Section 3.6.3.

First the total **resistance** of the cable (in ohms) at well temperature is calculated from Eq. (2.36), reproduced here:

$$R_T = \frac{L_c r}{1,000} [1 + 0.00214(BHT - 77)] \quad (5.24)$$

where

R_T = cable resistance at temperature T , ohms

L_c = cable length, ft

r = resistance of conductor at 77°F, ohm/1,000 ft, and

BHT = bottomhole temperature, °F.

The power **loss** in the cable (in kW units) is calculated from the cable resistance and the motor current:

$$\Delta P_c = \frac{3I^2 R_T}{1,000} \quad (5.25)$$

where

ΔP_c = power loss in cable, kW

R_T = cable resistance at temperature T , ohms, and

I = required motor current, amps.

Finally, the monthly **cost** of this power loss (\$/month) is found as

$$C_{cl} = 720 \Delta P_c \frac{c_e}{100} \quad (5.26)$$

where

C_{cl} = monthly cost of power loss in cable, \$/month

ΔP_c = power loss in cable, kW, and

c_e = cost of electric power, c/kWh.

The total monthly operating cost of the cable is the sum of the **investment** (Eq. 5.22) and the **power** (Eq. 5.26) costs; it is clearly a function of the cable conductor size. The optimum cable size is found where this total cost is at a **minimum**. When using the calculation model described previously, the steps of the cable size selection are as follows:

1. Based on the ampacity chart of the selected cable type (for example see Fig. 3.48), the **allowed** amperage values of each available size are determined at the well's bottomhole temperature.

2. Cable sizes with allowed currents greater than the required motor current are selected for further economic analysis.
3. After performing the calculation of total costs for each candidate, the cable with the **minimum** of monthly costs is chosen.

5.3.6.4 Checking Motor Startup

It should be noted that the cable size selected affects the operating conditions of the motor especially immediately after **startup**. At system startup ESP motors draw more than five times greater **instantaneous** currents than their nameplate current. Due to the increased current a high voltage drop develops across the cable and the voltage reaching the ESP motor may be insufficient to start the motor. Normal starts require about 50% of the motor's nameplate voltage to be available at the motor's terminals.

As discussed in Section 3.3.1.3.3, the following simple formula, although being an estimate only, permits a rapid check on the cooperation of the cable and the motor at startup conditions. According to Eq. (3.8), repeated here, the ratio of startup and nameplate voltages is found from

$$\frac{U_{\text{start}}}{U_{np}} = \frac{U_{np} - 4IR_T}{U_{np}} \quad (5.27)$$

where

U_{start} = motor voltage at startup, volts

U_{np} = motor nameplate voltage, volts

I = required motor current, amps, and

R_T = cable resistance, ohms.

The selected cable size is satisfactory if this ratio is greater than 0.5, because in those conditions more than 50% of the nameplate voltage reaches the motor's terminals and no startup problems will occur.

5.3.7 Switchboard and Transformer Selection

When specifying switchboards and transformers, the most important information is the maximum **power** of the ESP system required at the surface. This can be found from the necessary surface voltage and the motor current. Surface voltage is composed of the motor's required terminal (nameplate) voltage plus the voltage drop across the power cable. Because at this phase of the installation design, the nameplate voltage has already been selected, and the cable voltage drop equals the product of cable resistance (calculated from Eq. 5.24) and the motor current, the installation's **surface voltage** requirement is found as

$$U_{\text{surf}} = U_{np} + 1.732 R_T I \quad (5.28)$$

where

U_{surf} = required surface voltage, volts

U_{np} = motor nameplate voltage (Eq. 5.20), volts

I = required motor current, amps, and

R_T = cable resistance, ohms.

From the available models, a switchboard with a rated voltage above this value is selected. The **power** rating of the switchboard (in kVA units) is found from the formula used to find three-phase electric power:

$$P_{\text{surf}} = \frac{\sqrt{3} U_{\text{surf}} I}{1,000} = 0.001732 U_{\text{surf}} I \quad (5.29)$$

where

P_{surf} = required surface power, kVA

U_{surf} = required surface voltage, volts, and

I = required motor current, amps.

The necessary three-phase step-up or step-down transformer should have the same kVA rating as the switchboard.

5.3.8 Miscellaneous Equipment

At this point of the design procedure, all important parts of the ESP installation are selected and a detailed order can be placed to the manufacturer. The only remaining pieces of equipment can easily be chosen from the manufacturer's catalog and only a listing with the most important considerations is given in the following.

5.3.8.1 Downhole Equipment

- **MLE** is a flat cable section whose length should be about 6 ft longer than the combined length of the gas separator, pump, and protector.
- **Flat cable guards** should be installed along the ESP unit and the necessary number of 18 in sections is chosen.
- **Cable bands** on the power cable are spaced every 15 ft along the tubing string and 2 ft apart along the flat cable.
- **Check and drain valves** are selected to be compatible with the tubing size.

5.3.8.2 Surface Equipment

- Up-to-date **wellheads** use power feed-through connections and are to be chosen to be compatible with the well's casinghead, tubing, and power cable sizes.
- A **junction box** is needed to provide a connection point for the power and surface cables and is selected from available models.
- **Surface cable** should have sufficient length to reach from the junction box to the switchboard and must be at least the same size as the power cable.
- **Service equipment** includes cable reels, reel supports, and cable guide wheels; they should match the selected cable sizes.

Example 5.3

Select the main components of an ESP installation running at a constant frequency of 60 Hz in a vertical well with the following data.

Well physical data

Casing ID	6.01 in
Tubing ID	1.995 in
Tubing OD	2.375 in
Tubing new?	Yes
Perforation depth	5,000 ft
Pump depth	4,900 ft

Well performance data

WHP @ desired rate	80 psi
CHP @ desired rate	20 psi
Desired liquid rate	1,300 STB/day
Water cut	90%
SBHP	1,800 psi
PI	0.9 bpd/psi
GOR produced	50 scf/bbl
Pump suction temperature	200 F

Fluid properties

Specific gravity of oil	0.86
Specific gravity of water	1.0
Specific gravity of gas	0.6

Power supply data

Available surface voltage	1,200 V

Solution

Following the steps of the conventional ESP design, first the inflow conditions of the well are determined.

Well inflow calculations

The flowing bottomhole pressure at the desired rate is found from Eq. (5.1) as

$$FBHP = 1,800 - 1,300 / 0.9 = 356 \text{ psi.}$$

The PIP is determined from the fluid gradient in the casing annulus:

$$\gamma_l = (0.91 + 0.10 \cdot 0.86) = 0.986,$$

and

$$grad_l = 0.986 \cdot 0.433 = 0.427 \text{ psi/ft and using Eq. (5.2):}$$

$$PIP = 356 - (5,000 - 4,900) \cdot 0.427 = 313 \text{ psi.}$$

To find the solution GOR at suction conditions, Eq. (5.3) is used, where the oil's API gravity, and the variable y is found as

$$^\circ API = 141.5 / 0.86 - 31.5 = 33.$$

$$y = 0.00091 \cdot 200 - 0.01215 \cdot 33 = -0.231,$$

and

$$R_s = 0.6 \left[313 / (18 \cdot 10^{-0.231}) \right]^{1.205} = 36 \text{ scf/bbl.}$$

The free gas/oil ratio at suction conditions is

$$R_{\text{free}} = GOR - R_s = 50 - 36 = 14 \text{ scf/bbl.}$$

To find the actual gas volume, the volume factor of gas, B_g , has to be calculated, for which the knowledge of the deviation factor is necessary. The pseudocritical state parameters are calculated from Eqs. (4.27) and (4.28):

$$p_{pc} = 709 - 58.7 \cdot 0.6 = 674 \text{ psia,}$$

and

$$T_{pc} = 170.5 + 307.3 \cdot 0.7 = 355 \text{ R.}$$

The reduced parameters and the deviation factor from Eq. (4.26):

$$p_{pr} = 328 / 674 = 0.486,$$

and

$$T_{pr} = (200 + 460) / 355 = 1.86.$$

$$Z = 1 - 3.52 \cdot 0.486 / 10^{0.9813 \cdot 1.86} + 0.274 \cdot 0.486^2 / 10^{0.8157 \cdot 1.86} = 0.98.$$

Gas volume factor now can be determined from Eq. (4.25):

$$B_g = 0.0283 \cdot 0.98 \cdot (200 + 460) / 328 = 0.056 \text{ cu ft/scf.}$$

The free gas volumetric rate is found as follows:

$$q_{\text{free}} = R_{\text{free}} \cdot q_o \cdot B_g = 14(1,300 \cdot 0.1) \cdot 0.056 = 102 \text{ cu ft/day,}$$

converted to bpd units:

$$q_{\text{free}} = 102 / 5.61 = 18 \text{ bpd.}$$

The calculation of the liquid volumetric rate follows with the determination of the oil volume factor from Eq. (5.5):

$$F = 36 (0.6/0.86)^{0.5} + 1.25 \cdot 200 = 280,$$

and

$$B_o = 0.972 + 1.47 \cdot 10^{-4} \cdot 280^{1.175} = 1.08 \text{ bbl/STB}.$$

The liquid volumetric rate handled by the pump is found from Eq. (5.4) and assuming $B_w = 1$:

$$q'_l = [1,300/(1-90/100)]1.08 + 1,300 \cdot 90/100 = 1,310 \text{ bpd}.$$

The percentage of the gas rate in the total fluid rate at the pump's suction conditions is $18/(1,310 + 18) = 0.01 = 1\%$, a negligible amount of gas; therefore the ESP system is designed for pumping liquid only.

TDH calculations

The dynamic liquid level at the desired liquid rate from Eq. (5.6), neglecting the gas gradient:

$$L_{\text{dyn}} = (4,900 \cdot 0.86 \cdot 0.433 + 20 - 313)/(0.86 \cdot 0.433) = 4,113 \text{ ft}.$$

Frictional head loss is determined from Figure A.1 for a new tubing as

$$\Delta h_{\text{fr}} = 44 \text{ ft}/1,000 \text{ ft}.$$

The total head loss, calculated from Eq. (5.7):

$$\Delta H_{\text{fr}} = (44 \cdot 4,900)/1,000 = 216 \text{ ft}.$$

Now the total dynamic head to be developed by the ESP can be determined by using Eq. (5.8):

$$TDH = 2.31 \cdot 80/0.986 + 4,113 + 216 = 4,516 \text{ ft}$$

Selection of the pump

Based on the casing size and the desired liquid rate, a 400 Series D1400N-type pump is selected from Table C.1 in **Appendix C**.

Using the performance data of the selected pump, the following specific parameters can be determined at the desired liquid rate of 1,310 bpd:

Head/stage	21.8 ft/stage
BHP/stage	0.33 HP/stage
Maximum head/stage	30.5 ft/stage
Allowed shaft power	200 HP
Shaft diameter	0.688 in
Housing burst pressure	5,000 psi

The number of pump stages required to develop the TDH value is determined from Eq. (5.9):

$$\text{Stages} = 4,516/21.8 = 207 \text{ stages.}$$

From manufacturer's catalog, the nearest pump with 221 stages (Housing 120) is selected.

The selected pump must be checked for mechanical strength, first by finding the maximum power occurring on its shaft from Eq. (5.11):

$$BHP_{\text{pump}} = 0.33 \cdot 221 \cdot 0.986(60/60)^3 = 72 \text{ HP}$$

Which is less than the allowed power of 200 HP, the shaft is OK.

Checking the strength of the housing necessitates calculation of the maximum internal pressure from Eq. (5.13):

$$p_{\text{max}} = 30.5 \cdot 221 \cdot 0.427(60/60)^2 = 2,878 \text{ psi}$$

which is less than the allowed burst pressure of the given pump housing.

Selection of the protector

The selected D1400N pump is a floating impeller pump; therefore, the selection of the necessary protector is based on the downthrust developed on the pump shaft. The maximum thrust acting on the protector's thrust bearing is found from Eq. (5.14) as follows:

$$F_{TB} = 0.785 \cdot 2,878 \cdot 0.688^2 = 1,069 \text{ lbs.}$$

A 400 Series labyrinth-type protector is selected, whose allowed thrust loading at the bottomhole temperature is found as 4,000 lbs from Fig. 3.44. Because this is greater than the actual load, F_{TB} , the selected protector checks OK. The protector shaft is rated at 256 HP, while the power required by the pump is only 72 HP; therefore the selection is right.

Motor selection

Motors from the 456 series are considered and first the annular flow velocity of the produced liquid is checked from Eq. (5.18):

$$v_l = 0.0119 \cdot 1,310 / (6.01^2 - 4.56^2) = 1.02 \text{ ft/s}$$

which is sufficient for the cooling of the motor.

The motor should develop a total power equal to the power required to drive the pump because no gas separator is needed. To cover the required system power of 72 HP, a motor with the following parameters is selected from Table D.1 from **Appendix D**:

Power of selected motor	84 HP
Nameplate voltage	968 V
Nameplate current	55.5 A

As seen, not the motor with the maximum voltage was selected because of the 1,200 V surface voltage limitation of the electric supply system. The actual current demand of the selected motor is found from Eq. (5.21) as

$$I = 55.5 \text{ } 72/84 = 48 \text{ Amps.}$$

Selection of the power cable

Cable length is selected as 100 ft longer than the pump setting depth, i.e., 5,000 ft.

Based on the temperature conditions, a RedaHot 4 kV cable with galvanized armor is selected. As seen from the ampacity chart of this cable type (Fig. 3.48), all sizes can handle the required 48 A at the well's bottomhole temperature.

For the selection of the optimum cable size the following data are assumed:

Expected cable life	60 months
Prime interest rate	12%/year
Cost of electricity	5 c/kWh

From manufacturer's catalog, the properties of the different cable sizes are determined as given here:

Size (AWG)	Resistance (ohm/1,000 ft)	Unit Price (\$/ft)
1	0.139	11.92
2	0.170	10.46
4	0.271	8.71
6	0.431	7.21

To find the optimum size, the pay back cost, the cost of the power wasted in the cable, and the total cost have to be calculated using Eqs. (5.22)–(5.26). Calculation results are given in the following:

Cable Size (AWG)	Cable Cost (\$)	Monthly PayBack (\$/month)	Cable Resistance (ohms)	Power Loss (kW)	Cost of Power Loss (\$/month)	Total Cost (\$/month)
1	59,600	1,326	0.88	5.9	214	1,540
2	52,300	1,163	1.07	7.3	262	1,425
4	43,550	969	1.71	11.6	417	1,386
6	36,050	802	2.72	18.4	664	1,466

Based on the total monthly costs, the AWG Size 4 is selected as the one giving the most economical solution with the lowest total cost.

To check the startup voltage at the motor terminals, the ratio of the startup voltage to the nameplate voltage is found from Eq. (5.27):

$$U_{\text{start}}/U_{\text{np}} = (968 - 4\,48\,1.71)/968 = 0.66 = 66\%.$$

The electric motor, therefore, receives 66% of its nameplate voltage during startup, which is sufficient for a successful starting of the ESP unit.

Switchboard and transformer selection

The required surface voltage of the installation is the sum of the motor nameplate voltage and the cable voltage drop, as calculated from Eq. (5.28):

$$U_{\text{surf}} = 968 + 1.732\,1.71\,48 = 1,110 \text{ volts.}$$

Selection of the switchboard and the necessary step-down transformer is based on the kVA requirement of the installation, found from Eq. (5.29):

$$P_{\text{surf}} = 0.001732\,1,110\,48 = 92 \text{ kVA.}$$

Miscellaneous equipment

The downhole and surface miscellaneous pieces of equipment are selected from manufacturers' catalogs.

5.4 CONVENTIONAL DESIGN CONSIDERING MOTOR SLIP

5.4.1 Introduction

The design procedure detailed in the previous section did not consider the operational **slip** of the ESP motor. The motor was supposed to turn at a **constant** speed (3,500 RPM at 60 Hz operation) independently from its actual loading conditions with the pump running at the same speed. With these assumptions, the use of standard pump performance curves established at a speed of 3,500 RPM is justified in the selection process. Installation design was thus simplified but a comprehensive analysis of the **interaction** of the ESP and the motor easily proves that motor slip must be considered during the design process.

Motor slip, as discussed in **Chapter 2**, is the reduction in rotational speed of an electric induction motor at different torque loads compared with the synchronous speed. It follows from the operational principle of electric induction type motors that motor speed always **decreases** as the torque load on the motor **increases**, as can be clearly seen in motor performance curves (Figs. 3.22–3.24). This inevitable slip is the reason why in actual operating conditions the speed of the ESP motor is almost **always** different from 3,500 RPM, the value assumed when establishing submersible pump performance curves. Proper installation design, therefore, requires that the **interaction** of the ESP motor and the submersible pump at actual operating conditions should be analyzed.

5.4.2 Interaction of the ESP Motor and the Pump

The principle of describing the interaction of an ESP motor and a submersible pump is that at any time the power **developed** by the motor **equals** the power **required** to operate the submersible pump because they are series-connected devices. Because both powers are functions of the operating speed, the **cooperation** of the motor and pump will occur at a certain rotational speed. Determination of this speed depends on the different ways rotational speed affects motor and pump powers, detailed in the following:

- The power developed by the ESP motor at different speeds is found from experimentally determined motor performance curves like those presented in Figs. 3.13–3.15 showing the variation of motor speed with changes in motor loading.
- The submersible pump's required power, *BHP*, changes with the **cube** of the rotational speed, as follows from the **affinity laws**, see Eq. (2.19).

The assumptions used in the determination of the common operating speed of an ESP motor and a submersible pump are as follows:

- The desired liquid rate is given.
- The pump type has been selected.
- The number of required pump stages is known.
- The motor is selected, i.e., its nameplate power is known.

The determination of the actual operating speed of the ESP motor plus the submersible pump system requires an **iterative** procedure, the basic steps of which are listed here [15]. In the following description 60 Hz operation is assumed.

1. Determine the required pump brake horsepower at 3,500 RPM from the formula:

$$BHP = \left(\frac{BHP}{stage} \right)_{3500} Stages \gamma_l \quad (5.30)$$

where

BHP = required pump power at 3,500 RPM, HP

$(BHP/stage)_{3500}$ = power per stage from performance curve at 3,500 RPM, HP

Stages = the number of required pump stages, —, and

γ_l = specific gravity of the produced liquid, —.

2. Assuming that the motor develops a power just equal to the required pump brake horsepower, find the percentage loading of the motor based on its nameplate power:

$$\text{Loading} = \frac{BHP}{HP_{np}} 100 \quad (5.31)$$

where

Loading = motor loading, %

BHP = required pump brake horsepower, HP, and

HP_{np} = nameplate power of the selected ESP motor, HP.

3. Find the motor's actual rotational speed, N_{slip} , from the given motor's performance curves at the loading just calculated.
4. Because the speed of the pump is always equal to the motor's actual speed, N_{slip} , the required pump brake horsepower will change due to the change in operating speed. The new brake horsepower at the actual speed is found from the **affinity laws** as

$$BHP_{mod} = BHP \left(\frac{N_{slip}}{3500} \right)^3 \quad (5.32)$$

where

BHP_{mod} = modified pump brake horsepower, HP

BHP = required pump brake horsepower, HP, and

N_{slip} = motor speed at actual loading, RPM.

5. Check the difference of the powers BHP and BHP_{mod} at the original and the modified rotational speeds. If the difference is relatively great (more than about 0.5 HP, as suggested in [16]), repeat the calculations from **Step 2** by using BHP_{mod} instead of BHP to find a next approximation for N_{slip} .
6. If the difference in powers is below the recommended limit, the speed read from the motor performance curve, N_{slip} , gives the actual rotational speed of the motor–pump system. The common operating point of the ESP motor and the submersible pump has thus been determined.

Using the iterative procedure just described the operating speed of the motor–pump series-connected **system** has been determined. As already mentioned the system's actual speed seldom equals the rotational speed used when establishing the performance curves of submersible pumps (3,500 RPM at 60 Hz operation). This brings about the need for converting all pump performance curves to the new speed, N_{slip} , using the **affinity laws**. This is especially important for the head–rate performance curve because the head developed by the pump will be different from that developed at 3,500 RPM.

To summarize the description of the interaction of the ESP motor and the submersible pump one should understand that the motor and the pump are **series connected** and can only operate at a common rotational speed. This speed, N_{slip} , is a function of the power required to drive the pump and the motor's slip characteristics. In the knowledge of the actual speed the pump's actually developed head is determined from the **affinity laws**. This is the head including the effects of the ESP motor's slip and it must be used in installation design calculations, instead of the head valid at 3,500 RPM.

5.4.3 Changes in the Conventional Design

When the effect of motor slip is also considered in the design of an ESP installation, most of the steps of the procedure described for a conventional design in Section 5.3 are followed without modification. New calculations are included only after the ESP motor's selection and are given in the following.

5.4.3.1 Finding Actual Motor Speed

The determination of the common rotational speed of the ESP motor and the submersible pump is started with the calculation of the pump's brake horsepower at 3,500 RPM, using Eq. (5.30). Because the motor must develop the same power, its loading is found from Eq. (5.31). Now the motor's performance curve is used to find the motor speed, N_{slip} , valid at the calculated motor loading.

At N_{slip} the pump requires a different power, this can be found from Eq. (5.32) as BHP_{mod} . For the cooperation of the motor and the pump, this power must be equal to the one previously calculated. If the powers differ considerably, a new **iteration** must be made with the value of BHP_{mod} just calculated to find another modified power. After **convergence**, the motor's actual rotational speed including the effect of motor slip is found.

The required motor power equals the last value of BHP_{mod} . This determines the actual loading of the motor, as follows from Eq. (5.31). Now, from motor performance curves, the following parameters are read at the actual loading: (1) motor **efficiency**, (2) motor **power factor**, and (3) percent motor **amperage**. This latter parameter allows the determination of the **actual** motor current from the nameplate current as

$$I = I_{np} \frac{\% \text{ Amps}}{100} \quad (5.33)$$

where

I = actual motor current, amps

I_{np} = nameplate amperage of motor, amps, and

%Amps = percent amperage at motor loading, %.

5.4.3.2 Finding Pump Head Including Motor Slip

After the actual rotational speed of the ESP equipment including the effect of motor slip is known, the **head** developed by the pump should be calculated. This will inevitably be different from the value read from a normal pump performance curve valid at a pump speed of 3,500 RPM at 60 Hz service. Because pump performance curves at any speed are not readily available, the following calculation model can be used to find the pump head utilizing the normal head—rate performance curve usually available at 3,500 RPM.

First find the required pumping rate at 3,500 RPM that corresponds to the desired total liquid rate q'_l using the **affinity laws**:

$$q_{3500} = q'_l \frac{3,500}{N_{\text{slip}}} \quad (5.34)$$

where

q_{3500} = required pumping rate at 3,500 RPM, bpd

q'_l = total liquid rate at pump suction conditions, bpd, and

N_{slip} = motor speed including slip, RPM.

Next find, from the normal head performance curve of the pump valid at a speed of 3,500 RPM, the **head per stage** developed by the pump. Because the number of pump stages is known, the total head developed by the ESP is easily found. This must be adjusted back to the actual motor speed N_{slip} using the **affinity laws** again. Therefore, the total head developed by the pump at the actual rotational speed is calculated from the next formula:

$$H_{\text{slip}} = \left(\frac{\text{Head}}{\text{stage}} \right)_{3500} \text{Stages} \left(\frac{N_{\text{slip}}}{3,500} \right)^2 \quad (5.35)$$

where

H_{slip} = pump head at actual speed, ft

$(\text{Head}/\text{stage})_{3500}$ = pump head at a liquid rate of q_{3500} , ft/stage

Stages = number of pump stages, —, and

N_{slip} = motor speed including slip, RPM.

At this point of the ESP installation design procedure, the TDH calculated from well inflow data at an earlier stage must be compared with the **actual** total head H_{slip} developed by the pump at the actual rotational speed of the motor. Depending on the relation of the two heads, two situations can occur:

- If $H_{\text{slip}} \geq TDH$, then the ESP system can produce rates **equal** or **greater** than the desired total liquid rate.
- If $H_{\text{slip}} \leq TDH$, then the ESP system produces a liquid rate **less** than required. To remedy the situation, use a larger pump or use a greater number of pump stages.

In practice, usually the first situation occurs because of two reasons. First, when selecting the pump housing, a greater than necessary number of stages could be chosen and this increases the head developed. Motor slip, on the other hand, decreases pump speed and the operational head as well. The relationship of these two effects is normally such that the head including motor slip is still greater than the system's TDH requirement.

Example 5.4

Using the same data as in [Example 5.3](#), perform the calculation steps necessary to include the effect of motor slip on the ESP system's performance.

Solution

As detailed previously, design calculations exactly follow the procedure described for conventional installation design, see [Section 5.3](#). This is the reason why the results of identical calculation steps are not repeated here (see the solution of [Example 5.3](#)). Only the calculation steps to be included after the selection of the ESP motor are given in the following.

Finding the actual motor speed

The required pump power was found earlier as $BHP = 72$ HP. Because the motor was already selected with a nameplate power of $HP_{np} = 84$ HP, the load on the motor is found from [Eq. \(5.31\)](#) as

$$\text{Loading} = 72/84 \cdot 100 = 86\%.$$

The motor speed at this load, found from motor performance curves, is $N_{\text{slip}} = 3485$.

The required pump brake horsepower at this speed is found from [Eq. \(5.32\)](#) as

$$BHP_{\text{mod}} = 72(3,485/3,500)^3 = 71 \text{ HP}.$$

Because the original and modified brake horsepowers are not identical, another iteration step should be performed. For this assume that the required pump brake horsepower equals the modified value just calculated. Motor loading is then found from [Eq. \(5.31\)](#) as

$$\text{Loading} = 71/84 \cdot 100 = 85\%.$$

The motor speed at this load, found from motor performance curves, is $N_{\text{slip}} = 3490$.

The required pump brake horsepower at this speed is found from [Eq. \(5.32\)](#) as

$$BHP_{\text{mod}} = 71(3,490/3,500)^3 = 70 \text{ HP}.$$

The difference of the calculated horsepowers is less than in the previous step, but still not sufficiently low. A new iteration is performed with the modified value just calculated. Motor loading is then found from [Eq. \(5.31\)](#) as

$$\text{Loading} = 70/84 \cdot 100 = 83\%.$$

The motor speed at this load, found from motor performance curves, is $N_{\text{slip}} = 3,492$.

The required pump brake horsepower at this speed is found from [Eq. \(5.32\)](#) as

$$BHP_{\text{mod}} = 70(3,492/3,500)^3 = 70 \text{ HP}.$$

Now convergence is found and the common rotational speed of the ESP motor—submersible pump system is established at $N_{\text{slip}} = 3,492$.

Motor performance parameters, recorded from published performance curves at the actual motor loading of 83%, are as follows:

Parameter	Value
Motor efficiency, %	82
Power factor	0.81
Percent amperage, %	83

Finally, the actual current taken by the ESP motor is found from Eq. (5.33), using the nameplate amperage of 55.6 A:

$$I = 55.6(83/100) = 46 \text{ Amps.}$$

Finding the pump head including motor slip

To find the head developed by the submersible pump at the calculated rotational speed of the motor, the pumping rate at 3,500 RPM corresponding to the desired total liquid rate of $q'_l = 1,311$ bpd has to be found first from Eq. (5.34):

$$q_{3500} = 1,311(3,500/3,492) = 1,314 \text{ bpd.}$$

The head developed by one stage of the ESP at this rate and a speed of 3,500 RPM is found from the pump's head performance curve as 21.6 ft/stage. Because the total number of stages was selected as 221, the total head developed by the pump is found from Eq. (5.35) as

$$H_{\text{slip}} = 21.6 \cdot 221(3,492/3,500)^2 = 4,752 \text{ ft.}$$

Because this value is greater than the required *TDH*, calculated previously from inflow data as *TDH* = 4,516 ft, the ESP system will be capable to produce the desired liquid rate.

The rest of the design calculations is identical with those presented in Example 5.3.

5.5 GASSY WELL DESIGN

As discussed in Section 4.3.4, gas interference in ESP installations can be eliminated by **separating** the gas from the liquid before it enters the pump or by using special pumps that can move free gas along with the liquid phase. The second solution requires special pumps and complex calculations not suited for hand calculations; these are usually performed on the computer using specially developed software packages. This is the reason why the design of installations using rotary gas separators is described in the present section. **Rotary** gas separators, in contrast to conventional reverse-flow separators applied in wells with low gas production rates, allow the production of wells with substantial amounts of free gas, as shown in Fig. 4.25.

The design procedure for gassy wells basically follows the conventional design described in the previous section; therefore, only the main differences will be emphasized in the following.

5.5.1 Inflow and Free Gas Calculations

The well's producing bottomhole pressure (*FBHP*) is found identically to the method described in the conventional design; here, too, any of the available inflow performance curves can be used. Calculation of the *PIP*, however, is different because in the case of gassy wells a substantial amount of free gas enters the ESP. For this reason, it is very probable that **multiphase flow** takes place in the casing from the perforations up to the pump setting depth. This situation can be checked by comparing the well fluid's **bubblepoint** pressure to the *FBHP*. Now, instead of a constant liquid gradient used in single phase flow (see Eq. 5.2), one must find the pump's suction pressure (*PIP*) from multiphase flow calculations. The objective is to determine the multiphase **pressure drop** in the casing string from the perforations up to the pump setting depth for the well's current production parameters.

In the knowledge of *PIP* the calculations of the amount of free gas, the natural separation efficiency, etc., are accomplished following the methods discussed in Section 4.4.2. The steps of the required procedure are shortly repeated in the following.

1. First the "in situ" free gas volume entering the well is found from Eq. (4.15).
2. Using **Alhanati's** approximate method, the efficiency of natural gas separation in the annulus, η_n , is estimated from Eq. (4.16).
3. Assuming no rotary gas separator is present, the free gas volume that would enter the pump is equal to that calculated from Eq. (4.21).
4. The "in situ" liquid volumetric rate, q'_l , entering the pump is found from Eq. (4.22).
5. Now, using the **Turpin Correlation** [8], the function Φ is calculated from Eq. (4.30).
6. To determine whether the ESP can accommodate the free gas production or a rotary gas separator is needed the value of Φ is compared with unity. In gassy wells, it usually holds that $\Phi > 1$; and a gas **separator** is normally selected.

The separation efficiency of rotary separators is published by ESP manufacturers in charts similar to the one shown in Fig. 4.19. Such charts depict the percentage of the gas separated and vented through the casing annulus in function of the gas void fraction and the "in situ" liquid rate entering the pump. From the appropriate chart, or from other data, an estimated **efficiency**, η_{sep} , is found and the amount of gas not diverted to the casing annulus but **ingested** by the pump is determined.

The previously calculated ingested gas volume (see Eq. 4.21) must be corrected to reflect the **combined** effects of the **natural** gas separation in the annulus and the **rotary** gas separator. After expressing gas volume in bpd units, we get the amount of free gas entering the ESP pump:

$$q'_{ing} = \frac{q'_g}{5.61} \left(1 - \frac{\eta_n}{100}\right) \left(1 - \frac{\eta_{sep}}{100}\right) \quad (5.36)$$

where

q'_{ing} = gas volume ingested by pump, bpd

q'_g = free gas rate at pump suction, scf/day

η_n = natural gas separation efficiency, %, and

η_{sep} = rotary gas separator efficiency, %.

The **total** fluid volume to be handled by the pump is the sum of the “in situ” liquid and gas volumes, i.e.:

$$q'_{fluid} = q'_l + q'_{ing} \quad (5.37)$$

where

q'_{fluid} = fluid volume handled by pump, bpd

q'_l = liquid rate at pump suction, bpd, and

q'_{ing} = gas volume ingested by pump, bpd.

Again, the **Turpin Correlation** is used to check the operation of the pump. If a stable operation is indicated then the design can be continued, otherwise the use of special gas handling equipment is advised.

5.5.2 Calculation of Total Dynamic Head

One of the great differences between the cases with low and high gas production rates is that a **multiphase** mixture flows in a gassy well's tubing string. This is the reason why the constant fluid gradients used in single-phase cases cannot be used without a great loss of accuracy. A proper design, therefore, must include the calculation of **multiphase** pressure drops in the tubing string, and this requirement usually necessitates the use of pressure traverse curves or computer programs.

The basics of *TDH* calculations for gassy wells can be explained with the help of Fig. 5.1. The figure shows the pressure distributions in the tubing string and in the annulus of an ESP installation. As seen, the components of the necessary pump discharge pressure, p_d , are the wellhead pressure, *WHP*, and the multiphase vertical pressure drop in the well tubing. The latter is estimated from the proper pressure traverse curve for hand calculations or calculated using a computer program.

For accurate design calculations, available computer programs [17,18] offer the use of several vertical multiphase pressure drop prediction models for

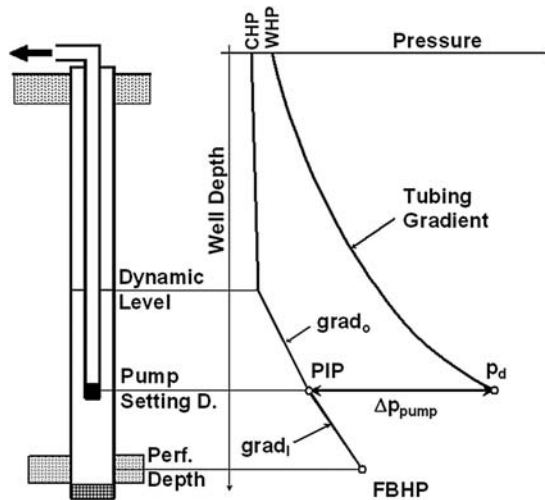


FIGURE 5.1 Pressure distributions in an electrical submersible pump installation. *CHP*, casinghead pressure; *FBHP*, flowing bottomhole pressure; *PIP*, pump intake pressure; *WHP*, wellhead pressure.

the calculation of the pressure traverse in the tubing string. The proper choice between them is very important because the different calculation models can produce very different results and their general accuracy is quite limited. Therefore, a basic understanding of the various models' (of which several are usually offered) behavior is required. Selection of the proper vertical multi-phase pressure drop calculation method is discussed by **Takacs** [19] and can help users to make the right choice.

Because the pump's suction pressure, *PIP*, is known, the pressure increase to be exerted by the pump, $\Delta p_{\text{pump}} = p_d - PIP$, is easily found. This value, however, if converted to head, gives exactly the pump's required **head**, i.e., the TDH value as found from the following formula:

$$TDH = \frac{144(p_d - PIP)}{\rho_l} \quad (5.38)$$

where

TDH = total dynamic head, ft

p_d = pump discharge pressure, psi

PIP = pump intake pressure, psi, and

ρ_l = average fluid density in the pump, lb/ft³.

An accurate calculation of the **average** fluid density requires the determination of the mixture's density in every stage of the centrifugal pump and this is a time-consuming task generally solved on the computer. For hand calculations, as shown in the following, this value is usually determined at pump **suction** conditions.

It should be noted that the **flowing GLR** to be used in vertical multiphase flow calculations for finding the pump discharge pressure should reflect the combined gas volumes removed from the wellstream by **natural** and **forced** separation. Thus, in addition to the gas dissolved at suction conditions, the **free** gas ingested by the pump should also be considered. The proper surface production *GLR* in scf/STB units is evaluated from

$$GLR = \frac{\frac{5.61 q'_{ing}}{B_g} + R_s q_o}{q_l} \quad (5.39)$$

where

GLR = production gas/liquid ratio, scf/STB

q'_{ing} = free gas rate ingested by pump (see Eq. 5.29), bpd

B_g = gas volume factor at suction conditions, ft³/scf

R_s = solution gas/oil ratio at suction conditions, scf/STB, and

q_o , q_l = oil and liquid production rates, STB/day.

The background for the derivation of the previous formula can be explained by studying the effect of free gas in an ESP. Theoretically, there are two possible **behaviors** for the free gas ingested by the ESP:

- it either goes back into **solution** due to the elevated pressure, or
- it gets **compressed** as free gas from stage to stage.

In practice, however, it is generally believed that most of the gas entering the pump behaves as **compressed** free gas because

- the **time** free gas stays in the stages is short for the gas to go back into solution, and
- the flow conditions in the pump stages (turbulence, high fluid velocities) work against gas solution.

These are the reasons why the gas volume ingested by the pump is treated as **compressed gas** that, after leaving the pump, may substantially decrease the multiphase pressure drop occurring in the tubing string.

5.5.3 The Rest of the Design Procedure

The selection of the ESP pump is identical to the previous design.

When selecting the electric motor, the system's total power requirement is the sum of the required pump power and the power needed to drive the rotary separator:

$$BHP_{system} = BHP_{pump} + BHP_{sep} \quad (5.40)$$

where

BHP_{system} = total power required, HP

BHP_{pump} = required pump power, HP, and

BHP_{sep} = required power to drive the separator, HP.

The selection procedure for the protector, the cable, the switchboard, and the transformers is accomplished according to the previously discussed design method.

Example 5.5

Select the main components of an ESP system running at a constant frequency of 60 Hz and installed on a gassy vertical well. Well data are detailed as follows.

Well physical data

Casing ID	6.331 in
Tubing ID	2.441 in
Tubing OD	2.875 in
Tubing new?	No
Perforation depth	6,500 ft
Pump depth	5,000 ft

Well performance data

WHP @ desired rate	100 psi
CHP @ desired rate	20 psi
Desired liquid rate	2,000 STB/day
Water cut	50%
SBHP	2,200 psi
PI	1.5 bpd/psi
GOR produced	300 scf/STB
Pump suction temperature	150 F

Fluid properties

Specific gravity of oil	0.86
Specific gravity of water	1.08
Specific gravity of gas	0.7

Power supply data

Available surface voltage	4,000 V

Solution

First the inflow conditions of the well are determined.

Well inflow calculations

At the desired rate, the flowing bottomhole pressure is found from Eq. (5.1) as

$$FBHP = 2,200 - 2,000 / 1.5 = 867 \text{ psi.}$$

Starting from this pressure at the well bottom and finding the pressure at the pump setting depth from multiphase pressure drop calculations in the well's annulus, we get the pump intake pressure that, in our case, was

$$PIP = 330 \text{ psi.}$$

The solution GOR at suction conditions is calculated from Eq. (4.24), where the oil's API gravity and the variable y are found as

$$^{\circ}API = 141.5/0.86 - 131.5 = 33.$$

$$y = 0.00091 \cdot 150 - 0.01215 \cdot 33 = -0.264, \text{ and}$$

$$R_s = 0.7 \left[330 / (18 \cdot 10^{-0.264}) \right]^{1.205} = 50 \text{ scf/bbl.}$$

To find the "in situ" gas volume, the gas volume factor, B_g , must be calculated, for which the knowledge of the deviation factor is necessary. The pseudocritical state parameters are calculated from Eqs. (4.27) and (4.28):

$$p_{pc} = 709 - 58.7 \cdot 0.7 = 668 \text{ psia,}$$

and

$$T_{pc} = 170.5 + 307.3 \cdot 0.7 = 386 \text{ R.}$$

The reduced parameters and the deviation factor from Eq. (4.26):

$$p_{pr} = 345/668 = 0.516,$$

and

$$T_{pr} = (150 + 460)/386 = 1.58.$$

$$Z = 1 - 3.52 \cdot 0.516 / 10^{0.9813 \cdot 1.58} + 0.274 \cdot 0.516^2 / 10^{0.8157 \cdot 1.58} = 0.95.$$

Now the gas volume factor can be determined from Eq. (4.25):

$$B_g = 0.0283 \cdot 0.95 (150 + 460) / 345 = 0.048 \text{ cu ft/scf.}$$

The volumetric rate of the free gas is found from Eq. (4.15):

$$q'_g = 1,000 (300 - 50) 0.048 = 12,000 \text{ ft}^3/\text{day}.$$

To find the liquid volumetric rate, the determination of the oil volume factor from Eq. (4.29) is as follows:

$$F = 50 (0.7/0.86)^{0.5} + 1.25 \cdot 150 = 232, \text{ and}$$

$$B_o = 0.972 + 1.47 \cdot 10^{-4} \cdot 232^{1.175} = 1.06 \text{ bbl/STB.}$$

The liquid volumetric rate handled by the pump is found from Eq. (4.22), and assuming $B_w = 1$:

$$q'_l = 1,000 \cdot 1.06 + 1,000 \cdot 1 = 2,061 \text{ bpd.}$$

Natural gas separation calculations

The cross-sectional area of the annulus equals

$$A = 0.0055(6.331^2 - 2.875^2) = 0.175 \text{ ft}^2.$$

From Eq. (4.17) the liquid superficial velocity in the annulus is calculated next with $WOR = 1$, and $B_w = 1$ as

$$v_{sl} = 6.5E^{-5} 2,000/0.175(1.06/2 + 1/2) = 0.77 \text{ ft/s}.$$

The density of the gas phase at suction conditions from Eq. (4.19):

$$\rho_g = 0.0764 0.7/0.048 = 1.1 \text{ lb/ft}^3.$$

The density of the liquid phase from Eq. (4.20):

$$\rho_l = 62.5 [0.86/(1.06/2) + 1.08/2] = 59.1 \text{ lb/ft}^3.$$

The terminal rising velocity of the gas bubbles can now be found, using Eq. (4.18):

$$v_b = 2^{1/2} \{ [0.0432(59.1 - 1.1)] / 59.1^2 \}^{1/4} = 0.54 \text{ ft/s}.$$

The natural gas separation efficiency of the annulus is calculated using Eq. (4.16):

$$\eta_n = [0.54/(0.54 + 0.77)]100 = 41\%.$$

The amount of gas entering the pump, as found from Eq. (4.21):

$$q'_{\text{ing}} = 12,000/5.61(1 - 0.41) = 1,262 \text{ bpd}.$$

The **Turpin** parameter Φ from Eq. (4.30):

$$\Phi = (2,000 1,262)/(3 330 2,061) = 1.24 > 1$$

Gas separation is required.

Rotary gas separation

Let us assume that a properly selected rotary gas separator achieves a separation efficiency of $\eta_{\text{sep}} = 60\%$. The total combined volume of gas ingested by the pump is determined from Eq. (5.36) as

$$q'_{\text{ing}} = 12,000/5.61(1 - 0.41)(1 - 0.60) = 504 \text{ bpd}$$

The total “in situ” fluid volume the ESP must handle is the sum of the liquid and gas rates, see Eq. (5.37):

$$q'_{\text{fluid}} = 2,061 + 504 = 2,565 \text{ bpd}.$$

The **Turpin** parameter Φ from Eq. (4.30):

$$\Phi = (2,000 504)/(3 330 2,061) = 0.49$$

Operation of the pump is stable.

TDH calculations

To find the pump's required discharge pressure, p_d , vertical multiphase pressure drop calculations must be performed in the tubing string from the wellhead to the pump setting depth. The gas/liquid ratio to be used is found from Eq. (5.39):

$$GLR = [(5.61\ 504)/0.048 + 50\ 1,000]/2,000 = 55\ \text{scf/STB}.$$

Using this value and computer calculations, the required pump discharge pressure was found as

$$p_d = 2,200\ \text{psi}.$$

The required *TDH* is now found from the pressure increase required and the liquid's density, as given by Eq. (5.38):

$$TDH = 144(2,200 - 330)/59.1 = 4,556\ \text{ft}.$$

Selection of the pump

Based on the casing size and the desired liquid rate, a 540 Series G2700-type pump is selected from Table C.1 in **Appendix C**.

Reading the performance curves of the selected pump at the desired total fluid rate of 2,565 bpd, the following specific parameters are determined:

Head/stage	44.5 ft/stage
BHP/stage	1.3 HP/stage
Maximum head/stage	52 ft/stage
Allowed shaft power	256 HP
Shaft diameter	0.875 in
Housing burst pressure	5,000 psi

The number of pump stages required to develop the *TDH* is determined from Eq. (5.9):

$$\text{Stages} = 4,556/44.5 = 102\ \text{stages}$$

From manufacturer's catalog, the nearest pump with 108 stages (Housing 90) is selected.

The selected pump is checked for mechanical strength, first by finding the maximum power occurring on its shaft from Eq. (5.11), using the specific gravity of the liquid. This is calculated from the liquid density at suction conditions as

$$\gamma_l = 59.1/62.4 = 0.947.$$

$$BHP_{\text{pump}} = 1.3\ 108\ 0.947(60/60)^3 = 133\ \text{HP}$$

which is less than the allowed power of 256 HP, the shaft is OK.

Checking the strength of the housing necessitates calculation of the maximum internal pressure from Eq. (5.13), using the pressure gradient of the liquid:

$$p_{\max} = 52\,108\,0.947\,0.433(60/60)^2 = 2,303\text{ psi}$$

which is less than the allowed burst pressure of the given pump housing.

Selection of the protector

The selected pump has floating impellers; therefore, the selection of the necessary protector is based on the downthrust developed on the pump shaft. The maximum thrust acting on the protector's thrust bearing is found from Eq. (5.14) as

$$F_{TB} = 0.785\,2,303\,0.875^2 = 1,384\text{ lbs.}$$

A 540 Series labyrinth-type protector is selected, with an allowed thrust loading of 9,500 lbs at the bottomhole temperature, as given by the manufacturer. Because this is greater than the actual load, F_{TB} , the selected protector checks OK. The protector shaft is rated at 637 HP, while the power required by the pump is 133 HP; therefore the selection is right.

Motor selection

Motors from the 562 series are considered and first the annular flow velocity of the produced liquid is checked from Eq. (5.18):

$$v_l = 0.0119\,2,061(6.331^2 - 5.62^2) = 2.88\text{ ft/s}$$

which is sufficient for the cooling of the motor.

The motor should develop a total power equal to the power required to drive the pump and the gas separator. Because the power needed by the gas separator is given as 5 HP, the system power is found from Eq. (5.40) as

$$BHP_{\text{system}} = 133 + 5 = 138\text{ HP.}$$

From Table D.1 in **Appendix D**, a motor with the following parameters is selected:

Power of selected motor	150 HP
Nameplate voltage	3,190 V
Nameplate current	28.5 A

The actual current demand of the selected motor is found from Eq. (5.21) as

$$I = 28.5\,138/150 = 26.2\text{ Amps.}$$

Selection of the power cable

Cable length is selected as 100 ft longer than the pump setting depth, i.e., 5,100 ft.

Based on the temperature conditions, a Redalene 4 kV cable with galvanized armor is selected. It can be determined from the ampacity chart of this cable type that all sizes can handle the current at the well’s bottomhole temperature.

For the selection of the optimum cable size, the following data are assumed:

Expected cable life	60 months
Prime interest rate	12%/year
Cost of electricity	5 c/kWh

From manufacturer’s catalog, the properties of the different cable sizes are determined as shown here:

Size (AWG)	Resistance (ohm/	Unit Price (\$/ft)
	1,000 ft)	
1	0.139	11.3
2	0.170	9.09
4	0.271	6.46
6	0.431	5.11

The monthly pay back cost, the power waste cost, and the total cost are calculated from Eqs. (5.22) to (5.26). Calculation results follow:

Cable Size (AWG)	Cable Cost (\$)	Monthly PayBack (\$/month)	Cable Resistance (ohms)	Power Loss (kW)	Cost of Power Loss (\$/month)	Total Cost (\$/month)
1	57,630	1,282	0.820	1.69	61	1,343
2	46,359	1,031	1.002	2.07	74	1,106
4	32,946	733	1.598	3.29	119	851
6	26,061	580	2.541	5.24	189	768

Based on the total monthly costs, the AWG **Size 6** gives the most economic solution. To check the startup voltage at the motor terminals, the ratio of the startup voltage to the nameplate voltage is found from Eq. (5.27):

$$U_{\text{start}}/U_{\text{np}} = (3,190 - 4 \cdot 26.2 \cdot 2.541)/3,190 = 0.92 = 92\%.$$

Thus, the ESP motor receives 92% of its nameplate voltage during startup, which is more than sufficient for a successful start.

Switchboard and transformer selection

The required surface voltage of the installation is the sum of the motor nameplate voltage and the cable voltage drop, as calculated from Eq. (5.28):

$$U_{\text{surf}} = 3,190 + 1.732 \cdot 2.541 \cdot 26.2 = 3,305 \text{ volts.}$$

Selection of the switchboard and the necessary step-down transformer is based on the kVA requirement of the installation, found from Eq. (5.29):

$$P_{\text{surf}} = 0.001732 \times 3,305 \times 26.2 = 150 \text{ kVA.}$$

Miscellaneous equipment

The downhole and surface miscellaneous pieces of equipment are selected from manufacturers' catalogs.

5.6 DESIGN OF A VARIABLE SPEED DRIVE INSTALLATION

5.6.1 Introduction

The use of a VSD unit to provide power to an ESP installation offers many advantages, the most important one being the ability to use the given equipment in a much **broader range** of liquid rates than made possible by the conventional 60 Hz power supply. The effects of design inaccuracies, improper information on well inflow conditions, etc., can easily be corrected if the ESP unit's speed can be adjusted to the desired level. As detailed in Section 4.7, the VSD unit can significantly stretch the application range of any ESP and is, therefore, a valuable tool in meeting the production objectives from wells with changing well conditions.

The installation design procedure introduced in this section can be used in gassy wells with **changing liquid rates** and is very similar to the gassy well design described in the previous section. As will be seen, most calculations are performed for two liquid rates, representing the well's expected **minimum** and **maximum** capacities. The steps of the design procedure follow those described in the previous sections and only the main differences will be fully described.

The necessary input data should include the expected **minimum** and **maximum** liquid flow rates from the given well. Most frequently, the reason of designing the ESP installation for two rates is the increase of the water-cut in the produced liquid when a decline in oil production is prevented by increasing the total liquid rate from the well.

The first steps of the design calculations are identical to those described in detail in Sections 5.5.1 and 5.5.2 and include

- the calculation of the well's **inflow** conditions,
- the determination, from multiphase flow calculations, of the *PIP*,
- the gas **separator** calculations covering natural and forced gas separation, and
- the calculation of the *TDH* requirement based on the multiphase pressure drop occurring in the tubing string.

These calculations are performed for each of the two expected liquid production rates, followed by the equipment selection procedure described in the following.

5.6.2 Pump Selection for Variable Speed Drive Service

At this point two liquid flow rates valid at pump suction conditions are known along with the corresponding two *TDH* values required to lift them to the surface. Based on these data, pump selection is accomplished by utilizing the **variable speed** performance curves of the pumps considered. (For an example see Fig. 4.39.) The **type** of the required ESP is selected to meet the following requirements:

- it should fit into the casing string, and
- the required minimum and maximum liquid rates should fall inside the recommended pumping rate range of the pump.

5.6.2.1 Driving Frequency and the Number of Stages

The maximum driving frequency and the required number of pump stages are determined for the **maximum** expected liquid rate as detailed in the following in conjunction with Fig. 5.2.

Using the head performance curves of the selected pump type at different frequencies, find the frequency where the pump produces the **maximum** liquid rate while it operates at its BEP. For this, connect the pump's BEPs belonging to the different frequencies and intersect this curve with a vertical line drawn at the maximum liquid rate. Now draw a head curve between the existing ones going through the point of intersection. Find the interpolated frequency, $f_{\max}$, belonging to the new head curve. Then read the **head** developed by one stage (the vertical coordinate of the intersection) at this condition.

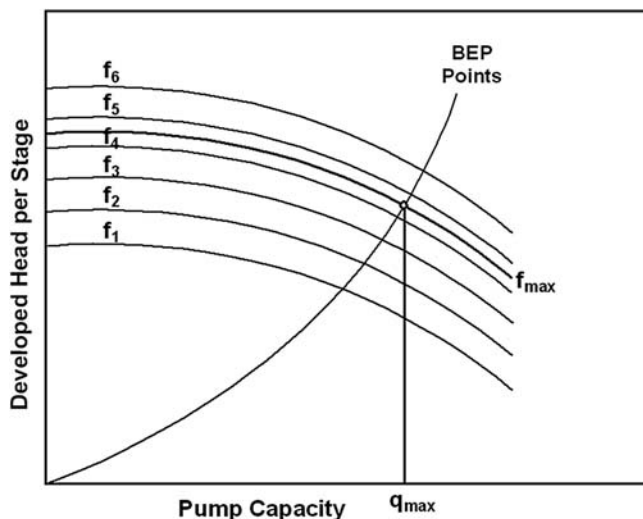


FIGURE 5.2 Determination of the maximum required frequency. *BEP*, best efficiency point.

Now the number of required pump stages is easily found from the following formula:

$$Stages = \frac{TDH_{max}}{head/stage} \quad (5.41)$$

where

$Stages$ = required number of pump stages, —
 TDH_{max} = required TDH at the maximum liquid rate, ft, and
 $head/stage$ = pump head at the maximum liquid rate, ft/stage.

A pump **housing** with the next larger number of stages is selected and the pump's mechanical **strength** is checked as described in [Section 5.3.3.4](#). When checking the strength of the pump shaft, the shaft rating is found from the allowed power at 60 Hz and the maximum driving frequency (see Eq. 4.40):

$$HP_{shaft} = HP_{shaft\ 60} \left(\frac{f_{max}}{60} \right) \quad (5.42)$$

where

HP_{shaft} = pump shaft rating at frequency f_{max} , HP
 $HP_{shaft\ 60}$ = pump shaft rating at 60 Hz, HP, and
 f_{max} = maximum driving frequency, Hz.

5.6.2.2 Checking Pump Operation at the Minimum Liquid Rate

The pump with the selected number of stages is **checked** at the required minimum liquid rate as follows. First the minimum required head per one stage is determined:

$$(head/stage)_{min} = \frac{TDH_{min}}{Stages} \quad (5.43)$$

where

$(head/stage)_{min}$ = minimum required head per stage, ft/stage
 TDH_{min} = required TDH for the minimum liquid rate, ft, and
 $Stages$ = number of pump stages selected, —.

By plotting the minimum required head per stage value versus the minimum liquid rate on the pump's variable frequency performance curve, one can determine the required driving **frequency**, f_{min} , needed to achieve that rate. For a proper design, this point, too, should fall very close to the curve connecting the BEPs of the selected pump.

5.6.3 Motor Selection

After the required protector is selected by following the procedure detailed in [Section 5.3.4](#), the ESP motor is selected. Its OD (series) is based on casing data and on providing a minimum liquid **flow velocity** of 1 ft/s past the motor.

The required **power** of the ESP motor should cover the powers necessary to drive (1) the pump, (2) the protector, and (3) the rotary gas separator. Because protector power is usually included in the pump requirements, only the remaining powers are calculated. The power required to drive the pump at the electrical frequency, $f_{\max}$, is found from the following formula:

$$BHP_{\text{pump}} = \left(\frac{BHP}{\text{stage}} \right) \text{Stages } \gamma_l \quad (5.44)$$

where

BHP_{pump} = required pump power at $f_{\max}$, HP

Stages = number of pump stages, —

BHP/stage = the power required per stage at $f_{\max}$, HP/stage, and

γ_l = specific gravity of the liquid pumped, —.

The power required to drive the **rotary** gas separator (a centrifugal machine just like the ESP) at the frequency $f_{\max}$ is found from its published power requirement at 60 Hz, $BHP_{\text{sep } 60}$, and the **affinity laws**, see Eq. (4.34):

$$BHP_{\text{sep}} = BHP_{\text{sep } 60} \left(\frac{f_{\max}}{60} \right) \quad (5.45)$$

where

BHP_{sep} = required separator power at $f_{\max}$, HP

$BHP_{\text{sep } 60}$ = required power at 60 Hz, HP, and

$f_{\max}$ = maximum electric frequency, Hz.

The ESP **system's** power requirement at the maximum driving frequency is thus found from

$$BHP_{\text{system}} = BHP_{\text{pump}} + BHP_{\text{sep}} \quad (5.46)$$

where

BHP_{system} = required system power at $f_{\max}$, HP

BHP_{pump} = required pump power at $f_{\max}$, HP, and

BHP_{sep} = required separator power at $f_{\max}$, HP.

The ESP motor should develop the same power at the same frequency, from which the required motor power at 60 Hz is found by using the performance relationships of electric motors, detailed in Section 4.7.4. According to Eq. (4.40), the following formula is to be used:

$$HP_{60} = BHP_{\text{system}} \left(\frac{60}{f_{\max}} \right) \quad (5.47)$$

where

HP_{60} = motor power at 60 Hz, HP

BHP_{system} = required system power at f_{max} , HP, and
 f_{max} = maximum electric frequency, Hz.

As done in the previous design procedures, a motor with a nameplate **rating** higher than this value is selected. The rest of the motor sizing is performed identically to the previous designs.

5.6.4 Switchboard and Transformers

After the selection of the power cable is accomplished with the same procedure as detailed for the previous design methods, the surface **voltage** and **power** requirements are found as follows.

Because the VSD unit keeps a **constant** ratio between its output voltage and its output frequency, the required motor **voltage** at the maximum frequency is calculated from Eq. (4.39):

$$U_{f \max} = U_{60} \left(\frac{f_{\max}}{60} \right) \quad (5.48)$$

where

$U_{f \max}$ = required motor voltage at $f_{\max}$, volts
 U_{60} = motor nameplate voltage at 60 Hz, volts, and
 $f_{\max}$ = maximum electric frequency, Hz.

After calculating the voltage drop in the cable, the **surface** voltage requirement at the maximum electric frequency is calculated and the power requirement is found from Eq. (5.29).

This concludes the design procedure for an ESP installation with VSD service.

Example 5.6

Select the main components of an ESP system to be installed on a gassy vertical well with changing inflow performance. Well data are given here.

Well physical data

Casing ID	6.331 in
Tubing ID	2.441 in
Tubing OD	2.875 in
Tubing new?	Yes
Perforation depth	6,500 ft
Pump depth	6,000 ft

Well performance data

WHP @ desired rates	100 psi
CHP @ desired rates	20 psi
minimum desired liquid rate	2,000 STB/day

@ a water cut of	50%
maximum desired	2,500 STB/day
liquid rate	
@ a water cut of	60%
SBHP	2,200 psi
PI	1.5 bpd/psi
GOR produced	300 scf/STB
BHT	150 F

Fluid properties

Specific gravity of oil	0.86
Specific gravity of water	1.08
Specific gravity of gas	0.7

Power supply data

Available surface voltage	4,500 V

Solution

The inflow conditions of the well are determined for the two liquid rates desired.

Well Inflow Calculations for the Minimum Rate.

At the minimum rate, the flowing bottomhole pressure is found from Eq. (5.1) as

$$FBHP = 2,200 - 2,000 / 1.5 = 867 \text{ psi.}$$

Starting from this pressure at the well bottom and finding the pressure at the pump setting depth from multiphase pressure drop calculations in the well's annulus, we get the pump intake pressure as

$$PIP = 660 \text{ psi.}$$

The solution GOR at suction conditions is calculated from Eq. (4.24), where the oil's API gravity and the variable y are found as

$$^{\circ}API = 141.5 / 0.86 - 131.5 = 33.$$

$$y = 0.00091 \cdot 150 - 0.01215 \cdot 33 = -0.264, \text{ and}$$

$$R_s = 0.7 \left[660 / (18 \cdot 10^{-0.264}) \right]^{1.205} = 116 \text{ scf/bbl}$$

To find the "in situ" gas volume, the gas volume factor, B_g , must be calculated, for which the knowledge of the deviation factor is necessary. The

deviation factor is a function of the reduced state parameters calculated from Eqs. (4.27) and (4.28):

$$p_{pc} = 709 - 58.7 \cdot 0.7 = 668 \text{ psia, and } T_{pc} = 170.5 + 307.3 \cdot 0.7 = 386 \text{ R.}$$

The reduced parameters and the deviation factor from Eq. (4.26):

$$p_{pr} = 674.7/668 = 1.01, \quad \text{and} \quad T_{pr} = (150 + 460)/386 = 1.58.$$

$$Z = 1 - 3.52 \cdot 1.01/10^{0.9813 \cdot 1.58} + 0.274 \cdot 0.516^2/10^{0.8157 \cdot 1.58} = 0.915.$$

Now the gas volume factor can be determined from Eq. (4.25):

$$B_g = 0.0283 \cdot 0.915(150 + 460)/674.7 = 0.023 \text{ ft}^3/\text{scf}.$$

The volumetric rate of the free gas is found from Eq. (4.15):

$$q'_g = 1,000(300 - 116)0.023 = 4,232 \text{ ft}^3/\text{day}.$$

To find the liquid volumetric rate, the determination of the oil volume factor from Eq. (4.29) is as follows:

$$F = 116(0.7/0.86)^{0.5} + 1.25 \cdot 150 = 292, \text{ and}$$

$$B_o = 0.972 + 1.47 \cdot 10^{-4} 292^{1.175} = 1.088 \text{ bbl/STB}.$$

The liquid volumetric rate handled by the pump is found from Eq. (4.22), and assuming $B_w = 1$:

$$q'_l = 1,000 \cdot 1.088 + 1,000 \cdot 1 = 2,088 \text{ bpd}.$$

Well inflow calculations for the maximum rate

At the maximum rate, the flowing bottomhole pressure is found from Eq. (5.1) as

$$FBHP = 2,200 - 2,500/1.5 = 533 \text{ psi}.$$

After calculating the pressure at the pump setting depth from multiphase pressure drop calculations, the pump intake pressure is

$$PIP = 330 \text{ psi}.$$

The solution GOR at suction conditions is calculated from Eq. (4.24):

$$R_s = 0.7 \left[330 / (18 \cdot 10^{-0.264}) \right]^{1.205} = 50 \text{ scf/bbl}.$$

The reduced state parameters are the same as before and the deviation factor from Eq. (4.26):

$$p_{pr} = 344.7/668 = 0.516, \quad \text{and} \quad T_{pr} = (150 + 460)/386 = 1.58.$$

$$Z = 1 - 3.52 \cdot 0.516/10^{0.9813 \cdot 1.58} + 0.274 \cdot 0.516^2/10^{0.8157 \cdot 1.58} = 0.95.$$

Now the gas volume factor can be determined from Eq. (4.25):

$$B_g = 0.0283 \cdot 0.95(150 + 460)/345 = 0.048 \text{ cu ft/scf.}$$

The volumetric rate of the free gas is found from Eq. (4.15) as

$$q'_g = 1,000 (300 - 50) 0.048 = 12,000 \text{ ft}^3/\text{day.}$$

To find the liquid volumetric rate, the determination of the oil volume factor from Eq. (4.29) follows:

$$F = 50(0.7/0.86)^{0.5} + 1.25 \cdot 150 = 232, \text{ and}$$

$$B_o = 0.972 + 1.47 \cdot 10^{-4} \cdot 232^{1.175} = 1.06 \text{ bbl/STB.}$$

The liquid volumetric rate handled by the pump is found from Eq. (4.22), and assuming $B_w = 1$:

$$q'_l = 1,000 \cdot 1.06 + 1,000 \cdot 1 = 2,061 \text{ bpd.}$$

Natural gas separation calculations for the minimum rate

The cross-sectional area of the annulus equals

$$A = 0.0055(6.331^2 - 2.875^2) = 0.175 \text{ ft}^2.$$

From Eq. (4.17) the liquid superficial velocity in the annulus is calculated with $WOR = 1$, and $B_w = 1$ as

$$v_{sl} = 6.5E - 5 \cdot 2,000/0.175(1.088/2 + 1/2) = 0.78 \text{ ft/s.}$$

The density of the gas phase at suction conditions from Eq. (4.19):

$$\rho_g = 0.0764 \cdot 0.7/0.023 = 2.3 \text{ lb/ft}^3.$$

The density of the liquid phase from Eq. (4.20):

$$\rho_l = 62.5 [0.86/(1.088 \cdot 2) + 1.08/2] = 58.5 \text{ lb/ft}^3.$$

The terminal rising velocity of the gas bubbles can now be found, using Eq. (4.18):

$$v_b = 2^{1/2} \{ [0.04 \cdot 32(58.5 - 2.3)] / 58.5^2 \}^{1/4} = 0.539 \text{ ft/s.}$$

The natural gas separation efficiency of the annulus is calculated using Eq. (4.16):

$$\eta_n = [0.539/(0.539 + 0.78)]100 = 41\%.$$

The amount of gas entering the pump, as found from Eq. (4.21):

$$q'_{\text{ing}} = 4232/5.61(1 - 0.41) = 454 \text{ bpd.}$$

The **Turpin** parameter Φ from Eq. (4.30):

$$\Phi = (2,000\,454)/(3\,660\,2,088) = 0.22 < 1$$

No gas separation is required.

Natural gas separation calculations for the maximum rate

From Eq. (4.17) the liquid superficial velocity in the annulus is calculated with $WOR = 1.5$, and $B_w = 1$ as

$$v_{sl} = 6.5E - 5\,2,500/0.175(1061/2.5 + 1/2.5) = 0.95 \text{ ft/s.}$$

The density of the gas phase at suction conditions from Eq. (4.19):

$$\rho_g = 0.0764\,0.7/0.048 = 1.1 \text{ lb/ft}^3.$$

The density of the liquid phase from Eq. (4.20):

$$\rho_l = 62.5[0.86/(1.061\,2.5) + 1.08/2.5] = 60.8 \text{ lb/ft}^3.$$

The terminal rising velocity of the gas bubbles can now be found, using Eq. (4.18):

$$v_b = 2^{1/2} \{ [0.04\,32(60.8-1.1)]/60.8^2 \}^{1/4} = 0.536 \text{ ft/s.}$$

The natural gas separation efficiency of the annulus is calculated using Eq. (4.16):

$$\eta_n = [0.536/(0.536 + 0.95)]100 = 36\%.$$

The amount of gas entering the pump, as found from Eq. (4.21):

$$q'_{\text{ing}} = 12,000/5.61\,(1 - 0.36) = 1,368 \text{ bpd.}$$

The **Turpin** parameter Φ from Eq. (4.30):

$$\Phi = (2,000\,1,368)/(3\,330\,2,561) = 1.08 > 1$$

Gas separation is required.

Rotary gas separation for the minimum rate

Let us assume that a properly selected rotary gas separator achieves a separation efficiency of $\eta_{\text{sep}} = 60\%$. The total combined volume of gas ingested by the pump is determined from Eq. (5.36) as

$$q'_{\text{ing}} = 4,232/5.61(1 - 0.41)\,(1 - 0.60) = 182 \text{ bpd}$$

The total “in situ” fluid volume the ESP must handle is the sum of the liquid and gas rates, see Eq. (5.37):

$$q'_{\text{fluid}} = 2,088 + 182 = 2,270 \text{ bpd.}$$

The **Turpin** parameter Φ from Eq. (4.30):

$$\Phi = (2,000\ 182)/(3\ 600\ 2,270) = 0.08, \text{ operation of the pump is stable.}$$

Rotary gas separation for the maximum rate

If a rotary gas separator achieves a separation efficiency of $\eta_{\text{sep}} = 60\%$, then the total combined volume of gas ingested by the pump is determined from Eq. (5.36) as

$$q'_{\text{ing}} = 12,000/5.61 (1 - 0.36) (1 - 0.60) = 547 \text{ bpd}$$

The total “in situ” fluid volume the ESP must handle is the sum of the liquid and gas rates, see Eq. (5.37):

$$q'_{\text{fluid}} = 2,561 + 547 = 3,108 \text{ bpd.}$$

The Turpin parameter Φ from Eq. (4.30):

$$\Phi = (2,000\ 547)/(3\ 330\ 3,108) = 0.36 < 1$$

Operation of the pump is stable.

TDH calculations for the minimum rate

To find the pump’s required discharge pressure, p_d , vertical multiphase pressure drop calculations must be performed in the tubing string from the wellhead to the pump setting depth. The gas/liquid ratio to be used is found from Eq. (5.39) as

$$GLR = [(5.61\ 182)/0.023 + 116\ 1,000]/2,000 = 80 \text{ scf/STB.}$$

Using this value and computer calculations the required pump discharge pressure was found as

$$p_d = 2,400 \text{ psi.}$$

The required *TDH* is now found from the pressure increase required and the liquid’s density, as given by Eq. (5.38):

$$TDH = 144(2,400 - 660)/58.5 = 4,283 \text{ ft.}$$

TDH calculations for the maximum rate

Vertical multiphase pressure drop calculations must be performed in the tubing string to find the pump’s required discharge pressure, p_d . The gas/liquid ratio to be used is found from Eq. (5.39) as

$$GLR = [(5.61\ 547)/0.048 + 50\ 2,500\ 0.4]/2,500 = 46 \text{ scf/STB.}$$

Using this value and computer calculations the required pump discharge pressure was found as

$$p_d = 2,600 \text{ psi.}$$

The required TDH is now found from the pressure increase required and the liquid's density, as given by Eq. (5.38):

$$TDH = 144(2,600 - 330)/60.8 = 5,376 \text{ ft.}$$

Selection of the pump for the maximum rate

Based on the casing size and the desired liquid rate range, a 540 Series G2700-type pump is selected from published variable speed performance curves.

At the desired total fluid rate of 3,108 bpd, reading the performance curve of the selected pump, and performing the operations described in conjunction with Fig. 5.2 the following specific parameters are determined:

Required frequency	68 Hz
Head/stage	54 ft/stage
BHP/stage	1.88 HP/stage
Maximum head/stage	68.5 ft/stage
Allowed shaft power @60 Hz	256 HP
Shaft diameter	0.875 in
Housing burst pressure	5,000 psi

The number of pump stages required to develop the TDH is determined from Eq. (5.41):

$$\text{Stages} = 5,376/54 = 100 \text{ stages.}$$

From manufacturer's catalog, the nearest pump with 108 stages (Housing 90) is selected.

The required pump power at the maximum frequency of 68 Hz is found from Eq. (5.44), using the specific gravity of the liquid at suction conditions:

$$BHP_{\text{pump}} = 1.88 \cdot 108 \cdot 60.8/62.4 = 198 \text{ HP}$$

The strength of the pump shaft is checked at the maximum frequency of 68 Hz from Eq. (5.42):

$HP_{\text{shaft}} = 256 \cdot 68/60 = 290 \text{ HP}$, much greater than the required pump power, the shaft is OK.

Checking the strength of the housing necessitates calculation of the maximum internal pressure from Eq. (5.13), using the pressure gradient of the liquid:

$$p_{\text{max}} = 68.5 \cdot 108 \cdot 0.433 \cdot 60.8/62.4(60/60)^2 = 3,121 \text{ psi}$$

which is less than the allowed burst pressure of the given pump housing.

Checking pump operation at the minimum rate

The selected pump is checked at the minimum required liquid rate by calculating the required head per stage from Eq. (5.43):

$$\text{Minimum head/stage} = 4,283/108 = 40 \text{ ft/stage.}$$

Fig. 5.3 contains the variable speed performance curve of the selected pump where the operating points for the minimum and maximum fluid rates are indicated. As seen, production of the minimum required rate is accomplished at a frequency of 56 Hz.

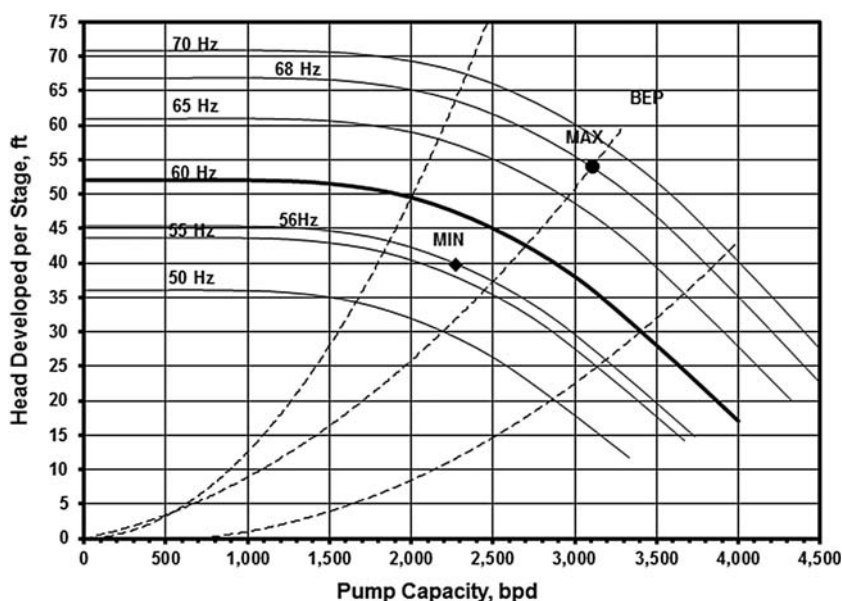


FIGURE 5.3 Illustration of variable speed drive design for Example 5.6. BEP, best efficiency point.

Selection of the protector

Because the selected pump has floating impellers, the selection of the necessary protector is based on the downthrust developed on the pump shaft. The maximum thrust acting on the protector's thrust bearing is found from Eq. (5.14) as

$$F_{TB} = 0.785 \cdot 3,121 \cdot 0.875^2 = 1,875 \text{ lbs.}$$

A 540 Series labyrinth-type protector is selected with an allowed thrust loading of 9,500 lbs at the bottomhole temperature. Because this is greater than the actual load, F_{TB} , the selected protector checks OK. The protector shaft is rated at 637 HP, while the power required by the pump is only 198 HP; therefore the selection is right.

Motor selection

Motors from the 562 series are considered and first the annular flow velocity of the produced liquid is checked from Eq. (5.18), using the maximum liquid flow rate:

$$v_l = 0.0119 \, 2,561 / (6.331^2 - 5.62^2) = 3.59 \text{ ft/s}$$

which is sufficient for the cooling of the motor.

In addition to the pump power of 198 HP, the motor must drive the rotary gas separator. The power requirement of the separator is 5 HP at 60 Hz, according to manufacturer data; at the maximum frequency of 68 Hz, it uses a higher power found from Eq. (5.45) as

$$BHP_{sep} = 5(68/60)^3 \, 60.8/62.4 = 7.1 \text{ HP.}$$

The total system power at 68 Hz operation is found from Eq. (5.46):

$$BHP_{system} = 198 + 7.1 = 205 \text{ HP.}$$

The equivalent power at 60 Hz is calculated from the **affinity laws**:

$$BHP_{system \, 60} = 205(60/68)^3 = 141 \text{ HP.}$$

The rating of the motor developing this power at the frequency of 68 Hz is calculated from Eq. (5.47):

$$HP_{60} = 205 \, 60/68 = 181 \text{ HP.}$$

From Table D.1 in Appendix D, a motor with the following parameters is selected:

Power of selected motor	210 HP
Nameplate voltage	3,720 V
Nameplate current	34 A

It can be verified here whether the motor is overloaded at the maximum frequency of 68 Hz. For this the maximum allowed frequency found from Eq. (4.41) is compared with the required value as follows:

$$f_{max} = 60(210/141)^{0.5} = 73 \text{ Hz}$$

which is greater than 68 Hz; therefore the operation is safe.

The actual current demand of the selected motor is found from Eq. (5.21) as

$$I = 34 \, 181/210 = 29.3 \text{ Amps.}$$

Selection of the power cable

Cable length is selected as 100 ft longer than the pump setting depth, i.e., 6,100 ft.

A Redalene 4 kV cable with galvanized armor is selected. It can be determined from the ampacity chart of this cable type that all sizes can handle the current at the well’s bottomhole temperature.

For the selection of the optimum cable size, the following data are assumed:

Expected cable life	60 months
Prime interest rate	12%/year
Cost of electricity	5 c/kWh

From manufacturer’s catalog, the properties of the different cable sizes are determined as given in the next table:

Size (AWG)	Resistance (ohm/	Unit Price (\$/ft)
	1,000 ft)	
1	0.139	11.3
2	0.170	9.09
4	0.271	6.46
6	0.431	5.11

The monthly pay back cost, the power waste cost, and the total cost are calculated from Eqs. (5.22) to (5.26) and are given in the following.

Cable	Cable	Monthly	Cable	Power		
Size (AWG)	Cost (\$)	PayBack (\$/month)	Resistance (ohms)	Loss (kW)	Cost of Power Loss (\$/month)	Total Cost (\$/month)
1	68,930	1,533	0.980	2.51	91	1,624
2	55,449	1,233	1.199	3.08	111	1,344
4	39,406	877	1.911	4.90	176	1,053
6	31,171	693	3.040	7.80	281	974

Based on the total monthly costs, the AWG **Size 6** gives the most economic solution. To check the startup voltage at the motor terminals, the ratio of the startup voltage to the nameplate voltage is found from Eq. (5.27):

$$U_{\text{start}}/U_{\text{np}} = (3,720 - 4 \cdot 29.2 \cdot 3.04)/3,720 = 0.90 = 90\%.$$

Thus, the ESP motor receives 90% of its nameplate voltage during startup, which is more than sufficient for a successful start.

Switchboard and transformer selection

The motor’s required voltage at the operating frequency of 68 Hz is found from Eq. (5.48):

$$U = 3,720 \cdot 68/60 = 4,206 \text{ volts.}$$

The required surface voltage of the installation is the sum of the motor voltage and the cable voltage drop, as calculated from Eq. (5.28):

$$U_{\text{surf}} = 4,206 + 1.732 \cdot 3.04 \cdot 29.2 = 4,360 \text{ V}$$

Selection of the switchboard and the necessary step-down transformer is based on the kVA requirement of the installation, found from Eq. (5.29):

$$P_{\text{surf}} = 0.001732 \cdot 4,360 \cdot 29.2 = 220 \text{ kVA.}$$

Miscellaneous equipment

Downhole and surface miscellaneous pieces of equipment are selected from manufacturers' catalogs.

5.7 DESIGN OF A PERMANENT MAGNET MOTOR INSTALLATION

5.7.1 Design Considerations

This section details the selection of ESP equipment for a well producing negligible amounts of gas and using a PMM. The design procedure and equipment data of a major PMM manufacturer are used for illustration purposes [20].

The equipment selection process follows the basic steps described in Section 5.3 with only the pump and motor selection being different. Because of this, well inflow and TDH calculations are not repeated here and only the description of the pump and motor design is detailed in the following.

5.7.1.1 Selection of the Pump

As before, pump **size** (series) is selected based on the casing size; the selection of the pump **type** is, however, different and fully utilizes the advantages of variable speed operation. These advantageous features are the following:

- Running the pump at greater than normal pumping speeds substantially increases the **head** capacity of the pump and at the same time reduces the required number of pump stages.
- The possibility of the wide variation of pumping speeds makes it possible to always use the pump at its BEP.

Based on the above principles, the pump type selection is done by trial and error. The performance parameters at the BEP of the firstly chosen pump are recorded, including the liquid rate q_{BEP} and the rated speed, N_{base} . In the knowledge of the desired production rate q_l the pumping speed N required to reach that rate is found from the **affinity laws**:

$$N = N_{\text{base}} \left(\frac{q_l}{q_{\text{BEP}}} \right) \quad (5.49)$$

where

q_l = desired pumping rate, bpd

q_{BEP} = pumping rate at BEP and N_{base} , bpd, and

N , N_{base} = required and base speeds, RPM.

After checking the calculated speed one can decide whether to keep the first pump or to try a different one; the selected speed must be within the operating limits of the available pumps and motors. The finally selected speed ensures that the pump is going to operate at its BEP; this is an important improvement over the traditional pump selection process. As shown in [Section 5.3](#), normally the pump is not operated at its best efficiency if the conventional ESP design is followed.

The pump type being determined, its operating parameters at the BEP are read from performance curves valid at the published speed, N_{base} . These include the head/stage, BHP/stage, shut-in head/stage, efficiency, and allowed shaft power. Now the performance of the pump is corrected to the required pumping speed, N , previously calculated; the **affinity law** formulas reproduced here are used (pump efficiency does not change):

$$head/stage = head/stage_{base} \left(\frac{N}{N_{base}} \right)^2 \quad (5.50)$$

$$BHP/stage = BHP/stage_{base} \left(\frac{N}{N_{base}} \right)^3 \quad (5.51)$$

$$HP_{shaft} = HP_{shaft\ base} \left(\frac{N}{N_{base}} \right) \quad (5.52)$$

where

$head/stage$ = pump head at speed N , ft

$head/stage_{base}$ = pump head at speed N_{base} , ft

$BHP/stage$ = pump power at speed N , HP

$BHP/stage_{base}$ = pump power at speed N_{base} , HP

HP_{shaft} = pump shaft rating at speed N , HP

$HP_{shaft\ base}$ = pump shaft rating at N_{base} , HP, and

N , N_{base} = required and base speeds, RPM.

Using the pump performance parameters at the required pumping speed, N , the necessary number of pump **stages** is easily found based on the previously established *TDH* value:

$$Stages = \frac{TDH}{head/stage} \quad (5.53)$$

where

$Stages$ = required number of stages, —

TDH = total dynamic head, ft, and

$head/stage$ = head developed by one stage of the selected pump, ft.

The actual number of **stages** is found from data of **housings** offered by the manufacturer; checking of the mechanical strength of the pump is done according to [Section 5.3.3.4](#) not repeated here.

Normally, one does not find a pump with exactly the **required** number of stages from the manufacturer and a pump with the **nearest** greater stage number should be selected. If **dummy** stages are used to fill up the pump housing, then no corrections are needed and the calculated number of stages is used in subsequent calculations. But if a different number of stages is selected than the design dictates pump operation will be affected and the pump will produce a rate different from the desired one. In such cases correction of pump speed is required so that the pump produces the desired rate against the *TDH*.

The procedure to find the required speed is illustrated in [Fig. 5.4](#). First the head performance curves at the original speed are constructed for the calculated and the selected (in this case greater than the calculated) number of stages. The latter involves increasing of the head values of the original curve by a factor representing the increase in stage numbers. As can be seen, the pump would produce a rate greater than desired against the *TDH*, so its speed must be reduced. The reduced speed is the one at which the pump performance curve will go through the desired operating point.

To find the corrected speed, first an “**affine parabola**” defined in [Section 2.3.2.6](#) is constructed through the desired operating point (indicated by a small square) using Eq. (2.21). This parabola, shown in dashed line, connects corresponding operating points (rate-head) at different speeds. Its intersection with the head curve of the selected stage at the original speed is indicated by a

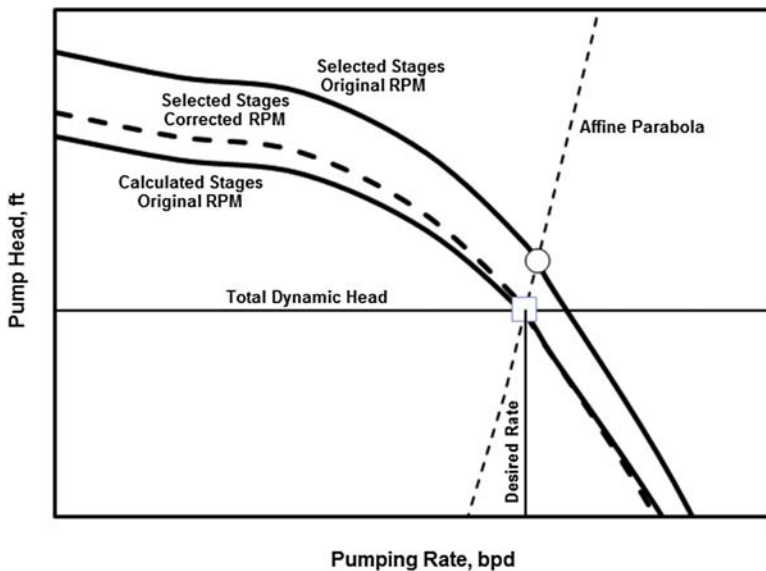


FIGURE 5.4 Finding the proper pumping speed for selected number of stages.

small circle. The liquid rate read at this point allows the calculation of the **corrected** speed, based on the affinity laws as follows:

$$N_{\text{corr}} = N_{\text{orig}} \left(\frac{q_l}{Q} \right) \quad (5.54)$$

where

$N_{\text{corr}}, N_{\text{orig}}$ = corrected and original speeds, RPM

q_l = desired pumping rate, bpd, and

Q = liquid rate read at the intersection, bpd.

The speed just calculated, N_{corr} , moves the operating point of the pump with the selected number of stages to the desired rate. The pump will produce the desired liquid rate, q_l , against the *TDH* required to lift the liquid to the surface. The efficiency of the pump at this speed can be determined easily and is normally quite close to that valid at the BEP at this speed.

The power required to run the pump is calculated after the power requirement of one stage is corrected to the new speed, N_{corr} , as follows:

$$BHP_{\text{pump}} = \frac{BHP}{\text{stage}} \text{ Stages } \gamma_l \quad (5.55)$$

where

BHP_{pump} = required pump power, HP

BHP/stage = power to drive one stage at speed N_{corr} , HP/stage

Stages = the selected number of pump stages, —, and

γ_l = specific gravity of the produced liquid, —.

5.7.1.2 Selection of the Permanent Magnet Motor

Motor size (series) is selected based on the well's casing dimension. The manufacturer recommends increasing the ESP system's power by 10% to cover for increased power requirements at startup and unusual conditions. The required motor power, therefore, is 1.1 BHP_{pump} and a PMM with a close enough power valid at a speed near the required pumping speed, N_{base} , is selected. Motor power at actual speed, N_{corr} , is checked using the next formula:

$$HP_{\text{motor}} = HP_{\text{base}} \left(\frac{N_{\text{corr}}}{N_{\text{base}}} \right) \quad (5.56)$$

where

HP_{motor} = motor nameplate power at speed N , HP

HP_{base} = motor nameplate power at N_{base} , HP, and

$N_{\text{corr}}, N_{\text{base}}$ = corrected and base motor speeds, RPM.

The calculated motor power should be just above the power requirement of the system; otherwise a different motor is chosen. Performance parameters of the finally selected motor are collected from manufacturer data including nameplate voltage, current, efficiency, and power factor.

Actual performance of the motor at the required motor speed is determined next. Motor current is kept constant to ensure saturation of the stator coils and the required voltage is found from the next formula:

$$U_{\text{motor}} = U_{np} \left(\frac{N_{\text{corr}}}{N_{np}} \right) \quad (5.57)$$

where

U_{motor} = motor voltage at speed N , volts

U_{np} = motor voltage at N_{np} , volts, and

N_{corr} , N_{np} = corrected and nameplate motor speeds, RPM.

Checking the liquid velocity around the motor for cooling is performed just like in the conventional design; PMMs usually require much lower velocities because of their lower heat generation.

The rest of the design procedure (selecting the protector, electric cable, etc.) is identical to the conventional design and is not described here; the reader is advised to study [Section 5.3](#).

Example 5.7

Using the same data as in [Example 5.3](#), perform the basic design of the ESP installation if a PMM is used.

Solution

Well inflow and *TDH* calculations are identical to those elaborated in [Example 5.3](#) and are not repeated here; main results of that example are listed here:

Liquid specific gravity = 0.986; required *TDH* = 4,516 ft; required pumping rate = 1,310 bpd.

The remaining calculation steps are detailed in the following.

Selection of the pump

In this case, based on the casing size a 406 Series **NHV (790–1000)H** pump was selected from a leading manufacturer's catalog. This pump's performance data at its BEP at a speed of 3,490 SPM are as follows:

Pumping rate	760 bpd
Head/stage	19.4 ft/stage
BHP/stage	0.156 HP/stage
Maximum head/stage	23.2 ft/stage
Pump efficiency	69%
Allowed shaft power	105 HP
Shaft diameter	0.55 in
Housing burst pressure	5,547 psi

Based on these data and the required pumping rate of 1,310 bpd, one can calculate the pump speed needed to reach that rate from [Eq. \(5.49\)](#):

$$N = 3,490 (1,310/760) = 6,016 \text{ RPM.}$$

Because this is an acceptable speed for PMMs, the performance of the pump at this speed is found from Eqs. (5.50)–(5.52); calculated pump performance at its BEP at 6,016 RPM is the following:

Head/stage	57.7 ft/stage
BHP/stage	0.80 HP/stage
Maximum head/stage	69.0 ft/stage
Pump efficiency	69%
Allowed shaft power	181 HP

The number of pump stages required to develop the *TDH* value is determined from Eq. (5.53):

$$\text{Stages} = 4,516/57.7 = 78 \text{ stages.}$$

From manufacturer's catalog, the nearest pump with 84 stages (Housing 6) is selected.

Calculations of the corrected pump speed according to Fig. 5.4 resulted in $N_{\text{corr}} = 5,859$ RPM, the pump's performance at this speed is the following.

Head/stage	54.7 ft/stage
BHP/stage	0.74 HP/stage
Maximum head/stage	65.4 ft/stage
Pump efficiency	69%
Allowed shaft power	176 HP

The selected pump must be checked for mechanical strength, first by finding the maximum power occurring on its shaft from Eq. (5.55):

$$BHP_{\text{pump}} = 0.74 \ 84 \ 0.986 = 61 \text{ HP}$$

which is less than the allowed power of 176 HP, the shaft is OK.

Checking the strength of the housing necessitates calculation of the maximum internal pressure from Eq. (5.13):

$$p_{\text{max}} = 65.4 \ 84 \ 0.433 \ 0.986 = 2,345 \text{ psi}$$

which is less than the allowed burst pressure of the given pump housing.

Motor selection

The required system power is 10% more than the pump requirement and equals $61 \ 1.1 = 67$ HP. From manufacturer data, an **N406 PM84 2020V 6000RPM** motor of the 406 series was selected. The motor's basic parameters at its nameplate speed of 6,000 RPM and a loading of 86% are as follows:

Motor power	80 HP
Nameplate voltage	2,020 V
Nameplate current	19.8 A
Efficiency	93%
Power factor	0.95
Minimum fluid velocity	0.49 ft/s

The performance of the motor at the corrected pumping speed of $N_{\text{corr}} = 5,859$ RPM is found from the affinity laws. Motor power is calculated from Eq. (5.56) as

$$HP_{\text{motor}} = 80 \cdot 5,859 / 6,000 = 78 \text{ HP}$$

This matches the required power, motor selection is right.

There are no other significant changes in this example case, but normally motor performance may change. Because motor current is kept constant to ensure magnetic saturation of the stator coils, the only parameter needed correction is the nameplate voltage, found from Eq. (4.39) as

$$U_{\text{motor}} = 2,020 \cdot 5,859 / 6,000 = 1,973 \text{ V.}$$

The annular flow velocity of the produced liquid around the motor is checked from Eq. (5.18):

$$v_l = 0.0119 \cdot 1,310 / (6.01^2 - 4.06^2) = 0.79 \text{ ft/s}$$

which is sufficient for the cooling of the motor.

The actual current demand of the selected motor is found from Eq. (5.21) as

$$I = 19.867 / 78 = 17 \text{ Amps.}$$

Selection of the power cable

Cable data are similar to those presented in Example 5.3, only calculation results different are presented here. An economic analysis of cable-related costs is given in the following.

Cable Size (AWG)	Cable Cost (\$)	Monthly PayBack (\$/month)	Cable Resistance (ohms)	Power Loss (kW)	Cost of Power Loss (\$/month)	Total Cost (\$/month)
1	59,600	1,326	0.88	0.77	28	1,353
2	52,300	1,163	1.07	0.94	34	1,197
4	43,550	969	1.71	1.49	54	1,022
6	36,050	802	2.72	2.37	85	887

Based on the total monthly costs, the AWG Size 6 is selected as the one giving the most economical solution with the lowest total cost.

Switchboard and transformer selection

The required surface voltage of the installation is the sum of the motor nameplate voltage and the cable voltage drop, as calculated from Eq. (5.28):

$$U_{\text{surf}} = 1,973 + 1.732 \cdot 2.72 \cdot 17 = 2,053 \text{ V.}$$

Selection of the switchboard is based on the kVA requirement of the installation, found from Eq. (5.29):

$$P_{\text{surf}} = 0.001732 \, 2,053 \, 17 = 61 \text{ kVA}.$$

5.8 COMPARISON OF INDUCTION MOTOR AND PERMANENT MAGNET MOTOR INSTALLATIONS

Because Examples 5.3 and 5.7 are designs for the same well conditions, comparison of their results allows establishment of several interesting conclusions on the application of induction versus PMMs. Table 5.1 presents main design parameters of the two cases; key conclusions are as follows:

- The installation with PMM requires considerably lower number of pump stages than the one using an induction-type motor. This is thanks to the much higher operating speed that increases the head developed per stage of the pump.

TABLE 5.1 Comparison of Electrical Submersible Pump Installations With Induction and Permanent Magnet Motors		
Parameter	Induction Motor	Permanent Magnet Motor
Operational speed	3,500 RPM	5,859 RPM
Required pump power	72 HP	61 HP
No. of pump stages	221	84
Pump efficiency	60%	69%
Pump length	17.6 ft	12.7 ft
Pump weight	378 lbs	344 lbs
Selected motor power	84 HP	80 HP
Motor current	48 A	17 A
Motor efficiency	84%	93%
Motor length	13.9 ft	9.9 ft
Motor weight	637 lbs	417 lbs
Required fluid velocity	1 ft/s	0.49 ft/s
Required cable size	AWG 4	AWG 6
Surface voltage	1,110 V	2,053 V
Required kVA	92 kVA	61 kVA

- A smaller capacity motor is needed for the PMM installation because of the much higher motor efficiency compared with the induction motor.
- The PMM's cooling requires about half the liquid velocity than the induction motor because of the lower developed heat in the PMM.
- A smaller and less expensive cable size is required when a PMM is used because of the lower motor current.
- The total length and weight of the pump and motor assembly for the PMM installation is about two-thirds of that of the case using an induction motor.
- The system with a PMM uses less electrical energy than the one employing a conventional induction motor.

A more detailed comparison of the two installations' energy utilization will be presented in Section 6.4.

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Chapter 6

Analysis and Optimization

6.1 INTRODUCTION

This chapter covers the most important topics of analyzing and optimizing electrical submersible pump (ESP) installations. **Systems (Nodal) analysis** is the latest tool available for the production engineer and can be used to investigate the effects of changes in operating parameters on the well's performance. The different ways of performing systems analysis calculations on ESP installations are detailed along with sample problems. An ESP unit, in addition to fluid lifting applications, can be used to determine the inflow performance of the well it is run in. Production well testing is the subject of the next section. The power conditions of ESP installations are examined and the possible sources of power losses are detailed. Power efficiencies of system components are derived, enabling the detection of excessive power wastes in the system. Finally, the basics of optimum selection of ESP equipment is covered and ways to find the most economic installation while considering both capital and operational costs are described.

6.2 NODAL ANALYSIS

The systems analysis principles outlined in Chapter 2 can readily be applied to the analysis of wells produced by ESPs [1–4]. The production system of an ESP installation is shown with the appropriate **node points** in Fig. 6.1. Although ideal operational conditions for ESPs require negligible or no gas production, in the following the specific features of the system are discussed for a generalized case when a considerable amount of solution gas enters the wellbore.

Well fluids entering through the sandface rise in the casing string up to the intake of the ESP (**Node 3**), where the flowing pressure equals the pump intake pressure (PIP). In this section of the well (in the casing string), **two-phase** vertical or inclined flow occurs whose free-gas content is determined by the pressure conditions and the composition of the incoming wellstream. Before entering the pump, because of natural and/or forced (performed by a downhole separator) **separation**, the free-gas stream, $q_{g \text{ form}}$, present at **Node 3** splits and part of it, $q_{g \text{ ann}}$, rises in the annulus to the casinghead, the rest being **ingested** by the ESP.

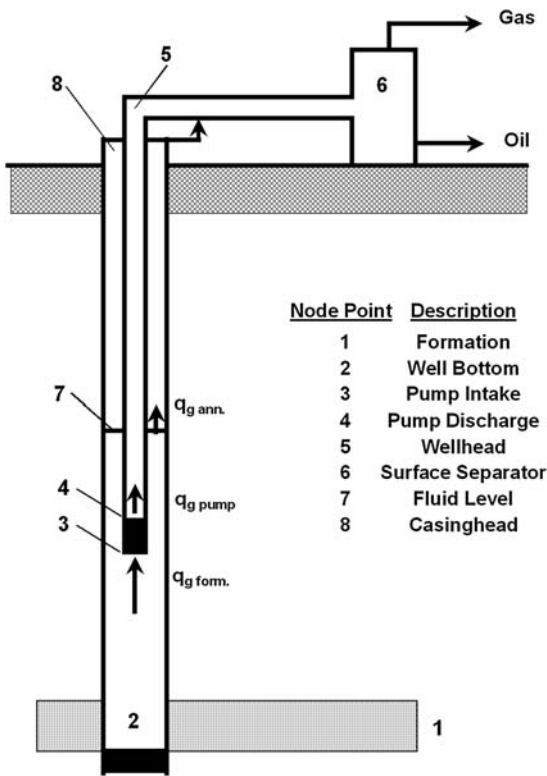


FIGURE 6.1 The production system of an oil well produced by an electrical submersible pump.

The operation of the ESP, compared to the ideal case when handling an incompressible liquid, significantly **deteriorates** because of the presence of free gas, as discussed in Chapter 4. Therefore its discharge pressure at **Node 4** varies with the gas content of the wellstream at suction conditions. The proper description of the pump's operation as well as the calculation of the pump discharge pressure require the exact knowledge of the amount of free gas **entering** the pump, $q_{g \text{ pump}}$. This gas rate, in turn, is a function of the *PIP* because the value of *PIP* determines the amount of gas remaining in solution with the oil. Consequently, the determination of the gas rate entering the pump, $q_{g \text{ pump}}$, is an iterative process. Because of this condition and the complexity of calculating the head developed by the stages of the ESP when pumping a multiphase mixture, hand calculations are cumbersome and **computer** programs are normally used.

Above **Node 4**, vertical or inclined multiphase flow occurs in the tubing string up to the wellhead where the produced fluids enter the flowline at **Node 5**, finally reaching the surface separator at **Node 6**. In the majority of cases, wells produced by ESP systems do not have a packer and continuously **vent**

produced gas to the surface. The annulus above the pump contains a static liquid column reaching up to the dynamic liquid level (**Node 7**), with a steady rate of gas bubbling through the stagnant liquid. This liquid column contains **oil** only; a condition caused by the natural separation of the oil and water phases. Above the dynamic liquid level, a static gas column occupies the casing-tubing annulus, with gas being vented to the flowline at the casinghead.

The node points of the ESP installation being defined, application of systems analysis principles permits the calculation of the **liquid rate** occurring in the production system. In general, the **NODAL analysis** procedure consists of the following basic steps:

1. A **solution node** is selected first. This can be any node point in the system, the proper selection of which facilitates the evaluation of different assumed conditions.
2. A range of liquid flow rates is selected for subsequent calculations.
3. Starting from the two endpoints of the system (**nodes 1 and 6** in Fig. 6.1) and working toward the **solution node**, the flowing pressures at every node point are calculated for the first assumed liquid rate.
4. After repeating the calculations in **step 3** for each and every assumed liquid flow rate, two sets of pressure—rate series will be available at the **solution node**. These values represent the performance curves for the two **sub-systems** created by the solution node.
5. According to a basic rule of systems analysis (see Chapter 2), inflowing and outflowing pressures at the solution node must be equal. Thus the intersection of the two performance curves defines the well's liquid flow rate under the given conditions.

The application of **NODAL analysis** principles and the solution described previously provide the basis for concrete analyses of existing or intended ESP installations and the next sections will detail several uses of this technique.

The generally accepted coordinate system used for systems analysis is the pressure versus rate coordinates, see Chapter 2. When treating ESP systems, however, the coordinate system head versus rate is also sometimes used; both treatments are described in the following.

6.2.1 Using the Head—Rate Coordinate System

If during NODAL analysis the coordinate system of the ESP's head performance curve is used, then the optimum placement of the **solution node** is at the **discharge** of the pump. The two subsystems thus created are (1) the formation plus the ESP and (2) the tubing string above the pump plus the flowline and the separator. The determination of the cooperation of these subsystems is done differently for cases with negligible or considerable free-gas production, as will be shown in the following sections.

6.2.1.1 Single-Phase Cases

In wells where there is a negligible amount of free gas present at the pump setting depth, **NODAL analysis** can be performed also by hand calculations. The solutions to be discussed in the following illustrate the basic principles and capabilities of applying **systems analysis** to the description of the operation of ESP installations. This approach is justified also by the fact that today there are very efficient downhole separators available to remove free gas from the wellstream before it enters the ESP.

To apply **NODAL analysis** to the ESP installation, the variation of flowing pressures in the well should be analyzed first. Fig. 6.2 depicts the pressures along the well depth (1) in the tubing string and (2) in the casing-tubing annulus. The well is assumed to produce a stabilized **incompressible** liquid flow rate at a flowing bottomhole pressure (FBHP) found from the well's **inflow performance relationship** (IPR) curve. In the following discussion, the well is assumed to produce against a **constant** wellhead pressure.

From the depth of the perforations up to the setting depth of the ESP, pressure in the casing changes according to the flowing pressure gradient, which, for simplicity, is approximated by the **static** gradient of the well fluid. This assumption is acceptable when low to medium flow rates are produced through big casing sizes; otherwise a pressure traverse including friction losses should be calculated. The casing pressure thus calculated at the pump setting depth is the **PIP**.

The pressure distribution in the tubing string starts at the wellhead pressure valid at the given rate and changes linearly with tubing length. Flowing tubing

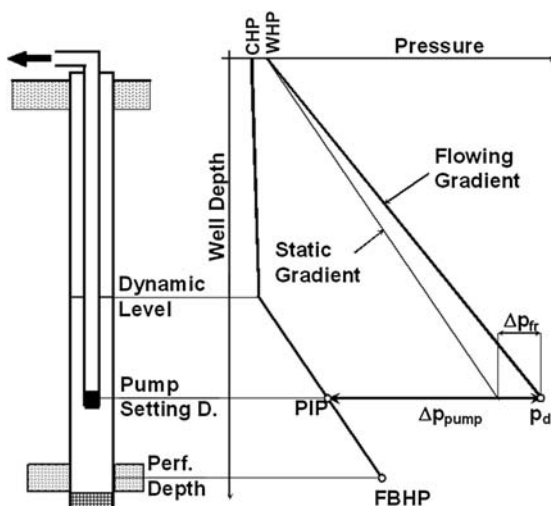


FIGURE 6.2 Pressure distributions in an electrical submersible pump installation producing incompressible liquid. CHP, casinghead pressure; FBHP, flowing bottomhole pressure; PIP, pump intake pressure; WHP, wellhead pressure.

pressure has two components: (1) one comes from the **hydrostatic** liquid gradient and (2) the other is caused by frictional pressure **losses**. At the depth of the pump discharge (which is practically at the same depth as the intake) the difference between the flowing tubing pressure and the PIP must be covered by the pressure increase developed by the ESP, denoted by Δp_{pump} .

Since the **solution node** is selected at the ESP's discharge, we have two subsystems to deal with:

- the formation plus the ESP
- the tubing string.

To find the performance curve of the first subsystem, we start from the *FBHP*, deduct the pressure losses in the annulus, and add the pressure increase developed by the pump to find the **available** pressure at the pump discharge:

$$p_d = FBHP - (L_{\text{perf}} - L_{\text{set}}) grad_l + \Delta p_{\text{pump}} \quad (6.1)$$

where

p_d = available pump discharge pressure, psi

FBHP = flowing bottomhole pressure, psi

$grad_l$ = liquid gradient, psi/ft

Δp_{pump} = pressure increase developed by the pump, psi

L_{set} = pump setting depth, ft

L_{perf} = depth of perforations, ft.

The performance curve of the tubing string [also called **vertical lift performance** (VLP)] represents the **required** discharge pressure of the ESP and is calculated from the wellhead pressure and the sum of the pressure losses in the tubing:

$$p_d^* = WHP + L_{\text{set}} grad_l + \Delta p_{\text{fr}} \quad (6.2)$$

where

p_d^* = required pump discharge pressure, psi

WHP = tubinghead pressure, psi, and

Δp_{fr} = frictional pressure drop in the tubing string, psi.

As the available and the required pressures must be equal at the **solution node**, the simultaneous solution of the two formulas for the pressure increase to be developed by the ESP results in the following expression:

$$\Delta p_{\text{pump}} = WHP + L_{\text{perf}} grad_l + \Delta p_{\text{fr}} - FBHP \quad (6.3)$$

where

Δp_{pump} = pressure increase developed by the pump, psi

WHP = tubinghead pressure, psi

L_{perf} = depth of perforations, ft

$grad_l$ = liquid gradient, psi/ft

$FBHP$ = flowing bottomhole pressure, psi

Δp_{fr} = frictional pressure drop in the tubing string, psi.

Because the ESP industry uses **head** instead of pressure, the previous equation is divided by the liquid gradient valid at average pump conditions to arrive at the necessary head of the pump:

$$\Delta H_{\text{pump}} = L_{\text{perf}} + \Delta H_{fr} - \frac{2.31}{\gamma_l} (FBHP - WHP) \quad (6.4)$$

where

ΔH_{pump} = required pump head, ft

L_{perf} = depth of perforations, ft

$FBHP$ = flowing bottomhole pressure, psi

WHP = tubinghead pressure, psi

ΔH_{fr} = frictional head drop in the tubing string, ft

γ_l = specific gravity of the produced liquid, —.

6.2.1.1.1 Constant Pumping Speed

Eq. (6.4) allows one to calculate the head **required** to produce a given amount of liquid by the ESP system against a **constant** wellhead pressure. If plotted for several rates, the calculated values of pump head represent the **performance curve** of the formation plus ESP subsystem. If the pump is driven at a constant electrical frequency the head **available** from the ESP is, of course, described by its **H—Q Curve** valid at a given frequency. The liquid rate developing in the production system of the ESP installation is found at the intersection of the two curves, where the **required** and the **available** heads are equal.

The following worked example illustrates the use of **NODAL analysis** for the given case, i.e., for a constant electrical frequency and constant wellhead pressure.

Example 6.1

Determine the ESP system's liquid production rate if the pump is represented by the performance curve shown in Fig. 6.3 (100 stages of an **SN3600** pump running at 60 Hz). Use wellhead pressures of 100, 200, and 300 psi, and other well data being given as follows.

Static bottomhole pressure = 2,000 psi

Tubing inside diameter (ID) = 2.441 in, new pipe

PI = 5 bpd/psi

Liquid SpGr = 1.0

Depth of perforations = 6,000 ft

Pump setting depth = 5,000 ft

Solution

To plot the VLP curve of the well the head required to produce various liquid rates has to be calculated for several possible liquid flow rates and wellhead pressures. In the following, the calculation for a rate of $q = 3,000$ bpd and a wellhead pressure of 100 psi is detailed only.

First calculate the well's *FBHP* valid at the given rate from the PI equation (Eq. 2.2):

$$FBHP = 2,000 - 3,000/5 = 1,400 \text{ psi.}$$

The frictional head loss in the tubing string is found from Fig. A.1 in the **Appendix**. For new $2\frac{7}{8}$ " tubing at the given rate, a value of 76 ft/1,000 ft is read, which gives a total head loss of

$$\Delta H_{fr} = 76 \cdot 5,000/1,000 = 380 \text{ ft.}$$

The required head can now be calculated from Eq. (6.4):

$$\Delta H_{pump} = 6,000 + 380 - 2.31/1 (1,400 - 100) = 3,377 \text{ ft.}$$

This gives one point on the well performance curve belonging to the wellhead pressure of 100 psi; results for other cases are given in Table 6.1.

A plot of the two subsystem's performance curves is given in Fig. 6.3. As seen, the common operating points of the well and the given ESP occur at liquid rates that fall in the recommended operating range of the pump.

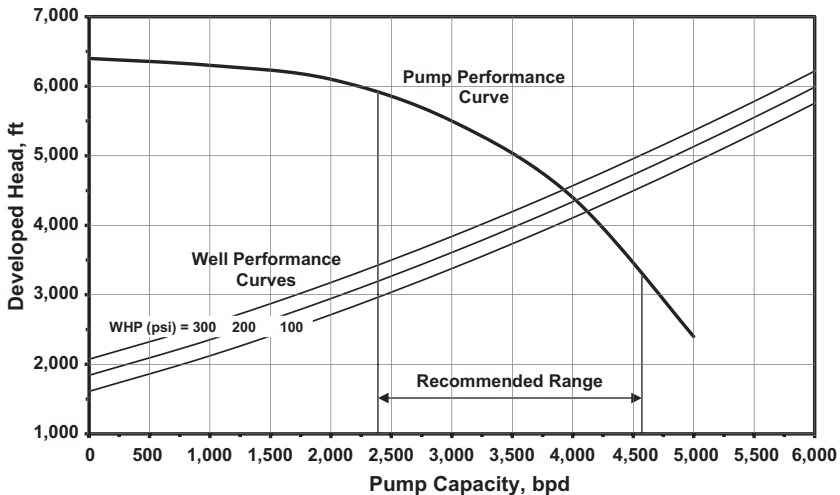


FIGURE 6.3 Results of systems analysis for Example 6.1.

TABLE 6.1 Calculation Results for **Example 6.1**

Liquid Rate (bpd)	FBHP (psi)	Friction Gradient (ft/100 ft)	Friction Drop (ft)	ΔH for Wellhead Pressures of		
				100 psi ft	200 psi ft	300 psi ft
0	2,000	0.00	0	1,611	1,842	2,073
1,000	1,800	1.00	50	2,123	2,354	2,585
2,000	1,600	3.59	180	2,715	2,946	3,177
3,000	1,400	7.61	380	3,377	3,608	3,839
4,000	1,200	12.96	648	4,107	4,338	4,569
5,000	1,000	19.58	979	4,900	5,131	5,362
6,000	800	27.43	1,372	5,755	5,986	6,217
7,000	600	36.48	1,824	6,669	6,900	7,131

6.2.1.1.2 Variable Pumping Speeds

If a variable-speed drive (VSD) unit is used and the ESP can be driven at different speeds, **NODAL analysis** calculations follow the procedure detailed previously, with the only difference that several **pump performance curves** belonging to the different possible pumping speeds are used. This kind of solution enables one to study the effect of changing the driving frequency on the liquid production rate of the ESP installation.

A sample comparison of an ESP's operation under a constant speed and at variable frequencies is given in Fig. 6.4. The installation was designed for a liquid rate of 3,000 bpd but well inflow information was limited and/or unreliable. Based on the data available, an **SN3600** pump with 100 stages was chosen, and the unit was run in the well. Pump performance at 60 Hz is indicated by the solid line representing the head **available** to lift well fluids to the surface. The head **required** is represented by the well performance curve determined in Example 6.1 for a wellhead pressure of 100 psi.

The intersection of the two previously mentioned curves at **point 1** shows the production rate that occurs in the given case. Because this rate (4,200 bpd) is much greater than the one desired, the operator had to use a wellhead **choke** to limit the well's liquid production. By using a choke of the right size, production is decreased to the required level (3,000 bpd) but this involves a head (pressure) drop proportional to the vertical distance between **points 2 and 3**. This happens because the pump develops a head corresponding to **point 2** but the well requires a much lower head as shown at **point 3**; the difference being consumed by the head (pressure) drop across the choke. The energy corresponding to the product of the rate and the pressure drop is completely **wasted**, very substantially reducing the ESP system's efficiency. It is interesting to note

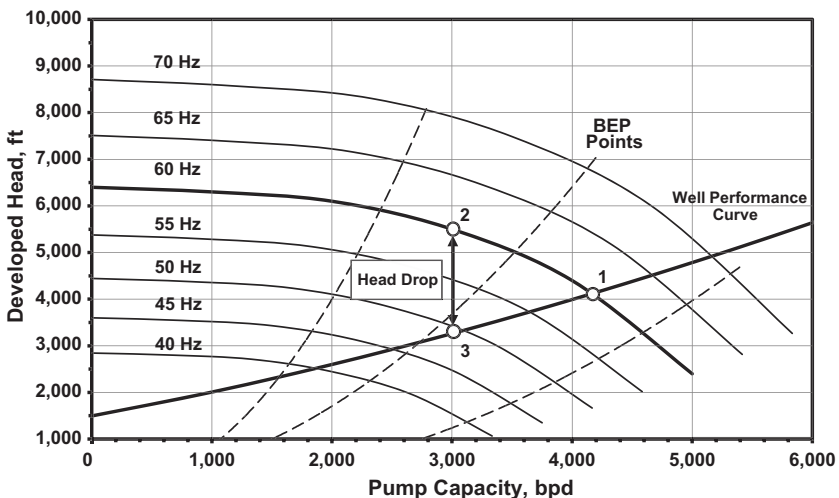


FIGURE 6.4 Comparison of the electrical submersible pump system's operation at 60 Hz and at variable frequencies. *BEP*, best efficiency point.

that the situation could be even worse if the desired rate fell outside the recommended operating range of the pump when mechanical deterioration of the ESP equipment would be inevitable.

By comparison, if a VSD unit is used in the same well, the desired rate of 3,000 bpd is easily attained by **reducing** the electrical frequency to about 49 Hz. The cooperation between the well and the ESP unit now occurs at **point 3**, where the well performance curve intersects the pump's head capacity curve valid for 49 Hz. As seen, the actual operating point falls very close to the pump's best efficiency point (BEP), indicating that a pump efficiency near the **optimum** value can be realized.

The detrimental effect of **choking** an ESP unit is clearly indicated by the amount of power **wasted** through the surface choke. This can be calculated from Eq. (4.12), expressing the brake horsepower needed to drive the pump at any rate and head. If we disregard the change in pump efficiency at the two electrical frequencies, the following formula can be derived:

$$BHP_{wasted} = 7.368 \cdot 10^{-6} \cdot q_l \Delta H_{choke} \gamma_l \quad (6.5)$$

where

BHP_{wasted} = power wasted through choke, HP

q_l = pumping rate, bpd

ΔH_{choke} = head loss across surface choke, ft

γ_l = specific gravity of the produced liquid, —.

Example 6.2

Determine the amount of the power wasted because of the surface choke in the ESP installation shown in Fig. 6.4.

Solution

After reading the head loss across the surface choke installed at the wellhead as

$$\Delta H_{choke} = 2,200 \text{ ft}$$

the wasted power is found from Eq. (6.5):

$$BHP_{wasted} = 7.368 \cdot 10^{-6} \cdot 3,000 \cdot 2,200 \cdot 1 = 48.6 \text{ HP.}$$

6.2.1.1.3 Variable Wellhead Pressures

In the previous sections the ESP system was assumed to operate against a **constant** wellhead pressure, i.e., the effect of flowline length and diameter on the operating conditions was not included. This is seldom acceptable because the wellhead pressure changes **significantly** in the broad liquid rate ranges

normally encountered in wells produced by ESPs. Therefore a **full** NODAL analysis is required that includes all components of the production system, and the calculation models described so far must be modified accordingly.

Still using the pump's discharge as the **solution** node, the two **subsystems** of the complete NODAL analysis become

- the formation plus the ESP
- the tubing string plus the flowline and the separator.

The necessary calculations mostly follow those described in the previous sections, with the only difference that now the wellhead pressure is **calculated** from the separator pressure and the pressure drop occurring in the flowline. The latter is usually found from the **Hazen–Williams** formula (Eq. 2.15) or from an appropriate chart, see **Appendix A**.

The wellhead pressure is the sum of the separator pressure and the pressure drop in the flowline:

$$WHP = p_{\text{sep}} + 0.433 \gamma_l \Delta H_{\text{fl}} \quad (6.6)$$

where

WHP = tubinghead pressure, psi

p_{sep} = surface separator pressure, psi

γ_l = specific gravity of the produced liquid, —

ΔH_{fl} = frictional head drop in the flowline, ft.

When calculating the **required** head capacity of the ESP the previous formula must be introduced into Eq. (6.4), instead of a constant wellhead pressure. The rest of the calculations are identical to those described earlier.

Example 6.3

Using the data in Example 6.1 find the production system's liquid rate for a separator pressure of 100 psi and a flowline length of 3,000 ft. Assume two values for the flowline ID (2" and 3") and the use of a VSD unit.

Solution

To superimpose the well performance curve on the ESP's performance curve, the head required to produce various liquid rates has to be calculated for the two flowline sizes and the given separator pressure. In the following the calculation for a rate of $q = 3,000$ bpd and a flowline ID of 2" is detailed only.

In the previous example, the following parameters were found at the given liquid rate: $FBHP = 1,400$ psi and the total head loss in the tubing string $\Delta H_{\text{tr}} = 380$ ft.

To find the wellhead pressure, the frictional pressure drop in the flowline has to be calculated using Eq. (2.15), assuming a new pipe and the first flowline ID of 2 in.

Converting the flow rate to gpm units

$$q = 3,000\ 42/1,440 = 87.5\ \text{gpm.}$$

$$\Delta h_{\text{fl}} = 0.2083(100/120)^{1.85} (87.5^{1.85} / 2^{4.86}) = 20\ \text{ft}/100\ \text{ft.}$$

The total frictional head equals

$$\Delta H_{\text{fl}} = 0.2\ 3,000 = 600\ \text{ft.}$$

The wellhead pressure is found from Eq. (6.6) as

$$WHP = 100 + 0.433\ 1\ 600 = 360\ \text{psi.}$$

The required pump head is calculated from Eq. (6.4):

$$\Delta H_{\text{pump}} = 6,000 + 380 - 2.31(1,400 - 360) = 3,978\ \text{ft.}$$

Table 6.2 contains results of the wellhead pressure and the required pump head calculations for the two flowline sizes investigated.

Complete calculation results in graphical form are given in Fig. 6.5. Intersections of the well and pump performance curves indicate the possible liquid rates at any electrical frequency and flowline size. This kind of **NODAL Analysis** allows one to investigate the effects of flowline size on the ESP installation's liquid rate or to find the proper electrical frequency to produce a desired liquid rate.

6.2.1.2 Multiphase Cases

As seen in the previous sections, production of a single-phase liquid by an ESP allowed the derivation of simple formulas facilitating the **NODAL analysis** of the ESP system's operation. This was made possible by the fact that all pressure traverses were linear functions of well depth because an **incompressible** fluid was pumped. As soon as a considerable amount of **free** gas enters the system, a compressible **multiphase** medium is transferred by the pump and pressure traverses in the casing annulus and the tubing string will have different gradients at different well depths because of the different flowing pressures. In conclusion, the simple calculation model developed in Section 6.2.1.1 cannot be used for cases when free gas is also pumped.

It must be noted that calculation of multiphase pressure drops in the tubing string is a tedious and relatively **inaccurate** process because of the complex nature of multiphase flow. Although there are many calculation models available, they all have their limitations and can produce very dissimilar results for the same case. As there is no "overall best" pressure drop calculation method, the accuracy and reliability of NODAL analysis can be increased only by using the model giving the least calculation error for the given conditions; this model can be and must be found beforehand. Basic considerations on selecting the proper multiphase flow calculation model are discussed by Takacs [5].

TABLE 6.2 Calculation Results for [Example 6.3](#)

Liquid Rate (bpd)	FL Frict. Grad.		FL Frict. Drop		WHP		ΔH for Flowline	
	2" ft/100 ft	3" ft/100 ft	2" ft	3" ft	2" psi	3" psi	2" ft	3" ft
0	0.00	0.00	0	0	100	100	1,611	1,611
1,000	2.63	0.37	79	11	134	105	2,202	2,134
2,000	9.47	1.32	284	40	223	117	2,999	2,754
3,000	20.04	2.79	600	84	360	136	3,978	3,461
4,000	34.12	4.76	1,024	143	543	162	5,131	4,250
5,000	51.56	7.19	1,547	216	770	193	6,447	5,116
6,000	72.25	10.07	2,167	302	1,038	231	7,922	6,057
7,000	96.09	13.39	2,883	402	1,348	274	9,552	7,071

FL, flowline; *WHP*, wallhead pressure.

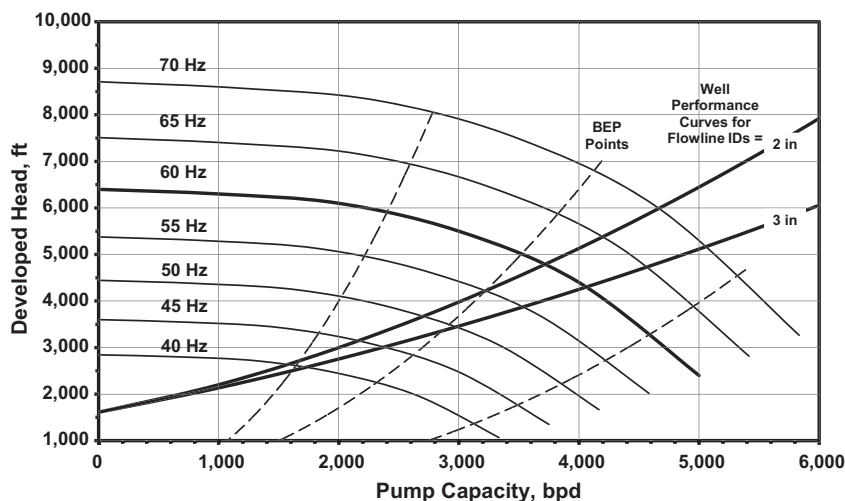


FIGURE 6.5 Electrical submersible pump system performance for various flowline sizes, Example 6.3. ID, inside diameter.

Although **NODAL analysis** calculations for ESP installations producing considerable amounts of formation gas necessitate the use of sophisticated [6–9] computer programs, the present section introduces readers to the basic principles used and illustrates some practical solutions.

6.2.1.2.1 Calculation Model

The pressure distributions in an ESP installation producing multiphase fluids are illustrated in Fig. 6.6. If a **NODAL analysis** is performed on this system with the **solution node** selected at the ESP's discharge, the performance relationships of the two subsystems thus created will change from those given in Eqs. (6.1) and (6.2).

The performance of the formation plus the ESP is described starting from the *FBHP* and includes the pressure drop in the casing annulus as well as the pressure increase (Δp_{pump}) developed by the pump. The pump discharge pressure thus calculated represents the pressure **available** to transfer well fluids to the surface and is described by the following formula:

$$p_d = FBHP - \Delta p_c + \Delta p_{\text{pump}} \quad (6.7)$$

where

p_d = available pump discharge pressure, psi

FBHP = flowing bottomhole pressure, psi

Δp_c = multiphase pressure drop in the annulus, psi

Δp_{pump} = pressure increase developed by the pump, psi.

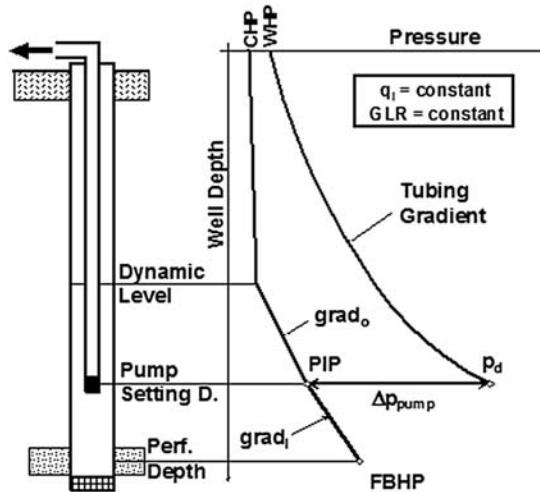


FIGURE 6.6 Pressure distributions in an electrical submersible pump installation producing compressible fluids. *CHP*, casinghead pressure; *FBHP*, flowing bottomhole pressure; *GLR*, gas–liquid ratio; *PIP*, pump intake pressure; *WHP*, wellhead pressure.

For simplicity, let us assume a **constant** wellhead pressure when the **required** discharge pressure of the ESP is composed of the wellhead pressure and the multiphase pressure drop in the tubing string. This defines the performance relationship of the subsystem consisting of the tubing string and the wellhead and can be written as

$$p_d^* = WHP + \Delta p_t \quad (6.8)$$

where

p_d^* = required pump discharge pressure, psi

WHP = tubinghead pressure, psi

Δp_t = multiphase pressure drop in the tubing string, psi.

According to **NODAL analysis** principles, these pressures should be identical; after equating the two terms and expressing the **necessary** head to be developed by the pump, we get

$$\Delta H_{\text{pump}} = \frac{WHP + \Delta p_t + \Delta p_c - FBHP}{0.433 \gamma_l} \quad (6.9)$$

where

ΔH_{pump} = required pump head, ft

WHP = tubinghead pressure, psi

$FBHP$ = flowing bottomhole pressure, psi

Δp_t = multiphase pressure drop in the tubing string, psi

Δp_c = multiphase pressure drop in the annulus, psi

γ_l = average specific gravity of fluid in pump, –.

As before, this formula must be modified if a **full** NODAL analysis is performed when the wellhead pressure is found from the separator pressure plus the pressure losses in the flowline.

Calculation of the **average** fluid gravity requires the determination of the mixture's flowing density through each stage of the centrifugal pump. This time-consuming task is generally solved on the computer; for hand calculations the specific gravity of the fluid is usually determined at pump **suction** conditions.

As seen from the formula, the analysis necessitates the calculation of **multiphase** pressure drops in both the tubing and the casing; this is usually solved by using a computer program. To find these pressure losses, it is necessary to know the multiphase mixture's actual production **gas—liquid ratio** (*GLR*). Depending on the part of the well being studied, the following considerations apply.

Pressure drop in the annulus between the perforations and the pump intake.

Flow in this section of the well may be single- or multiphase depending on the bubblepoint pressure. The *GLR* to be used for multiphase pressure drop calculations equals the well's **production GLR**.

Pressure drop in the tubing from the pump discharge to the wellhead.

When calculating the pressure losses in the tubing string, the actual gas content of the wellstream above the pump should reflect the amount of gas (1) separated by **natural** separation in the well's annulus and (2) separated by the ESP rotary **separator** (if any). The resulting *GLR* is found from Eq. (5.39), reproduced here:

$$GLR = \frac{\frac{5.61 q'_{ing}}{B_g} + R_s q_o}{q_l} \quad (6.10)$$

where

GLR = tubing gas—liquid ratio, scf/STB

q'_{ing} = free-gas rate ingested by pump, bpd

B_g = gas volume factor at suction conditions, ft³/scf

R_s = solution gas—oil ratio at suction conditions, scf/STB

q_o , q_l = oil and liquid production rates, STB/day.

The variables figuring in the formula are found as given in the following. The free gas **ingested** by the pump (in bpd units) is determined by considering the efficiencies of the natural and the forced gas separation processes:

$$q'_{ing} = \frac{q'_g}{5.61} \left(1 - \frac{\eta_n}{100}\right) \left(1 - \frac{\eta_{sep}}{100}\right) \quad (6.11)$$

where

q'_{ing} = gas volume ingested by pump, bpd

q'_g = free-gas rate at pump suction, scf/d

η_n = natural gas separation efficiency, %

η_{sep} = rotary gas separator efficiency, %.

Finally, the volumetric rate of free gas (in ft³/day units) entering the well is

$$q'_g = q_o(GOR - R_s)B_g \quad (6.12)$$

where

q'_g = free-gas volumetric rate, ft³/day

q_o = oil volumetric rate, STB/day

GOR = production gas–oil ratio, scf/STB

R_s = solution GOR at PIP, scf/STB

B_g = gas volume factor at PIP, ft³/scf.

6.2.1.2.2 Applications

The previous section described how the **required** pump head for any liquid rate can be calculated when the ESP system handles a **multiphase** mixture. If the heads calculated from Eq. (6.9) are plotted for several liquid production rates and the resulting curve is superimposed on the head performance curve of the given ESP, the production rate(s) occurring in the system are represented at the intersection(s) of those curves. The number of possible liquid rates is a function of the available electrical frequencies to drive the ESP system.

Using this approach, **NODAL analysis** of all the previously described cases (constant or variable pumping speeds, variable wellhead pressures) can be performed. In the following the main steps of the solution are listed.

1. Based on the well's IPR data, a set of liquid flow rates (in STB/day units) are assumed.
2. The first liquid rate is taken and the necessary *FBHP* is calculated.
3. Using a computer program or pressure traverse curves, the pressure drop, Δp_c , occurring in the casing string between the depth of the perforations and the pump setting depth is determined. The *GLR* to be used is the well's production *GLR*.
4. The *PIP* at the given rate is found by deducting the annular pressure drop, Δp_c , from the *FBHP* just calculated.
5. The "in situ" volumetric liquid rate, q'_l , to be handled by the pump is found from Eq. (5.4). This is the rate to be later plotted on the pump's performance curve.
6. The "in situ" free-gas rate at pump suction conditions, q'_g , is calculated from Eq. (6.12).
7. The efficiency of the natural gas separation in the well's annulus, η_n , is found by applying the procedure detailed in Section 4.4.2.

- 8. The volume of gas ingested by the pump, q'_{ing} , can be determined next by considering the efficiency of the rotary gas separator; Eq. (6.11) must be used.
- 9. The multiphase pressure drop occurring in the tubing string between the pump setting depth and the wellhead, Δp_t , is calculated by a computer program or found from pressure traverse curves. The *GLR* of the multiphase flow is given by Eq. (6.10).
- 10. The head, ΔH_{pump} , to be developed by the ESP to produce the given liquid rate is determined from Eq. (6.9).
- 11. Steps 2–10 are repeated for the rest of the assumed liquid flow rates.
- 12. The corresponding $q'_l - \Delta H_{\text{pump}}$ points plotted in an $H - Q$ coordinate system define the performance curve of the formation plus ESP subsystem. The intersection(s) of this curve with the ESP's rate–head performance curve(s) determine the liquid rate(s) possible from the given installation.

The procedure detailed previously allows one to analyze the operation of the ESP system for several possible cases: with an ESP running at a constant or variable speed, with constant or variable wellhead pressures, etc. The next example problem illustrates the case in which the wellhead pressure and the pump's operational speed are constant.

Example 6.4

Find the liquid production rate occurring in the production system characterized by the following data.

Well Physical Data

Casing ID	6.276 in
Tubing ID	3.476 in
Tubing OD	4 in
Perforation Depth	7,500 ft
Pump Depth	6,000 ft

ID, inside diameter; *OD*, outside diameter.

Well Performance Data

WHP	200 psi
Water Cut	50%
SBHP	2,500 Psi
PI	3 bpd/psi
GOR produced	300 scf/STB
BHT	150 F
WHT	80 F

BHT, bottomhole temperature; *GOR*, gas–oil ratio; *PI*, productivity index; *SBHP*, static bottomhole pressure; *WHP*, wallhead pressure; *WHT*, wellhead temperature.

Fluid Properties

SpGr of Oil	0.85
SpGr of Water	1.1
SpGr of Gas	0.65

ESP Equipment Data

ESP Type	S5000 N
No. of Stages	86
Rotary Gas Separator	60%
Efficiency	

Solution

Detailed calculations for a liquid rate of $q_l = 3,000$ STB/day are given in the following.

The well's *FBHP* is found from Eq. (5.1) as

$$FBHP = 2,500 - 3,000/3 = 1,500 \text{ psi.}$$

From multiphase flow calculations and using the $GLR = 150$ scf/STB value, the pressure drop in the annulus between the perforations and the pump setting depth was found as

$$\Delta p_c = 540 \text{ psi.}$$

The *PIP* is easily found:

$$PIP = FBHP - \Delta p_c = 1,500 - 540 = 960 \text{ psi.}$$

To find the “in situ” volumetric liquid rate entering the pump, the solution gas—oil ratio and the oil volume factor at suction conditions must be determined first. For these the temperature at the pump suction is easily found from

$$\begin{aligned} T_{\text{pump}} &= \text{WHT} + L_{\text{pump}}(\text{BHT} - \text{WHT})/L_{\text{perf}} \\ &= 80 + 6,000(150 - 80)/7,500 = 136^\circ\text{F.} \end{aligned}$$

R_s at suction conditions is calculated from Eq. (5.3), where the oil's American Petroleum Institute (API) gravity and the variable y are found as

$$\begin{aligned} ^\circ\text{API} &= 141.5/0.85 - 131.5 = 35 \\ y &= 0.00091 \cdot 136 - 0.01215 \cdot 35 = -0.301 \end{aligned}$$

and

$$R_s = 0.7[960/(18 \cdot 10^{-0.301})]^{1.205} = 194 \text{ scf/bbl.}$$

B_o at suction conditions from Eq. (5.5) follows

$$F = 194(0.65/0.85)^{0.5} + 1.25 \cdot 136 = 339$$

and

$$B_o = 0.972 + 1.47 \cdot 10^{-4} \cdot 339^{1.175} = 1.1 \text{ bbl/STB.}$$

The liquid volumetric rate handled by the pump is found from Eq. (5.4), and assuming $B_w = 1$

$$q'_l = 1,500 \cdot 1.1 + 1,500 \cdot 1 = 3,150 \text{ bpd.}$$

To find the “in situ” gas volume, the gas volume factor, B_g , must be calculated, for which the knowledge of the deviation factor is necessary.

The pseudocritical parameters of the gas phase are calculated from Eqs. (4.27) and (4.28).

$$p_{pc} = 709 - 58.7 \cdot 0.65 = 671 \text{ psia}$$

and

$$T_{pc} = 170.5 + 307.3 \cdot 0.65 = 370 \text{ R.}$$

The reduced parameters and the deviation factor from Eq. (4.26):

$$p_{pr} = 960/671 = 1.45$$

and

$$T_{pr} = (136 + 460)/370 = 1.61$$

$$Z = 1 - 3.52 \cdot 1.45/10^{0.9813 \cdot 1.61}$$

$$+ 0.274 \cdot 1.45^2/10^{0.8157 \cdot 1.61} = 0.893$$

Now the gas volume factor can be determined from Eq. (4.25):

$$B_g = 0.0283 \cdot 0.893(136 + 460)/960 = 0.0156 \text{ cu ft/scf.}$$

The volumetric rate of the free gas is found from Eq. (6.12) as

$$q'_g = 1,500(300 - 187)0.0156 = 2,644 \text{ ft}^3/\text{day.}$$

The next task is the determination of the efficiency of natural separation in the annulus. First find the superficial liquid velocity in the annulus where the annular cross-sectional area is

$$A = 0.0055(6.276^2 - 4^2) = 0.129 \text{ ft}^2.$$

The superficial liquid velocity from Eq. (4.17), assuming $B_w = 1$,

$$v_{sl} = 6.5E^{-5} \cdot 3,000(1.1/2 + 1/2)/0.129 = 1.6 \text{ ft/s.}$$

Gas and liquid densities from Eqs. (4.19) and (4.20), respectively,

$$\rho_g = 0.0764 \cdot 0.65/0.0156 = 3.18 \text{ lb/ft}^3$$

$$\rho_l = 62.4[0.85/(1.1 \cdot 2) + 1.1 \cdot 1/2] = 58.4 \text{ lb/ft}^3.$$

The terminal bubble rise velocity can now be calculated from Eq. (4.18) by assuming $\sigma = 0.04 \text{ lb/s}^2$ and substituting $g = 32 \text{ ft/s}^2$:

$$v_b = 2^{1/2} \{ [0.04(58.4 - 3.18)32] / 58.4^2 \}^{1/4} = 0.537 \text{ ft/s.}$$

The efficiency of natural gas separation in the annulus is found from Eq. (4.16):

$$\eta_n = 0.537 / (0.537 + 1.6) 100 = 25\%.$$

The free-gas volume ingested by the ESP is determined from the combined effects of the natural and forced gas separation; Eq. (6.11) gives

$$q'_{ing} = 2,644 / 5.61(1 - 0.25)(1 - 0.60) = 141 \text{ bpd.}$$

This gas rate corresponds to a *GLR* in the tubing string as found from Eq. (6.10):

$$GLR = [(5.61 \cdot 141) / 0.0156 + 187 \cdot 1,500] / 3,000 = 110 \text{ scf/STB.}$$

Now multiphase pressure drop calculations are required to find the pressure drop in the tubing string; these resulted in a pressure drop of

$$\Delta p_t = 2,280 \text{ psi.}$$

The head required from the ESP is found from Eq. (6.9), where the liquid specific gravity is determined from the liquid density at suction conditions as follows:

$$\gamma_l = \rho_l / 62.4 = 58.4 / 62.4 = 0.936$$

$$\Delta H_{\text{pump}} = (200 + 2,280 + 540 - 1,500) / 0.433 / 0.936 = 3,750 \text{ ft.}$$

Now one point on the performance curve of the formation plus ESP subsystem is defined by the two coordinates: $q'_l = 3,150 \text{ bpd}$ and $\Delta H_{\text{pump}} = 3,750 \text{ ft.}$

Calculation results for other assumed liquid rates are contained in Table 6.3; NODAL analysis of the problem is illustrated in Fig. 6.7. As seen, the actual liquid rate from this installation (about 3,700 bpd) falls outside the recommended operating range of the pump. Depending on the rate desired, corrective action is needed to ensure proper operation of the ESP–well system.

6.2.2 Using the Pressure–Rate Coordinate System

In the petroleum industry, **NODAL analysis** calculations are most commonly presented in a coordinate system of bottomhole pressure versus rate, the usual coordinate system of the **IPR curves**. In this case the ideal solution node is set at the formation; dividing the total system into the two subsystems of (1) the

TABLE 6.3 Detailed Calculation Results for [Example 6.4](#)

q_l (STB/ day)	$FBHP$ (psi)	Δp_c (psi)	PIP (psi)	q_l' (bpd)	q_g' (ft ³ / day)	v_b (ft/ s)	η_n (%)	q'_{ing} (bpd)	GLR (scf/ STB)	Δp_t (psi)	ΔH_{pump} (ft)
500	2,333	583	1,750	539	0	0.532	66	0	150	2,040	1,232
1,000	2,167	667	1,500	1,078	0	0.534	50	0	150	2,040	1,836
2,000	1,833	583	1,250	2,137	501	0.536	33	24	134	2,050	2,499
3,000	1,500	540	960	3,150	2,644	0.537	25	141	110	2,280	3,750
4,000	1,167	487	680	4,163	7,885	0.538	20	448	90	2,360	4,605
5,000	833	433	400	5,146	22,615	0.539	17	1,336	71	2,440	5,435
6,000	500	300	200	6,132	62,239	0.540	15	3,780	60	2,520	6,081

$FBHP$, flowing bottomhole pressure; GLR , gas–liquid ratio; PIP , pump intake pressure.

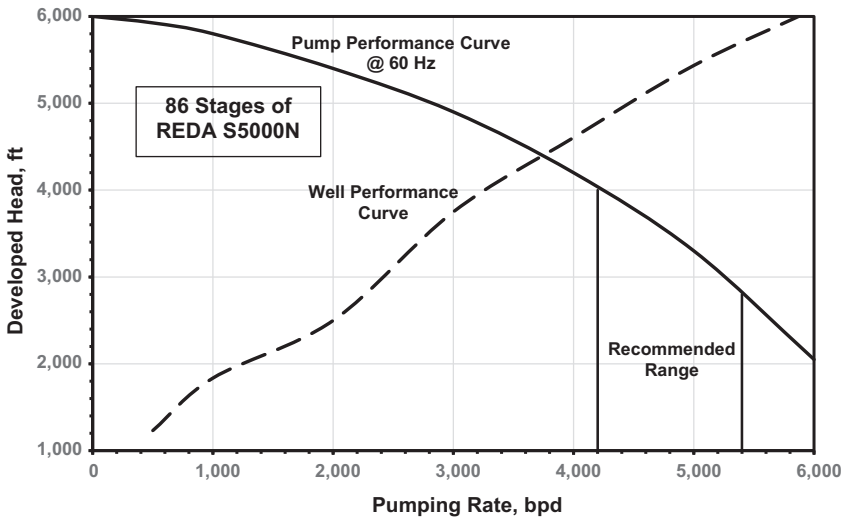


FIGURE 6.7 NODAL analysis results for Example 6.4.

formation and (2) the well plus the ESP. The performance curve of the formation is identical to the well's **IPR curve**; the performance curve (or curves if a VSD unit is used) of the other subsystem is found by calculating the *FBHP* values in the well–ESP subsystem.

6.2.2.1 Single-Phase Cases

With the **solution node** at the well bottom, the calculation of one point on the performance curve of the well–ESP subsystem is easily derived if the head ΔH_{pump} developed by the ESP at a liquid rate q_l is known from the **pump performance curve**. As shown in Fig. 6.2, the well's *FBHP* belonging to this case is found from the pump discharge pressure, p_d ; the pressure increase through the pump, Δp_{pump} ; and the hydrostatic pressure of the liquid column below the pump:

$$FBHP = p_d - \Delta H_{\text{pump}} \text{ grad}_l + (L_{\text{perf}} - L_{\text{set}}) \text{ grad}_l \quad (6.13)$$

where

- $FBHP$ = flowing bottomhole pressure, psi
- p_d = pump discharge pressure, psi
- ΔH_{pump} = head developed by the pump, ft
- grad_l = liquid gradient, psi/ft
- L_{perf} = depth of perforations, ft
- L_{set} = pump setting depth, ft.

The pump discharge pressure, however, is easily found from the pressure distribution in the tubing string; by assuming a **constant** wellhead pressure, we get

$$p_d = WHP + L_{\text{set}} \text{ grad}_l + \Delta p_{\text{fr}} \quad (6.14)$$

where

p_d = pump discharge pressure, psi

WHP = tubinghead pressure, psi

L_{set} = pump setting depth, ft

grad_l = liquid gradient, psi/ft

Δp_{fr} = frictional pressure drop in the tubing string, psi.

After substitution of the previous formula into Eq. (6.13) the following final expression is obtained:

$$FBHP = WHP + (L_{\text{perf}} - \Delta H_{\text{pump}}) \text{ grad}_l + \Delta p_{\text{fr}} \quad (6.15)$$

where

$FBHP$ = flowing bottomhole pressure, psi

WHP = tubinghead pressure, psi

L_{perf} = depth of perforation, ft

grad_l = liquid gradient, psi/ft

ΔH_{pump} = head developed by the pump, ft

Δp_{fr} = frictional pressure drop in the tubing string, psi.

During the derivation of Eq. (6.15), it was assumed that the static liquid gradients in the tubing and the casing string are equal, which is fully justified in single-phase flow.

Using this formula, the construction of the performance curve(s) for the well plus ESP subsystem is straightforward. The calculation steps given in the following assume that the ESP is driven by a VSD unit.

1. Take one possible driving **frequency** and use the appropriate performance curve of the ESP in the following.
2. Select several **liquid rates** in the recommended operating range of the pump.
3. Take the first selected rate and find the head developed by the pump at that rate.
4. Calculate the $FBHP$ achieved in the well by using Eq. (6.15).
5. Plot the bottomhole pressure just calculated in function of the liquid rate.
6. Take the next liquid rate and repeat from **step 4** until all rates are processed.
7. Select the next driving frequency and repeat the calculations from **step 2**.

After connecting the points belonging to the identical driving frequencies the performance of the subsystem well—ESP is given in the coordinate system of the **IPR curve**. The possible operating points of the ESP installation will be

found at the **intersections** of these curves with the well's inflow performance curve.

This kind of solution is useful when evaluating the effects of a VSD unit on the well's liquid production rate. Usually, only the portions of the pump performance curves falling in the recommended operating ranges are processed; therefore, it is easy to see if the pump operates in its optimum liquid rate range.

Example 6.5

Perform a **NODAL analysis** using the coordinate system of the IPR curve for the data in [Example 6.3](#). Assume a constant wellhead pressure of 100 psi.

Solution

The inflow performance of the well is characterized by a straight line starting at a static bottomhole pressure (*SBHP*) of 2,000 psi and having a slope equal to the productivity index (*PI*) of 5 bpd/psi, see the solid line in [Fig. 6.8](#). The ESP's performance curves at different driving frequencies were given in [Fig. 6.5](#).

In the following the finding of one point on the performance curve of the well plus ESP subsystem is provided. This is for 60-Hz operation and a liquid rate of $q_l = 3,000$ bpd.

The head increase developed by the pump at this rate is found from the pump performance curve valid at 60 Hz as

$$\Delta H_{\text{pump}} = 5,500 \text{ ft.}$$

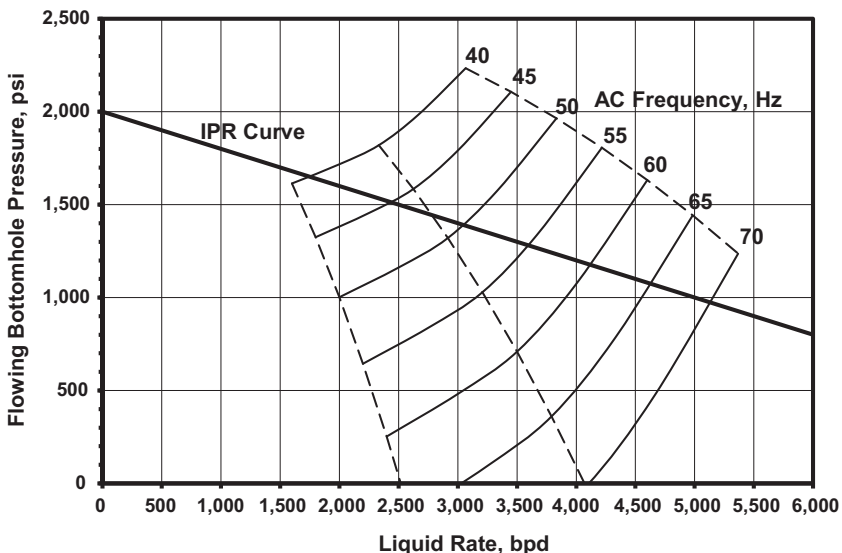


FIGURE 6.8 NODAL analysis results for [Example 6.5](#). IPR, inflow performance relationship.

TABLE 6.4 Detailed Calculation Results for [Example 6.5](#)

60 Hz		40 Hz		45 Hz		50 Hz		55 Hz		65 Hz		70 Hz	
q_l (bpd)	FBHP (psi)	q_l (bpd)	FBHP (psi)	q_l (bpd)	FBHP (psi)	q_l (bpd)	FBHP (psi)	q_l (bpd)	FBHP (psi)	q_l (bpd)	FBHP (psi)	q_l (bpd)	FBHP (psi)
2,400	252	1,600	1,614	1,800	1,325	2,000	1,002	2,200	644	2,600	−174	2,800	−634
3,000	481	2,000	1,717	2,250	1,455	2,500	1,162	2,750	837	3,250	94	3,500	−324
3,500	709	2,333	1,820	2,625	1,585	2,917	1,321	3,208	1,029	3,792	360	4,083	−16
4,000	1,073	2,667	1,984	3,000	1,791	3,333	1,575	3,667	1,336	4,333	787	4,667	478
4,600	1,632	3,067	2,235	3,450	2,108	3,833	1,965	4,217	1,807	4,983	1,442	5,367	1,236

FBHP, flowing bottomhole pressure.

Now the frictional pressure drop in the tubing string is calculated. From Fig. A.1 in the Appendix a friction value of 76 psi/1,000 ft is read, giving a total head loss of

$$\Delta H_{fr} = 76.5,000/1,000 = 380 \text{ ft.}$$

Converting the previous value to pressure follows

$$\Delta p_{fr} = 0.433 \times 380 = 165 \text{ psi.}$$

Now Eq. (6.15) can be applied to find the corresponding *FBHP* of the well:

$$FBHP = 100 + (6,000 - 5,500)0.433 + 165 = 4,81.5 \text{ psi.}$$

The previous calculations are repeated for the other points on the 60-Hz curve and then for other frequencies; detailed results are displayed in Table 6.4. A graphical presentation of the NODAL analysis results is given in Fig. 6.8. The liquid rates achievable when driving the ESP system with different electrical frequencies are represented by the intersections of the IPR curve and the performance curves of the well-ESP system.

As seen, this kind of analysis permits a quick evaluation of the possible well rates from an existing ESP installation for different well inflow performance behaviors. This is because only the **IPR curve** changes when well inflow parameters vary with production time; the performance curves of the well-ESP system do not change and must not be recalculated.

The previous example assumed a **constant** wellhead pressure and was not a **full** NODAL analysis. If the complete production system is considered for a full analysis, then the effect of the **flowline** must also be included. Assuming a separator pressure of 100 psi and a flowline length and diameter of 4,000 ft and 2 in, respectively, the performance curves of the complete system change as given in Fig. 6.9. As seen in the figure, the production rates attainable from the same well for the different electrical frequencies have decreased by about 500 bpd as compared to the constant wellhead pressure case given in Fig. 6.8. This shows that the change in wellhead pressure due to the pressure drop in the flowline must always be considered in NODAL analysis.

Fig. 6.9 also includes two additional IPR curves, indicated by dashed lines, representing a $\pm 20\%$ error **margin** in the value of the well's *PI*. The effects of *PI* errors on the well's liquid rate can be evaluated based on the chart. It must be noted that the performance curves for different electrical frequencies are plotted using **published** head performance curves of the ESP used. If the actual pump does not perform according to its published head-rate curve because of mechanical **wear** or other reasons, the performance curves for different frequencies should be modified. Thus the effects of possible errors in (1) inflow performance information and/or (2) pump performance on well liquid rates can be evaluated.

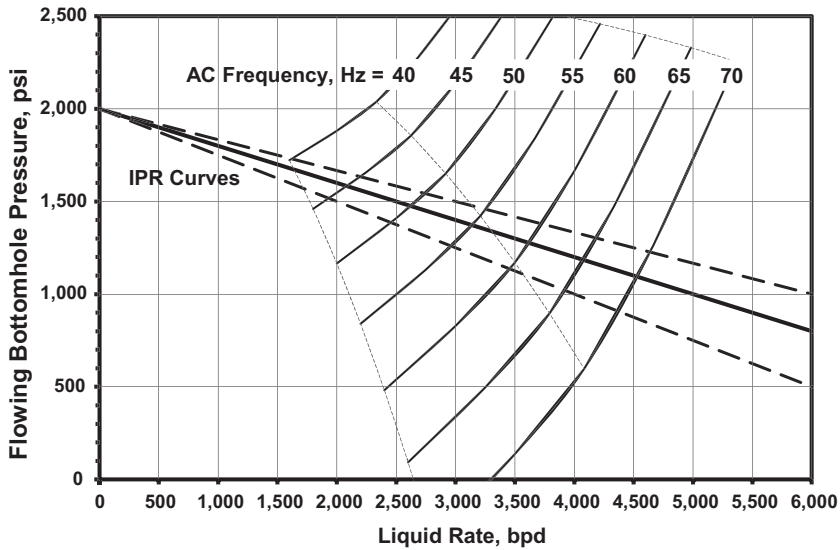


FIGURE 6.9 Results of a full NODAL analysis for Example 6.5. IPR, inflow performance relationship.

6.2.2.2 Multiphase Cases

When the ESP produces a **multiphase** mixture, the pressure distributions in the tubing string and in the annulus can no longer be described by constant pressure gradients. Just as before, multiphase pressure drop calculations are necessary to develop the performance curve of the formation–ESP subsystem. To prepare the *FBHP*–rate plots, the *FBHP* is found from a modification of Eq. (6.15), where a constant wellhead pressure is assumed:

$$FBHP = WHP + \Delta p_t - \Delta p_{\text{pump}} + \Delta p_c \quad (6.16)$$

where

- FBHP* = flowing bottomhole pressure, psi
- WHP* = tubinghead pressure, psi
- Δp_t = multiphase pressure drop in the tubing string, psi
- Δp_{pump} = pressure increase developed by the pump, psi
- Δp_c = multiphase pressure drop in the annulus, psi.

The situation is further complicated by the fact that the multiphase pressure drop in the tubing string, Δp_t , can only be found if the *GLR* valid for the flow in the tubing is known. This in turn is a function of

- the *PIP* because the amount of free gas present at the pump suction is determined by the flowing pressure at that level,

- the amount of gas transferred to the annulus because of **natural** gas separation,
- the amount of gas **separated** by an optional rotary gas separator.

Direct determination of the *FBHP* is, therefore, impossible and an **iterative** process is required. The main steps of the iterative calculation model are described in conjunction with Fig. 6.10. The flowchart illustrates the determination of the performance curve of the formation–ESP subsystem for a given electrical frequency (constant pump speed).

After selecting a series of liquid flow rates, the first rate, q_l , is selected. From an **assumed** *GLR*, GLR_a , the multiphase pressure drop, Δp_t , between the wellhead and the pump setting depth is determined by using a properly chosen vertical multiphase flow calculation model. This defines the pressure at the pump **discharge**; the pressure at the pump intake, *PIP*, is found by deducting the pressure increase developed by the ESP from the discharge pressure.

Based on the *PIP*, the **free**-gas volume at pump suction conditions can be calculated next. Considering the efficiencies of the **natural** and **forced** gas separations the amount of gas **ingested** by the pump, q'_{ing} , can be found. This, in turn, determines the *GLR* valid for the multiphase flow in the tubing string.

The calculated and the previously assumed *GLR* values are compared and, if in agreement, the right value is found. Otherwise a new iteration step is required and the process is repeated with the calculated value of *GLR*.

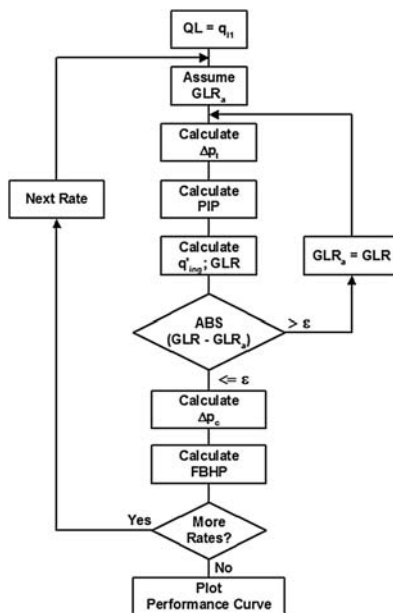


FIGURE 6.10 Flowchart of formation–pump performance curve determination. *FBHP*, flowing bottomhole pressure; *GLR*, gas–liquid ratio; *PIP*, pump intake pressure.

After the proper *GLR* value was found, the **multiphase** pressure drop, Δp_c , in the casing string between the pump setting depth and the depth of the perforations is determined. The well's *FBHP* can be calculated from Eq. (6.16). When *FBHP* values for all the assumed liquid rates are known, plotting of the performance curve for the given driving frequency is possible. The whole process must be repeated for other possible frequencies if a **VSD unit** is used to drive the ESP installation.

The actual liquid rate(s) occurring in the ESP installation is found, as before, by superimposing on this plot the well's **IPR curve** and finding the intersection(s) of the two curve sets. Owing to the complexity of this type of NODAL analysis, the use of computer program packages is needed.

6.3 DETERMINATION OF WELL INFLOW PERFORMANCE

The knowledge of the producing well's **inflow performance** is crucial for the design and analysis of any kind of artificial lift installations. In case of ESP installations, however, this factor is **essential** because of the special features of this lift method. As discussed in the previous chapters, ESP units can be applied economically in relatively **narrow** ranges of liquid production rates only, depending on the features of the centrifugal pump used. Improper information on the possible liquid rate from the well may cause the pump to operate outside the **recommended** liquid rate range and this leads to an abrupt loss of efficiency as well as mechanical failures. Consequently, the exact knowledge of the well's liquid rate capacity is an absolute **necessity** for the proper design of an ESP installation.

The importance of knowing the well's IPR curve at the time of installation design is shown by **Robinson and Divine [10]** who stated that based on their 4-year experience in well testing, only 60% of the original ESP designs proved to be right and the rest needed redesigning.

The following sections describe the ways how an ESP unit can be used to conduct a **production test** for the determination of the well's inflow performance curve. The results of such tests can be utilized

- to initially select the proper ESP unit to produce the well or
- to evaluate the operation of an existing installation, with the objective of determining the proper ESP equipment.

6.3.1 The Conventional Method

If using a conventional ESP installation where (1) the unit is connected to an electrical power supply having a constant frequency and (2) the well is producing liquid only, the well testing procedure originally developed by the **REDA Co.** can be used [11,12]. The test is based on the fact that for **short** periods of 1 or 2 min, ESPs can be operated safely with the tubing valve **closed**

without damaging the unit. Under such conditions the ESP develops the “shut-in” head valid at zero flow rate, easily found from the pump’s performance curve. By measuring the wellhead pressure and assuming that the dynamic liquid level did not change during the short shut-in period, one can calculate the actual bottomhole pressure and the inflow parameters, as detailed in the following.

The only equipment needed for the well test is a pressure **gauge**, installed upstream of the tubing valve on the wellhead. The steps of the test are given in the following.

1. **Shut** down the electrical supply to the ESP unit for a sufficient time necessary to allow the **static** fluid level to be reached. In wells not being produced, fill up the tubing completely with well fluids.
2. **Close** the tubing valve and start the ESP unit. To prevent damage to the submersible pump, the shut-in time must not be longer than 2 min.
3. Record the **stabilized** wellhead pressure, WHP_1 .
4. Open the tubing valve and allow the ESP unit to produce the well. Let the well produce until conditions **stabilize**.
5. Measure the stabilized liquid production rate q_l .
6. Close the tubing valve again and **record** the wellhead pressure, WHP_2 . As before, the well must not be shut-in for a period longer than 2 min.

The evaluation of test results starts with describing the conditions in **step 3**. As the tubing is closed at the wellhead and the liquid rate is zero, the ESP develops its **maximum** (shut-in) head. On the other hand, as no flow takes place in the tubing string, there is no **frictional** pressure drop occurring. Consequently, the head developed by the pump must balance the sum of the (1) head of the liquid column in the tubing string above the static liquid level and (2) the head corresponding to the measured wellhead pressure, WHP_1 . Expressing this condition in mathematical form, we get

$$H_0 = L_{st} + \frac{2.31 WHP_1}{\gamma_l} \quad (6.17)$$

where

H_0 = the pump’s shut-in head, ft

L_{st} = static liquid level in the annulus, ft

WHP_1 = measured tubinghead pressure, psi

γ_l = liquid specific gravity, —.

After opening the well and operating the ESP unit, a **stabilized** liquid rate must be reached. Now the liquid rate is measured, and the tubing valve is closed again. Assuming that during the short shut-in time the depth of the liquid level in the annulus does not change, we can write up the balance of the

pump's shut-in head and the sum of the dynamic level plus the head corresponding to the measured wellhead pressure, WHP_2 :

$$H_0 = L_{\text{dyn}} + \frac{2.31 \text{ } WHP_2}{\gamma_l} \quad (6.18)$$

where

H_0 = the pump's shut-in head, ft

L_{dyn} = dynamic liquid level in the annulus, ft

WHP_2 = measured tubinghead pressure, psi

γ_l = liquid specific gravity, —.

Solving the previous formulas for the liquid levels, we can calculate those unknowns as follows:

$$L_{\text{st}} = H_0 - \frac{2.31 \text{ } WHP_1}{\gamma_l} \quad (6.19)$$

$$L_{\text{dyn}} = H_0 - \frac{2.31 \text{ } WHP_2}{\gamma_l} \quad (6.20)$$

where

$L_{\text{dyn}}, L_{\text{st}}$ = dynamic and static liquid levels in the annulus, respectively, ft

H_0 = the pump's shut-in head, ft

WHP_1, WHP_2 = measured tubinghead pressures, psi

γ_l = liquid specific gravity, —.

Using the calculated liquid levels, the *SBHP* and *FBHP* can be evaluated from the following expressions:

$$SBHP = 0.433(L_{\text{perf}} - L_{\text{st}})\gamma_l \quad (6.21)$$

$$FBHP = 0.433(L_{\text{perf}} - L_{\text{dyn}})\gamma_l \quad (6.22)$$

where

$SBHP, FBHP$ = static and flowing bottomhole pressures, respectively, psi

L_{perf} = depth of the perforations, ft

$L_{\text{dyn}}, L_{\text{st}}$ = dynamic and static liquid levels in the annulus, respectively, ft.

The calculations discussed previously give two points on the well's **IPR curve**: one belonging to the *SBHP* and the other defined by the measured liquid rate, q_l , and the corresponding *FBHP* calculated from Eq. (6.22). Because the well produces incompressible liquid only, its **IPR curve** is a straight line, and the *PI* is easily found from the following formula:

$$PI = \frac{q_l}{SBHP - FBHP} \quad (6.23)$$

where

PI = productivity index, STB/day/psi

q_l = the well's stabilized liquid rate, STB/day

$SBHP$, $FBHP$ = static and flowing bottomhole pressures, respectively, psi.

Example 6.6

The ESP's shut-in head equals 4,000 ft. The wellhead pressure after starting the pump with the surface valve closed was 600 psi. After opening the tubing valve, the stabilized liquid rate was measured as 1,500 bpd. After shut-in, the measured wellhead pressure was 200 psi. Find the well's PI , if the depth of the perforations is 6,000 ft and the well fluid's specific gravity equals $\gamma_l = 0.85$.

Solution

The static liquid level is evaluated from Eq. (6.19):

$$L_{st} = 4,000 - 2.31/0.85 \cdot 600 = 2,369 \text{ ft.}$$

The dynamic liquid level is evaluated from Eq. (6.20):

$$L_{dyn} = 4,000 - 2.31/0.85 \cdot 200 = 3,456 \text{ ft.}$$

The well's $SBHP$ is determined from Eq. (6.21):

$$SBHP = 0.433(6,000 - 2,369)0.85 = 1,336 \text{ psi.}$$

The well's $FBHP$ is determined from Eq. (6.22):

$$FBHP = 0.433(6,000 - 3,456)0.85 = 936 \text{ psi.}$$

The PI of the well is found from Eq. (6.23) as

$$PI = 1,500/(1,336 - 936) = 3.75 \text{ bpd/psi.}$$

6.3.2 Use of Variable-Speed Drives

As discussed in the previous sections, an ESP installation driven by a VSD unit produces different amounts of liquid from the same well at different electrical frequencies. At any frequency, the ESP's and the well's liquid-producing capacities are **equal**, provided a continuous operation is possible. This can be easily seen in Fig. 6.11 in which the performance of a hypothetical formation and an ESP operating in the well is depicted, based on **NODAL analysis** calculations (compare to Fig. 6.8). The figure shows the performance of the ESP in the given well in the recommended operational ranges valid for different electrical **frequencies**. Thin dashed lines indicate the boundaries of the minimum and maximum liquid rate ranges as well as the variation of the BEPs. Inflow from the formation, as shown, represents a generalized case and is characterized by a **composite IPR curve** (see Section 2.2.3.3). Cooperation of the formation and the ESP installation is possible at the intersections of the two curve sets indicated by small circles.

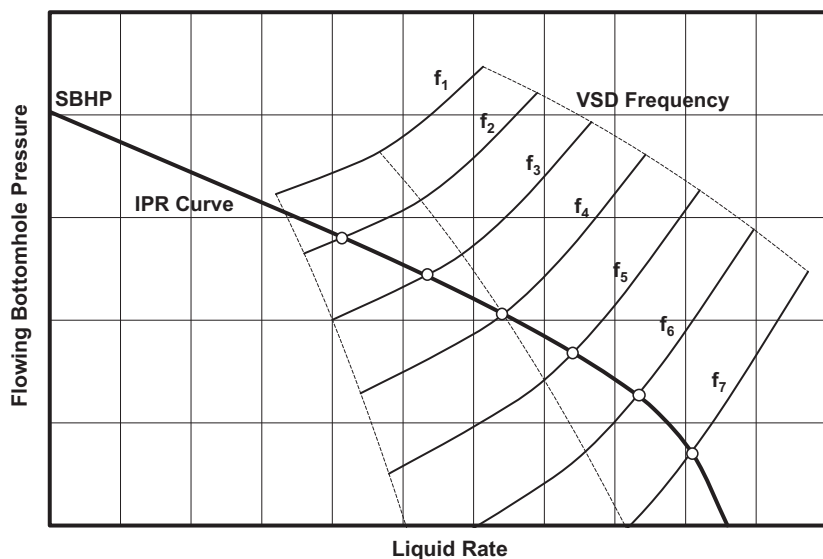


FIGURE 6.11 Schematic depiction of well testing results when using a variable-speed drive (VSD) unit. *IPR*, inflow performance relationship; *SBHP*, static bottomhole pressure.

The principle of **well testing** with VSDs is simple: the performance characteristics of the ESP being known, by running the unit at different speeds in a well with unknown inflow behavior while measuring the liquid rates, several points on the well's **IPR curve** can be determined. These points define the inflow properties of the well investigated. This approach, however, assumes that pump performance curves and well fluid properties are exactly known so that the pressure—rate curves representing the operation of the well plus the ESP can accurately be drawn. To prevent the effects of inevitable inaccuracies, it is recommended to either measure or calculate the flowing pressures belonging to the measured rates at different frequencies.

Production testing of oil wells using a VSD unit, therefore, consists of running the ESP at different frequencies, measuring the well's **stable** liquid rates, and calculating or measuring the *FBHPs*. If several such tests are conducted, the resulting *FBHP*—liquid rate pairs will describe the inflow performance of the well. Then to find the parameters of the **IPR curve**, curve-fitting techniques are used to find the proper formula (see Section 2.2) describing the measured data.

The well testing procedure described previously requires the use of the following equipment:

- a VSD unit,
- a complete ESP unit including surface and downhole components,
- a pressure sensor run into the well along with the ESP unit,
- an optional acoustic fluid level measuring device.

Most of these items are packaged in a **portable** well test unit that can be trailed to the well to be tested [13]. The acoustic device, in addition to providing a way to calculate bottomhole pressures from measured liquid levels, can be used to calibrate downhole pressure sensors [14].

The usual steps of a production test are the following:

1. The *SBHP* is determined, preferably by actual measurement with a downhole pressure sensor, or based on an acoustic measurement of the static liquid level.
2. The well is produced at a sufficiently low electrical frequency until **stabilization**.
3. After stabilization, liquid rate and *FBHP* are measured, preferably with the downhole pressure sensor. An acoustic survey may also be run to confirm the accuracy of the pressure measurement.
4. The VSD unit is set to a higher frequency and a new operating point is determined.

The main **advantages** of using VSD units for production testing of wells produced by submersible centrifugal pumps are the following:

- In contrast to previous practice when ESP units run at a constant frequency were used, inflow performance is determined without **pulling** the ESP equipment from the well [15].
- If the well test unit is connected to a SCADA system, **remote**-operated well testing is possible [16].
- Well test units can be leased [10,17] and only the equipment perfectly fitting actual well conditions has to be purchased.
- Installation designs based on the results of such well tests ensure that **optimum** conditions prevail with sufficiently long ESP run lives.
- Existing installations can be evaluated for design problems and the perfectly suitable proper ESP unit can be selected.

6.3.3 Calculation of Bottomhole Pressures

6.3.3.1 Introduction

Although in up-to-date ESP installations the actual value of bottomhole pressure is **continuously** measured and transmitted to the surface using downhole instrumentation, in the absence of pressure sensors, bottomhole pressures must be calculated. This is conveniently accomplished by summing up the hydrostatic pressures of the gas and liquid columns present in the well's annulus. The basis of this task is the determination of the annular liquid level by an **acoustic survey**.

Acoustic surveying, often called **well sounding** operates on the principles of the propagation and reflection of pressure waves in gases. Using a wave source, a pressure pulse (usually an acoustic impulse) is produced at the

surface in the casing-tubing annulus, which travels in the form of pressure waves along the length of the annular gas column. The pressure waves are **reflected** from every depth where a change in cross-sectional area occurs, caused by tubing collars, casing liners, well fluids, etc. The reflected waves are picked up and converted to electrical signals by a **microphone**, also placed at the surface, and recorded on paper or electronically. An evaluation of the reflected signals allows the determination of the **depth** to the liquid level in the well.

When the annular liquid level is known, the bottomhole pressure (static or flowing) is calculated as the sum of the following pressures:

- the measured **casinghead** pressure (*CHP*),
- the pressure exerted by the static **gas column** present in the annulus
- the hydrostatic pressure of the **liquid column** extending from the liquid level to the depth of the perforations.

Surface casing pressure, *CHP*, is measured with a high-precision pressure **gauge** or a deadweight tester. Accurate measurements are required because, in most cases, this component accounts for a large part of the final bottomhole pressure.

The gas column present in a well's annulus in an ESP installation is usually considered to be in a **semistatic** state, as flow velocities in the annular space are most often negligible. This is due to the large cross-sectional areas involved and the relatively low gas flow rates. In such cases, frictional losses can be disregarded and the pressure distribution in the column is affected by **gravitational** forces only.

Investigations of calculated pressure traverses in static gas columns have shown that pressure versus depth curves valid for a given gas specific gravity and a given surface pressure can very closely be fitted by **straight** lines. An explanation for this is the observation that the effects of increasing downhole temperatures and pressures on the gas density tend to **offset** each another. The use of **average** pressure gradients is, therefore, a viable approach to the calculation of gas column pressures.

Fig. 6.12 presents a chart constructed using the previously detailed logic, where pressure gradients are plotted for different gas specific gravities in the function of surface pressure. The chart was constructed with proper consideration of gas deviation factors and thus provides higher accuracy than hand calculations [18].

6.3.3.2 Annular Liquid Gradients

Under both static and flowing conditions the pressure exerted by the liquid column situated below the annular gas column is found from the gradients of the liquids present. The following sections give a detailed treatment of the proper calculation models to be applied.

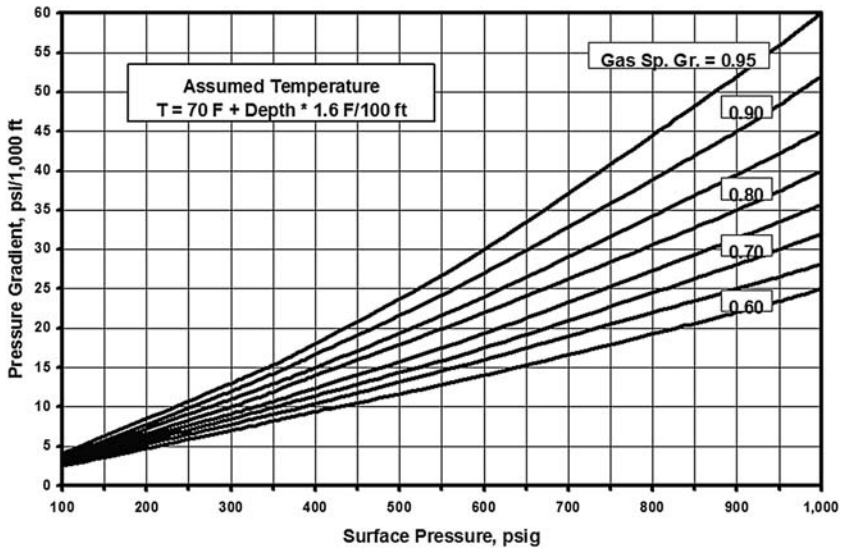


FIGURE 6.12 Static gas column pressure gradient chart [18].

6.3.3.2.1 Static Conditions

The liquid column's hydrostatic pressure, Δp_l , is mainly governed by the hydrostatic gradient of the annular liquid. Determination of this gradient poses no problems if pure oil is produced, but water-cut oil production needs special considerations. As shown by McCoy et al. [19], the composition of the annular liquid column depends on the relation of the dynamic liquid level to pump setting depth, valid just before the well was shut down. A general rule is that in steady-state pumping conditions, because of the gravitational separation of oil from water, all liquid **above** the pump is **oil**. After the well is shut down, a water-cut liquid with approximately the same water–oil ratio (*WOR*), as measured in a test separator, flows below the oil column.

After the *WOR* distribution in the liquid column has been established, the hydrostatic pressure is calculated with the help of static gradients for the water and the oil. For approximate calculations, average gradients taken at an average column temperature and pressure may be used. More accurate bottomhole pressure predictions require **stepwise** calculations when the effects of actual pressure and temperature on oil and water densities can properly be accounted for.

Finally, the *SBHP* is found from the formula

$$SBHP = CHP + \Delta p_g + \Delta p_l \quad (6.24)$$

where

SBHP = static bottomhole pressure, psi

CHP = measured casinghead pressure, psi

Δp_g = hydrostatic pressure of the annular gas column, psi

Δp_l = hydrostatic pressure of the annular liquid column, psi.

6.3.3.2.2 Flowing Conditions

In wells with negligible gas production, the calculation of *FBHPs* is quite simple and is accomplished with the method described for static conditions. Usually, surface *CHP* is the main contributor to bottomhole pressure, but liquid column pressure can also be considerable. Again, it must be remembered that under stabilized pumping conditions, all liquid **above** the pump is **oil** because of gravitational separation.

If formation gas is also produced, the calculation of bottomhole pressures is not so straightforward because, in these cases, the casing annulus contains a **gaseous** liquid column. It is customary to assume that most of the gas produced from the formation enters the annulus, and this gas continuously **bubbles** through the annular liquid column and leaves the well at the surface. The main effect of the produced gas is the **reduction** of the hydrostatic gradient of the annular liquid. In light of this, the accurate calculation of *FBHPs* heavily depends on the proper description of this phenomenon.

In the well's annulus, a special case of multiphase flow takes place: formation gas continuously bubbles through a **stagnant**, static liquid column. This column contains **only oil** above the pump setting depth, because of gravitational separation. Below that level the liquid's water content is much higher than the well's producing *WOR*; practically a column of formation **water** occupies the annulus between the pump setting depth and the perforations [20]. Depending on the gas volumetric rate, the flow patterns observed are bubble flow at low gas volumes and slug flow at higher gas rates. In both cases, the **density** of the mixture can be found from the basic formula:

$$\rho_m = \rho_l \varepsilon_l + \rho_g \varepsilon_g \quad (6.25)$$

where

ρ_m = two-phase mixture density, lbs/cu ft

ρ_l, ρ_g = liquid and gas densities, respectively, lbs/cu ft

ε_l = liquid holdup, —

ε_g = gas void fraction, —.

The contribution of gas density to mixture density is **negligible** and, therefore, can be deleted from the equation. As the hydrostatic pressure gradient is directly proportional to liquid density, the gradient of the gassy liquid column can be expressed with the gradient of the gas-free liquid as given below:

$$grad_m = grad_l \varepsilon_l \quad (6.26)$$

where

$grad_m$ = hydrostatic gradient of the gassy annular liquid, psi/ft

$grad_l$ = hydrostatic gradient of gas-free annular liquid, psi/ft

ε_l = liquid holdup, —.

The liquid holdup, ε_l , can be considered as a correction of the liquid gradient; hence, it is also called **gradient correction factor**. Owing to its practical importance in *FBHP* calculations, several correlations are available to determine its value.

The technique proposed by McCoy et al. [21] and used generally [22] was based on field measurements and involves the use of the chart reproduced in Fig. 6.13.

Application of this model eliminates the often-difficult determination of annular gas flow rates. Instead of using the gas flow rate as the input variable, a quick measurement on the wellhead is required. For this, the casing valve is **closed** in for a short period, while the pump is operating and the casing

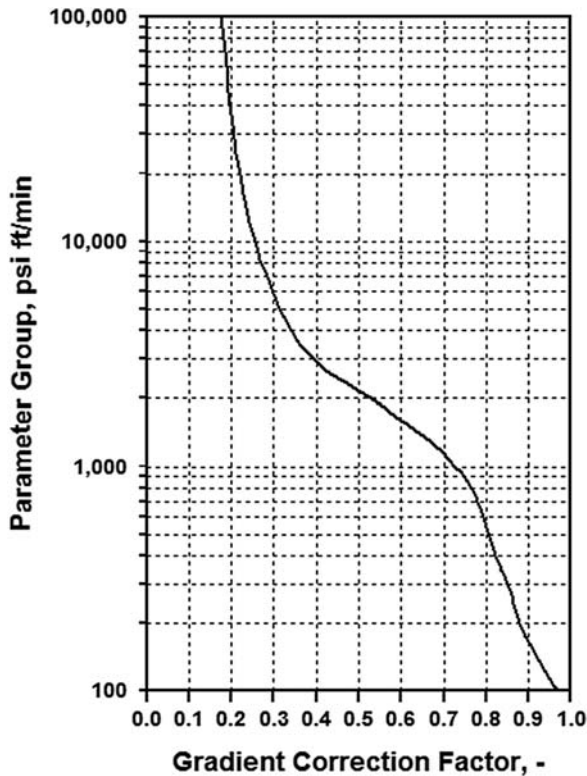


FIGURE 6.13 Liquid gradient correction factor correlation [21]. Copyright 1988, Society of Petroleum Engineers Inc. Reproduced with permission of SPE. Further reproduction prohibited without permission.

pressure **buildup** rate is recorded. For best results, a minimum pressure increase of 10 psi, or a 10-min test period should be used. From these data, the following parameter group is calculated:

$$[L_{\text{dyn}} + (1 - \varepsilon_l)h_l] \frac{dp}{dt} \quad (6.27)$$

where

L_{dyn} = dynamic liquid level in annulus, ft

ε_l = liquid holdup, —

h_l = height of the gaseous liquid column, ft

dp = casinghead pressure buildup, psi

dt = pressure buildup time, min.

The group just calculated is used in the correlation depicted in Fig. 6.13 to find the gradient correction factor. The calculation of the correction factor, however, requires an **iterative** process because ε_l figures in the corrected dynamic liquid level (see Eq. 6.27), which, in turn, is used to correlate ε_l in Fig. 6.13. Usually, only a few iterations are necessary to converge to the proper value of the gradient correction factor, if a starting value of $\varepsilon_l = 1$ is used.

Accurate bottomhole pressure predictions require **stepwise** calculations when the effects of actual pressure and temperature on oil and water densities can properly be accounted for. For approximate calculations, **average** gradients taken at an average column temperature and pressure may be used. At least two sections should be considered: (1) the **oil** column above the pump setting depth and (2) the **water-cut** column below the pump setting depth. The calculation of liquid column pressures is based on the following equation:

$$\Delta p_l = \Delta h \text{ grad}_l \varepsilon_l \quad (6.28)$$

where

Δp_l = liquid column pressure, psi

Δh = liquid column height, ft

grad_l = liquid gradient, psi/ft

ε_l = liquid holdup, —.

The well's **FBHP** is the sum of the three components described earlier and can be calculated similarly to the static pressure:

$$FBHP = CHP + \Delta p_g + \Delta p_l \quad (6.29)$$

where

FBHP = flowing bottomhole pressure, psi

CHP = measured casinghead pressure, psi

Δp_g = hydrostatic pressure of the annular gas column, psi

Δp_l = hydrostatic pressure of the annular liquid column, psi.

Example 6.7

Calculate the *FBHP* in a well with the following parameters.

Dynamic liquid level = 5,500 ft

Formation depth = 6,200 ft

Pump setting depth = 6,000 ft

Casinghead pressure = 150 psi

Gas SpGr = 0.75

Pressure buildup time = 10 min

Oil SpGr = 0.85

Pressure buildup = 7 psi

Water SpGr = 1.0

Water cut = 50%

Solution

The pressure gradient of the gas column is found from Fig. 6.12 at a surface pressure of *CHP* = 150 psi as 4 psi/1,000 ft; the gas column pressure is thus

$$\Delta p_g = 5,500 \text{ ft} / 1,000 = 22 \text{ psi}.$$

Now calculate the gradient correction factor for the oil column, $h_l = 500$ ft, above the pump setting depth. First assume a starting value of $\varepsilon_l = 1.0$ and find the correlation parameter from Eq. (6.27):

$$[L_{\text{dyn}} + (1 - \varepsilon_l)h_l]dp/dt = [5,500 + 0]7/10 = 3,850 \text{ psi ft/min}.$$

The correction factor read from Fig. 6.13 equals $\varepsilon_l = 0.36$.

Using this value to find a new correlation parameter, we get

$$[L_{\text{dyn}} + (1 - \varepsilon_l)h_l]dp/dt = [5,500 + (1 - 0.36)500]7/10 = 4,074 \text{ psi ft/min}.$$

The correction factor read from Fig. 6.13 equals $\varepsilon_l = 0.33$.

A new iteration step is required using this value:

$$[L_{\text{dyn}} + (1 - \varepsilon_l)h_l]dp/dt = [5,500 + (1 - 0.33)500]7/10 = 4,085 \text{ psi ft/min}.$$

The correction factor read from Fig. 6.13 equals $\varepsilon_l = 0.33$, which is the final converged value.

For simplicity, the change of oil density with temperature is neglected (accurate calculations would include this effect) so the hydrostatic pressure of the oil column, using the gradient correction factor just calculated is

$$\begin{aligned} \Delta p_o &= 0.433(L_{\text{set}} - L_{\text{dyn}})Sp.Gr. \varepsilon_l = 0.433(6,000 - 5,500)0.85 \times 0.33 \\ &= 60.7 \text{ psi}. \end{aligned}$$

Next calculate the gradient correction factor for the water column, $h_l = 200$ ft, below the pump setting depth. First assume a starting value of $\varepsilon_l = 1.0$ and find the correlation parameter from Eq. (6.27):

$$[L_{\text{dyn}} + (1 - \varepsilon_l)h_l]dp/dt = [5,500 + 0]7/10 = 3,850 \text{ psi ft/min.}$$

The correction factor read from Fig. 6.13 equals $\varepsilon_l = 0.36$.

Using this value to find a new correlation parameter, we get

$$[L_{\text{dyn}} + (1 - \varepsilon_l)h_l]dp/dt = [5,500 + (1 - 0.36)200]7/10 = 3,940 \text{ psi ft/min.}$$

The correction factor read from Fig. 6.13 equals $\varepsilon_l = 0.35$.

A new iteration step is required using this value:

$$[L_{\text{dyn}} + (1 - \varepsilon_l)h_l]dp/dt = [5,500 + (1 - 0.35)200]7/10 = 3,941 \text{ psi ft/min.}$$

The correction factor read from Fig. 6.13 equals $\varepsilon_l = 0.35$, which is the final converged value.

The hydrostatic pressure of the water column, using the gradient correction factor just calculated is

$$\Delta p_w = 0.433(L_{\text{form}} - L_{\text{set}})Sp.Gr. \quad \varepsilon_l = 0.433(6,200 - 6,000)1.0 \cdot 0.35 = 30.3 \text{ psi.}$$

The *FBHP* is found from Eq. (6.29):

$$FBHP = 150 + 22 + 60.7 + 30.3 = 263 \text{ psi.}$$

6.4 POWER EFFICIENCY OF ESP INSTALLATIONS

The economics of an ESP or any other type of artificial lift installation can be best evaluated by considering the **lifting costs** in monetary units per volume of liquid lifted. As ESP units are driven by electrical motors, the biggest part of the operating cost is represented by the **electrical** power bill. Because of the worldwide trend of electrical power price increases, this single item has become a **decisive** constituent of the total operating expenditure in ESP operations. Consequently, the never-ending pursuit of operating cost reduction can be translated to the reduction of energy losses occurring downhole and on the surface.

The power **consumed** by the ESP system comprises, in addition to the energy required to lift well fluids to the surface, all the energy **losses** occurring in downhole and surface equipment. Therefore any effort to reduce these losses should start with a perfect understanding of their nature and magnitude. The following discussion introduces the possible sources of energy losses along the flow path in the ESP system.

6.4.1 Power Flow in the ESP System

The ESP installation's useful **output** work is done by the centrifugal pump when it lifts a given amount of liquid from the pump setting depth to the surface. This work is usually described by the so-called useful **hydraulic power**, P_{hydr} , and can be calculated as the power consumed for increasing the potential energy of the liquid pumped. The following formula, recommended by **Lea et al.** [23], can be applied to any artificial lift installation and gives a standardized way to compare the effectiveness of different installations:

$$P_{\text{hydr}} = 1.7 \cdot 10^{-5} q_l (0.433 \gamma_l L_{\text{pump}} - PIP) \quad (6.30)$$

where

P_{hydr} = hydraulic power used for fluid lifting, HP

q_l = liquid production rate, STB/day

γ_l = specific gravity of the produced liquid, —

L_{pump} = pump setting depth, ft

PIP = pump suction pressure, called PIP, psi.

It can be demonstrated that in contrast to previous models, [Eq. \(6.30\)](#) excludes the power **wasted** for overcoming the wellhead pressure and other related hydraulic losses occurring in the tubing string. Therefore it represents the possible **minimum** of power required to lift well fluids to the surface. As its value is constant as long as the PIP is constant, it provides a standard way to compare the energy efficiency of the same system under different conditions or the efficiencies of different pumping systems.

At the other end of the ESP system, the electrical power **input** from the surface power supply, P_e , covers the energy requirements of the ESP installation, including the useful power used for fluid lifting and the energy losses occurring in the downhole and surface equipment. It is usually measured by a three-phase **power meter** at the wellsite or can be calculated from the system's kilovolt-ampere requirement (see [Eq. 5.29](#)) and the power factor:

$$P_e = \frac{\sqrt{3} U_{\text{surf}} I (\cos \varphi)_{\text{system}}}{1,000} = 1.732 \cdot 10^{-3} U_{\text{surf}} I (\cos \varphi)_{\text{system}} \quad (6.31)$$

where

P_e = total input electrical power, kW

U_{surf} = the system's required surface voltage, V

I = motor current, A

$(\cos \varphi)_{\text{system}}$ = power factor of the ESP system, —.

Among the parameters mentioned previously, only the system's **power factor** needs explanation. The ESP motor and the downhole electrical cable are connected in **series** and their combined power factor is found from the

individual power factors of the motor and the cable. As the usual ESP cable's **inductance** is low (one magnitude less than its resistance), the system's combined power factor is very close to the motor's power factor. Because of the maximum 1%–3% differences [24], the use of the ESP motor's power factor is justified.

Example 6.8

Find the ESP system's power factor for the case given in Example 5.3.

Solution

The following data are available from the solution to Example 5.3.

Motor nameplate voltage = 968 V

Motor current = 48 A

Motor loading = 86%

Cable size = AWG 4

Cable length = 5,000 ft

The impedance of the motor is found from **Ohm's Law** valid for three-phase electrical circuits as

$$Z = U_m / (30.5 I) = 968 / (1.732 \cdot 48) = 11.643 \text{ ohms.}$$

From motor performance curves (such as in Fig. 3.24) the motor power factor was found at the present load of 86% as $\cos \varphi = 0.82$.

The electrical resistance of the motor is found from the power factor as

$$R_m = Z \cos \varphi = 11.643 \cdot 0.82 = 9.547 \text{ ohms.}$$

The motor's inductive reactance is calculated with $\sin \varphi = (1 - \cos^2 \varphi)^{0.5} = 0.572$ as follows:

$$X_m = Z \sin \varphi = 11.643 \cdot 0.572 = 6.66 \text{ ohms.}$$

The AWG 4 cable's electrical properties are

$$\text{Resistance} = 0.441 \text{ ohms/1,000 ft}$$

$$\text{Reactance} = 0.037 \text{ ohms/1,000 ft.}$$

As the cable is 5,000 ft long, the ESP cable's parameters are

$$R_c = 5 \cdot 0.441 = 2.205 \text{ ohms}$$

$$X_c = 5 \cdot 0.037 = 0.185 \text{ ohms.}$$

The ESP motor and the cable are connected in series; therefore, the system's electrical properties are calculated as follows:

$$R_{\text{system}} = R_m + R_c = 9.547 + 2.205 = 11.752 \text{ ohms}$$

and

$$X_{\text{system}} = X_m + X_c = 6.66 + 0.185 = 6.845 \text{ ohms.}$$

The system's impedance from Eq. (2.28) is

$$Z_{\text{system}} = (R_{\text{system}}^2 + X_{\text{system}}^2)^{0.5} = (11.752^2 + 6.845^2)^{0.5} = 13.6 \text{ ohms.}$$

The system's power factor is now easily calculated:

$$(\cos \varphi)_{\text{system}} = R_{\text{system}}/Z_{\text{system}} = 11.752/13.6 = 0.864.$$

As seen, the difference between the system's power factor and the ESP motor's power factor (0.82) is 5.3% only; this justifies that the ESP system's power factor is practically identical to the motor's power factor.

Because the power efficiency of any system, in general, equals the ratio of the **useful** power to the **input** power, the powers defined previously enable one to calculate the efficiency of the ESP installation. The system's useful power is represented by the hydraulic power spent on fluid lifting, and the total energy input equals the surface electrical power requirement. In case the electrical power is actually measured by a power meter, the **efficiency** is easily calculated after converting the unit kilowatt to horsepower:

$$\eta_{\text{system}} = \frac{P_{\text{hydr}}}{\frac{P_e}{0.746}} = 0.746 \frac{P_{\text{hydr}}}{P_e} \quad (6.32)$$

where

η_{system} = power efficiency of the ESP system

P_{hydr} = hydraulic power used for fluid lifting, HP

P_e = total input electrical power, kW.

The formula represents the ESP installation's **overall power efficiency** and provides a quick way to evaluate the effectiveness of existing installations. In the absence of measured input power data, e.g., when designing a new installation, the system's input power requirement must be calculated. This is possible if proper estimations of the power losses occurring in the system are made.

The following sections describe the possible power losses in the different components of the ESP installation along with their calculation models.

6.4.2 Energy Losses and Efficiencies

In systems where energy losses of different nature in **series-connected** system components occur, the system's total efficiency can be broken down into **individual** efficiencies representing the different points in the energy flow. Total or overall system efficiency is then calculated as the product of the constituting efficiency items. In our case, one has to assign separate efficiency

figures to the individual kinds of energy losses occurring in the ESP system [23], as will be detailed in the following.

Classification of the energy losses in ESP systems can be made according to the place they occur; one can thus distinguish between **downhole** and **surface** losses. Another way to group these losses, and used in the following discussion, is based on their nature and categorizes them as **hydraulic** and **electrical**. Fig. 6.14 gives a graphical presentation of the typical energy losses in the ESP system where losses of electrical nature are indicated by shaded boxes.

6.4.2.1 Hydraulic Losses

The sources of energy losses of hydraulic nature are the tubing string, the backpressure acting on the well, the ESP, and the optional rotary gas separator.

6.4.2.1.1 Tubing Losses

Flow of the produced fluids to the surface involves frictional pressure losses in the tubing string; the power wasted on this reduces the effectiveness of the ESP

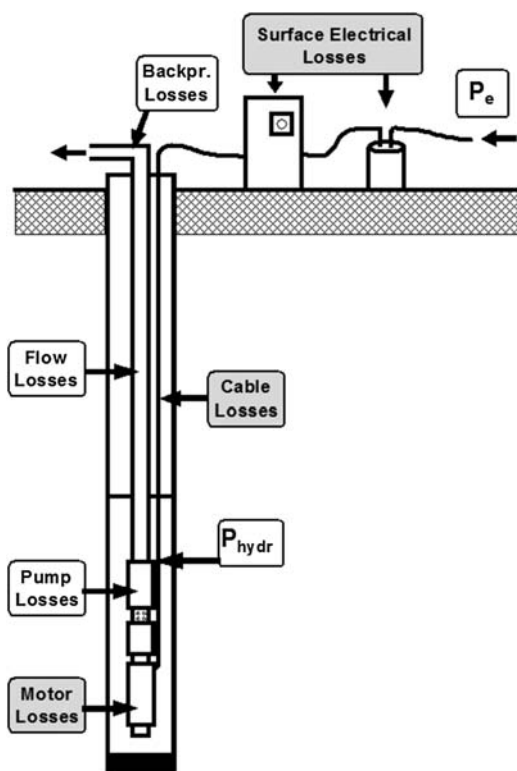


FIGURE 6.14 Energy flow in an ESP installation.

installation. If a single-phase liquid is produced, the frictional loss in the tubing string is determined from the total head loss, ΔH_{fr} , usually estimated from charts such as in Fig. A.1 in **Appendix A** and calculated using Eq. (5.7). The **power loss**, ΔP_{fr} in horsepower, associated with this frictional pressure drop is found from the following formula:

$$\Delta P_{fr} = 7.368 \cdot 10^{-6} \cdot q_l \cdot \Delta H_{fr} \cdot \gamma_l \quad (6.33)$$

where

ΔP_{fr} = frictional power loss in the tubing, HP

q_l = liquid production rate, bpd

ΔH_{fr} = frictional head loss in the tubing, ft

γ_l = specific gravity of the produced liquid, —.

If an efficiency describing the effect of the tubing losses is desired, one can express it from the input and output powers at the two ends of the tubing string:

$$\eta_{fr} = \frac{P_{hydr}}{P_{hydr} + \Delta P_{fr}} \quad (6.34)$$

where

η_{fr} = efficiency for tubing frictional losses

P_{hydr} = hydraulic power used for fluid lifting, HP

ΔP_{fr} = frictional power loss in the tubing, HP.

6.4.2.1.2 Backpressure Losses

The ESP unit has to work against the well's surface wellhead pressure, and the power consumed by overcoming this backpressure is **not included** in the useful power, as described in accordance with Eq. (6.30). The necessary power to overcome the wellhead pressure (in horsepower) is found from

$$\Delta P_{bp} = 1.7 \cdot 10^{-5} \cdot q_l \cdot WHP \quad (6.35)$$

where

ΔP_{bp} = power losses due to backpressure, HP

q_l = liquid production rate, bpd

WHP = wellhead pressure, psi.

Based on this power, one can define an efficiency value representing the relative amount of power **consumed** in the ESP system due to the wellhead backpressure as follows:

$$\eta_{bp} = \frac{P_{hydr} + \Delta P_{fr}}{P_{hydr} + \Delta P_{fr} + \Delta P_{bp}} \quad (6.36)$$

where

η_{bp} = efficiency for backpressure losses

P_{hydr} = hydraulic power used for fluid lifting, HP

ΔP_{fr} = frictional power loss in the tubing, HP

ΔP_{bp} = power losses due to backpressure, HP.

6.4.2.1.3 Pump Losses

Energy losses in the ESP are mostly of hydraulic nature, as described in Section 2.3.2.3, and are represented by published pump efficiency curves. In most cases the pump efficiencies, as given by the manufacturer, include the effect of the additional power required to drive the ESP unit's **protector**. Based on the actual pump efficiency the power lost in the pump (in horsepower) is easily found:

$$\Delta P_{\text{pump}} = BHP(1 - \eta_{\text{pump}}) \quad (6.37)$$

where

ΔP_{pump} = power loss in the centrifugal pump, HP

BHP = brake horsepower required to drive the pump, HP

η_{pump} = pump efficiency, —.

6.4.2.1.4 Power Loss in Gas Separator

In wells with substantial free-gas production, rotary separators are used to improve ESP operations. The power used to drive such rotary separators can be considered as **wasted** and a related efficiency can be defined as given in the following:

$$\eta_{\text{sep}} = \frac{BHP_{\text{pump}}}{BHP_{\text{pump}} + P_{\text{sep}}} \quad (6.38)$$

where

η_{sep} = gas separator efficiency, —

BHP_{pump} = brake horsepower required to drive the pump, HP

P_{sep} = brake horsepower required to drive gas separator, HP.

6.4.2.2 Electrical Losses

Electrical power losses in the ESP installation occur, starting from the motor and proceeding upward,

- in the ESP motor,
- in the power cable,
- in the surface equipment.

These losses **decrease** the power efficiency of the system and the appropriate efficiencies for each component can be defined as detailed in the following.

6.4.2.2.1 Motor Losses

The ESP motor converts the electrical energy input at its terminals into mechanical work output at its shaft. The efficiency of the energy conversion is characterized by the **motor efficiency**, published for available motors by the manufacturer. Energy loss in the motor is determined by motor efficiency and calculated as follows:

$$\Delta P_{\text{motor}} = BHP_{\text{motor}}(1 - \eta_{\text{mot}}) \quad (6.39)$$

where

ΔP_{motor} = power loss in the motor, HP

BHP_{motor} = motor power, HP

η_{mot} = motor efficiency, —.

The electrical power required by the motor, $P_{\text{motor } e}$, is found from the motor's mechanical power and its efficiency as follows:

$$P_{\text{motor } e} = 0.746 \frac{BHP_{\text{motor}}}{\eta_{\text{mot}}} \quad (6.40)$$

where

$P_{\text{motor } e}$ = electrical power input to the motor, kW

BHP_{motor} = motor power, HP

η_{mot} = motor efficiency, —.

6.4.2.2.2 Cable Losses

As the ESP motor is connected to the power supply through a **long** power cable, there is a considerable voltage **drop** occurring across this cable. The voltage drop creates a power loss proportional to the square of the current flowing through the system, as given by Eq. (5.25) and reproduced here:

$$\Delta P_c = \frac{3 I^2 R_T}{1,000} \quad (6.41)$$

where

ΔP_c = electrical power loss in the ESP cable, kW

I = required motor current, A

R_T = resistance of the power cable at well temperature, ohms.

An efficiency representing the amount of power **wasted** in the ESP cable can now be defined:

$$\eta_c = \frac{P_{\text{motor } e}}{P_{\text{motor } e} + \Delta P_c} \quad (6.42)$$

where

η_c = cable efficiency

$P_{\text{motor } e}$ = electrical power input to the motor, kW

ΔP_c = electrical power loss in the ESP cable, kW.

6.4.2.2.3 Surface Electrical Losses

The ESP installation's surface components include the surface cable, the switchboard, and the power transformer(s). These pieces of equipment are very **efficient** in transmitting the required electrical power to the ESP unit; their usual efficiencies are around $\eta_{\text{surf}} = 0.97$. The energy wasted in surface equipment can thus be found from

$$\Delta P_{\text{surf}} = P \frac{(1 - \eta_{\text{surf}})}{\eta_{\text{surf}}} \quad (6.43)$$

where

ΔP_{surf} = surface electrical power loss, kW

P = sum of the hydraulic power and power losses, kW

η_{surf} = efficiency of surface equipment, —.

6.4.3 System Efficiency

Based on the individual efficiencies of the ESP system components, one can easily write the **overall efficiency** of the installation as

$$\eta_{\text{system}} = \eta_{\text{bp}} \eta_{\text{fr}} \eta_{\text{pump}} \eta_{\text{sep}} \eta_{\text{mot}} \eta_c \eta_{\text{surf}} \quad (6.44)$$

It can be demonstrated that this formula gives identical results to the formulas calculated from the basic system efficiency equation given in Eq. (6.32), valid for known system power consumptions. The **advantage** of the detailed formula lies in providing the possibility of detecting system components where exceptionally great power losses may occur. Use of this formula, therefore, helps in achieving optimum economic conditions and in reducing fluid lifting costs.

Example 6.9

Evaluate the power losses and efficiencies in the ESP installation designed in Example 5.3.

Solution

First calculate the system's useful hydraulic power from Eq. (6.30) using data calculated in Example 5.3:

$$P_{\text{hydr}} = 1.7 E - 5 \ 1,300(0.433 \ 0.986 \ 4,900 - 313) = 39.3 \text{ HP.}$$

Frictional pressure drop in the tubing causes an energy loss calculated from Eq. (6.33):

$$\Delta P_{\text{fr}} = 7.368 E - 6 \ 1,310 \ 216 \ 0.986 = 2 \text{ HP.}$$

The related efficiency is found from Eq. (6.34):

$$\eta_{\text{fr}} = 39.3 / (39.3 + 2) = 0.95.$$

The energy loss due to the surface backpressure is found from Eq. (6.35) as

$$\Delta P_{\text{bp}} = 1.7 E - 5 \ 1,310 \ 80 = 1.8 \text{ HP.}$$

The efficiency calculated from this energy loss according to Eq. (6.36):

$$\eta_{\text{bp}} = (39.3 + 2) / (39.3 + 2 + 1.8) = 0.96.$$

The ESP's efficiency is read from its performance curve as, $\eta_{\text{pump}} = 0.60$.

The power lost in the pump is found from Eq. (6.37):

$$\Delta P_{\text{bp}} = 72(1 - 0.6) = 29 \text{ HP.}$$

As no rotary gas separator is needed, $\eta_{\text{sep}} = 1.0$, or the term can be dropped from the system efficiency formula.

The selected 456 series motor's efficiency is found from its loading. The required brake horsepower to drive the ESP system is 72 HP, whereas a motor with a nameplate rating of 84 HP was selected; the motor load is

$$\text{Motor load} = 72/84 = 0.857 = 85.7\%.$$

The motor's efficiency at this loading is found from the manufacturer's catalog as

$$\eta_{\text{motor}} = 0.84.$$

Using this efficiency, the electrical power loss in the motor is found from Eq. (6.39):

$$\Delta P_{\text{motor}} = 72(1 - 0.84) = 11.5 \text{ HP.}$$

The electrical power required by the motor is calculated from Eq. (6.40) as follows:

$$P_{\text{motor } e} = 0.746 \ 72 / 0.84 = 64 \text{ kW.}$$

Using the selected AWG 4 cable, the power loss across the cable was calculated in Example 5.3 as $\Delta P_c = 11.6$ kW, and the cable efficiency is thus, using Eq. (6.42),

$$\eta_c = 64 / (64 + 11.6) = 0.85.$$

Finally, surface electrical efficiency is assumed as $\eta_{\text{surf}} = 0.97$, and the power lost in the surface system is found from Eq. (6.43):

$$\begin{aligned}\Delta P_{\text{surf}} &= (0.746(39.3 + 2 + 1.8 + 29 + 11.5) + 11.6)(1 - 0.97) / 0.97 \\ &= 2.3 \text{ kW}.\end{aligned}$$

The system's power requirement can now be calculated by adding the powers consumed by the different components:

$$P_e = 0.746(39.3 + 2 + 1.8 + 29 + 11.5) + 11.6 + 2.3 = 76.2 \text{ kW}.$$

The ESP system's overall efficiency is found from Eq. (6.44).

$$\eta_{\text{system}} = 0.95 \cdot 0.96 \cdot 0.60 \cdot 0.84 \cdot 0.85 \cdot 0.97 = 0.378 = 37.8\%.$$

Fig. 6.15 shows the distribution of power in the different components of the ESP system for the example case.

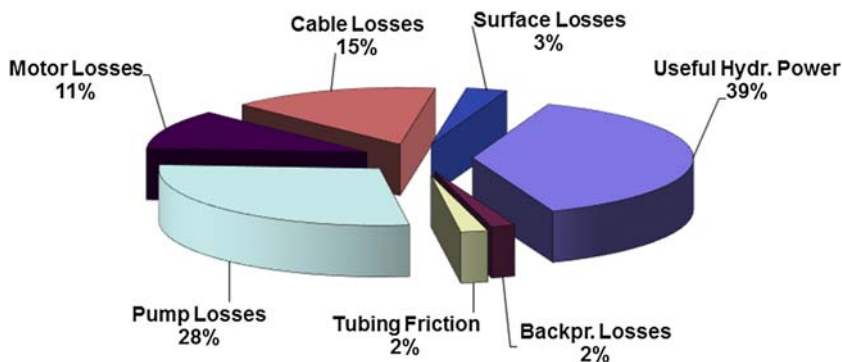


FIGURE 6.15 Power distribution in the electrical submersible pump system in Example 6.9.

Example 6.10

Evaluate the power losses and efficiencies in the ESP installation using a permanent magnet motor (PMM) as designed in Example 5.7.

Solution

Because the design in Example 5.7 was performed for the same conditions as in Example 5.3, the following data can be used from the previous example:

Hydraulic power, P_{hydr}	39.3	HP
Tubing friction loss, ΔP_{fr}	2	HP
η_{fr}	0.95	—
Tubing backpressure loss, ΔP_{bp}	1.8	HP
η_{bp}	0.96	—

The power lost in the pump is found from Eq. (6.37) and using a pump efficiency of 0.69 is

$$\Delta P_{\text{bp}} = 61(1 - 0.69) = 19 \text{ HP.}$$

As no rotary gas separator is needed, $\eta_{\text{sep}} = 1.0$, or the term can be dropped from the system efficiency formula.

The selected 406 series motor's efficiency at the current loading was found from the manufacturer's catalog as $\eta_{\text{motor}} = 0.93$.

Using this efficiency, the electrical power loss in the motor is found from Eq. (6.39):

$$\Delta P_{\text{motor}} = 67(1 - 0.93) = 4.7 \text{ HP.}$$

The electrical power required by the motor is calculated from Eq. (6.40) as follows:

$$P_{\text{motor } e} = 0.746 \, 67 / 0.93 = 53.7 \text{ kW.}$$

Using the selected AWG 6 cable, the power loss across the cable was calculated in Example 5.7 as $\Delta P_c = 2.4 \text{ kW}$, and the cable efficiency is thus, using Eq. (6.42),

$$\eta_c = 53.7 / (53.7 + 2.4) = 0.958.$$

Finally, the surface electrical efficiency is assumed as $\eta_{\text{surf}} = 0.97$, and the power lost in the surface system is found from Eq. (6.43):

$$\begin{aligned} \Delta P_{\text{surf}} &= [0.746(39.3 + 2 + 1.8 + 19 + 4.7) + 2.4](1 - 0.97) / 0.97 \\ &= 1.6 \text{ kW.} \end{aligned}$$

The system's power requirement can now be calculated by adding the powers consumed by the different components:

$$P_e = 0.746(39.3 + 2 + 1.8 + 19 + 4.7) + 2.4 + 1.7 = 53.9 \text{ kW.}$$

The ESP system's overall efficiency is found from Eq. (6.44).

$$\eta_{\text{system}} = 0.95 \, 0.96 \, 0.69 \, 0.93 \, 0.958 \, 0.97 = 0.544 = 54.4\%.$$

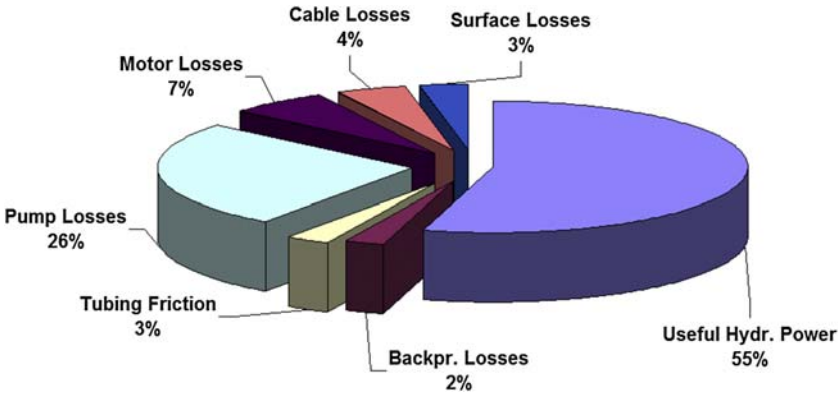


FIGURE 6.16 Power distribution in the electrical submersible pump system in Example 6.10.

Fig. 6.16 shows the distribution of power in the different components of the ESP system for the example case.

The previous two examples clearly indicate the advantages of using PMMs to drive ESP systems. The basic indicators of power use for the two cases are presented here; it can be observed that usage of a PMM permits an energy conservation of 27%, as compared to the use of an induction-type electrical motor.

Motor Type	Motor Losses (HP)	System Efficiency (%)	Power Requirement
			(kW)
Induction	11.5	37.8	76.2
PMM	4.7	54.4	53.9

6.5 OPTIMIZATION OF ELECTRICAL SUBMERSIBLE PUMP OPERATIONS

6.5.1 Introduction

Optimum ESP operations, in a general sense, occur when the **profit** from the production of oil is at a maximum. Because in most cases the pumping rate is prescribed and thus the revenues from oil are given, system optimization is translated into attaining the minimum of **operating costs**. Operating costs, however, are mainly related to the electrical power bill, which, in many cases, can reach the level of the initial investment cost in the unit's first year of operation itself.

Initial **investment** costs can be disregarded and only operational costs are considered in cases when (1) equipment from only one vendor are available and (2) longer run lives are expected. In such cases, optimum conditions are

reached when the ESP system's **power efficiency** is at the maximum. After investigating the relevant formula (Eq. 6.44), it can be concluded that power losses in some of the ESP system components are usually negligible, provided selection of those components was done by following a proper installation design procedure. These losses are listed in the following:

- negligible **hydraulic** losses in the tubing string, the wellhead, and the rotary gas separator,
- negligible **electrical** losses across surface equipment (switchboard, transformers, etc.).

It follows from the previous discussion that maximum power efficiency of the ESP system can be basically ensured by attaining the maximum **pump** and **motor** efficiencies while minimizing the power loss in the electrical **cable** [25]. As shown in Chapter 5, these objectives are accomplished through a proper equipment selection that yields

- the use of the pump with the best efficiency,
- a properly loaded motor,
- the proper size of cable, providing minimum life costs.

In conclusion, it may be stated that a proper installation design results in the system's **optimum** power efficiency and thus gives a **minimum** of operating costs for producing an oil well by ESP equipment.

In contrast to the previous discussion a complete economic analysis is necessitated when both **initial** and **operating** costs are considered.

6.5.2 Basics of Economic Optimization

Costs of operating an ESP installation can be classified into two broad groups:

- **CAPEX** (capital expenditure) or investment costs including capital **investment** and **installation** costs occurring only once at the start of operations.
- **OPEX** (operational expenditure) or operating costs are the periodic costs occurring during the life of the ESP system. Their greatest component is the cost of the **energy** continuously supplied to the system, although **maintenance** and **workover** costs can also be significant.

As the two types of expenditures occur at different times over the life of the ESP system, they can only be compared by taking into account the **time value of money**. This can be done in two ways: (1) the investment cost is **spread** over the life of the equipment or (2) the **present value** of the operating costs is calculated. Either way, the corrected costs can be compared and summed to find the total cost of the system's use [26,27].

Using the first of these solutions, the monthly cost related to the investment is found similar to the calculation of partial payments of loans over a period. This involves the use of the **capital recovery factor** (CRF) that defines the monthly payment necessary to **pay back** the capital outlay:

$$CRF = \frac{(1+p)^N}{(1+p)^N - 1} p \quad (6.45)$$

where

CRF = capital recovery factor

N = expected system life, months

$p = P/1,200$ = monthly interest rate, % per month

P = prime interest rate, % per year.

The total **monthly** expenditure can now be calculated as the sum of the capital payback and the monthly operating cost:

$$TMC = CRF CI + MOC \quad (6.46)$$

where

TMC = total monthly cost, \$ per month

CRF = capital recovery factor, 1 per month

CI = total capital investment, \$

MOC = monthly operating cost, \$ per month.

Let us now investigate the main influencing parameters of the total monthly cost.

CRF. Its value determines the monthly fraction of the original investment, considering the operating life of the equipment and the current interest rate. As shown in Fig. 6.17, CRF rapidly declines with increased system lives; the interest rate plays a secondary role. The figure demonstrates that the economics of ESP installations with long run lives is dominated by the production costs, i.e., the electrical power bill. Optimizations disregarding the investment costs, mentioned previously, are thus justified when relatively long service lives are expected.

Expected System Life. It has a major influence on the economics of ESP operations, as discussed in conjunction with Fig. 6.17. Although it varies with well conditions, data of past experience with similar ESP equipment under similar well conditions can help estimate its value.

Monthly Operating Cost. As already mentioned, the major component of the operating costs is the expenditure covering the **power** consumed by the ESP system. It is found from the specific cost of electricity and the ESP system's total power requirement (see Eq. 6.31). Proper installation design gives the parameters needed for the calculation of the required system power

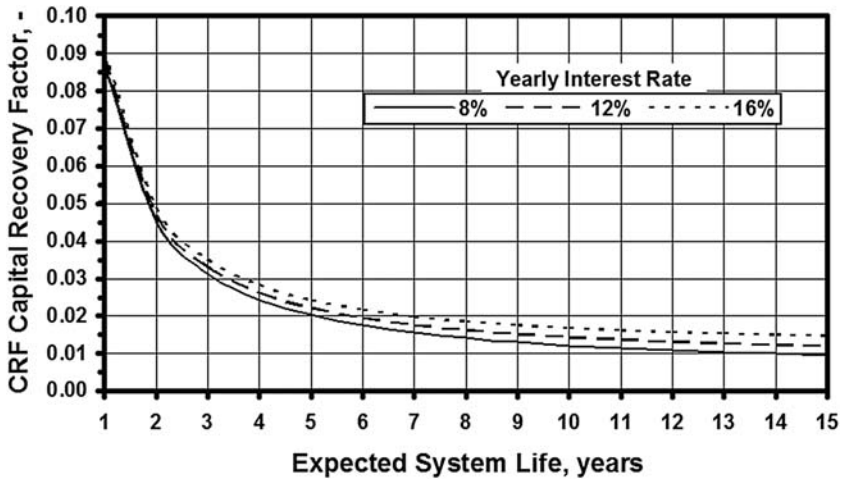


FIGURE 6.17 Change of capital recovery factor (CRF) with system life and interest rate.

in kilowatts, which, after multiplication with the cost of electrical energy, gives the monthly costs in \$ per month:

$$MOC = 1.247 U_{\text{surf}} I (\cos \phi)_{\text{system}} c_e \quad (6.47)$$

where

MOC = monthly operating cost, \$ per month

U_{surf} = the system's required surface voltage, V

I = motor current, A

$(\cos \phi)_{\text{system}}$ = power factor of the ESP system, —

c_e = cost of electrical power, \$/kWh.

The monthly cost represented by this formula does not include maintenance and workover costs and must be used in simplified economic analyses only.

Based on the previous discussions the selection of an **optimum** ESP installation is accomplished by following the main steps described here:

1. Detailed designs of ESP installations meeting production goals from the given well using equipment from several vendors, if available, are performed.
2. Investment costs of the possible cases are determined along with the power requirements.
3. Using the current interest rate and estimates on expected system lives, the total monthly cost of operation is found from Eq. (6.46).
4. The system with the lowest monthly cost is selected as the one giving an economically optimum operation of the given well.

Example 6.11

Select the economic optimum from the three cases designed for the conditions given in Example 5.3. Case #1 represents the original design detailed in the original example problem, case #2 is a cheaper unit, and case #3 is an advanced but more expensive one. Use an interest rate of 12%/year and an electricity cost of 5 cents/kWh. Output data of installation designs are as follows:

Case	Investment (\$)	Life (years)	U_{surf} (V)	I (A)	Power Factor
#1	60,000	4	1,110	48	0.864
#2	45,000	2	1,400	52	0.760
#3	70,000	5	1,100	45	0.860

Solution

Only case #1 is detailed in the following.

For comparison calculate the system efficiency first. The hydraulic power for this case was found as $P_{\text{hydr}} = 39.3$ HP in Example 6.9. The required surface electrical power is found from Eq. (6.31) as

$$P_e = 1.732 \cdot 10^{-3} \cdot 1,110 \cdot 48 \cdot 0.864 = 79.7 \text{ kW}.$$

System efficiency is calculated from Eq. (6.32)

$$\eta_{\text{system}} = 0.746 \times 39.3 / 79.7 = 0.368 = 36.8\%.$$

The economic calculations require finding of the *CRF* from Eq. (6.45), with a monthly interest rate of $p = 12/1200 = 0.01\%$ /month and a system life of $12 \cdot 4 = 48$ months:

$$CRF = 0.01(1 + 0.01)^{48} / \left[(1 + 0.01)^{48} - 1 \right] = 0.026 \text{ 1/month}.$$

The monthly investment cost is

$$CRF \text{ CI} = 0.026 \cdot 60,000 = 1,580 \text{ $/month}.$$

The monthly operating cost is calculated from Eq. (6.47):

$$MOC = 1.247 \cdot 1,110 \cdot 48 \cdot 0.864 \cdot 0.05 = 2,870 \text{ $/month}.$$

Finally, the total monthly expenditure from Eq. (6.46) is

$$TMC = 1,580 + 2,870 = 4,450 \text{ $/month}.$$

Calculation results for all cases are given in the following table.

Case	P_e (kW)	η_{system} (%)	<i>CRF</i> (1/month)	<i>CRF CI</i> (\$/month)	<i>MOC</i> (\$/month)	<i>TMC</i> (\$/month)
#1	79.7	36.8	0.026	1,580	2,870	4,450
#2	95.8	30.6	0.047	2,118	3,450	5,568
#3	73.7	39.8	0.022	1,668	2,654	4,323

As seen, the most expensive ESP unit, mostly because of its longer expected life, gives the optimum solution based on the total monthly costs.

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Chapter 7

Operation, Monitoring, and Surveillance of Electrical Submersible Pumping Systems

7.1 INTRODUCTION

After a proper design of the electrical submersible pump (ESP) installation the operator tries to achieve as long a run **life** as possible. To meet this goal, proper installation of the equipment and regular **monitoring** and **surveillance** of the unit's operation is required. Note that a typical ESP design is based on production of relatively constant flow rates and phase fractions. Mature fields and, more recently, unconventional and tight reservoirs experience significant and rapid declines in flow rates accompanied with changes in phase fractions. Such production behavior significantly challenges the ESP system performance. Although ESPs are generally reliable, such dynamic field conditions demand constant adjustment of operating parameters to prevent ESP failure and the resulting downtime as well as loss of production. Therefore monitoring and surveillance of production and ESP parameters is considered **standard** operating practice. Although proper monitoring can postpone system **failures**, failures eventually happen; these can be related to many possible general causes described in a further chapter.

This chapter covers the available methods of ESP monitoring and surveillance and the description of the devices used as well as discusses the use of the information received from such devices.

7.2 GUIDELINES FOR RUNNING AND PULLING

Although this section is not intended to be used instead of a manufacturer's detailed operations manual, **basic** guidelines on ESP equipment installation and running and pulling operations are presented. For detailed instructions the vendor's documentation should be studied along with the publication **API RP 11S [1]**.

7.2.1 Transportation to the Wellsite

Pieces of ESP equipment are normally transported in shipping **boxes** with flatbed trailers or trucks. For equipment shipped without boxes, care should be taken not to damage any special coatings that may exist on the housing of some equipment pieces. Boxes or equipment longer than 15 ft (especially ESP motors) must be handled so as not to be **bent**, as even the slightest permanent bend can substantially reduce the life of the equipment.

It must be checked that all equipment and material, as specified in the original order, is **present** on site. Any items **damaged** during transportation or dropped while handled at the wellsite should be sent **back** to the vendor for checking. Unloading the reel of ESP cable from the trailer is best done by using a heavy pipe through the center of the reel with heavy chains on both sides. The reel should be placed from 75 to 100 ft from the wellhead in such a way that the operator can have a **clear** view of the reel at all times during cable handling.

Surface equipment such as transformers and motor controllers have lifting lugs for unloading and should be moved using proper **straps** or cables. These pieces of equipment may be filled with oil so they must not be turned over. Normally they are installed 100 ft from the wellhead, with the motor controller fixed to a stable pad and the transformer to a pole. The transformer must not be positioned above the controller for safety reasons.

The surface **junction box** is normally placed at a distance of 15–25 ft from the **wellhead**, at a place that will not obstruct future workover operations. The **switchboard** should be located about 50–100 ft from the well, it must be grounded to the well casing or the power system neutral.

7.2.2 Equipment Installation

When installing ESP equipment, the first important item to check is whether the workover rig is lined up **correctly** over the well. Improper centering damages the electrical cable as it is run through the wellhead into the well. The next essential action is, especially in new installations, to check casing **restrictions**; for this a solid tool must be run to the setting depth of the motor. The tool should be longer and have a larger outside diameter than the motor and must not show signs of heavy **dragging** when run to the proper depth.

Lifting the equipment during assembly must be done **slowly** and with care to prevent dropping or hitting against hard objects. Proper slings should be used when tailing in equipment to the wellhead.

7.2.2.1 Motor Installation

With a properly sized **clamp** installed at the top of the motor, it should be picked up carefully without bending it. During lifting, the motor must not be allowed to swing or hit any objects.

Carefully lower the motor into the well until the motor clamp rests on the tubing slips or the blowout preventer unit. Shortly remove the motor's shipping **cap** to check the motor shaft for normal rotation and then replace the cap. Lift the motor until its lower fill valve is at a proper working height. By using a pump, **fill** the motor with the recommended **oil** from the lower fill valve until no air bubbles are visible and then replace the fill plug.

After lowering the motor until its clamp is resting on the wellhead remove the shipping cap completely and **add** more oil through the center of the motor's shaft until no more air bubbles are visible. Rotate the motor shaft when checking for air bubbles.

Next it must be assured that no air is **trapped** inside the motor because that may cause premature failure of the motor. For this the shipping cap must be firmly **tightened** and a pressure gauge must be installed to measure the motor's inside pressure. The motor is raised again with the fill pump installed and oil is added to the motor while checking the pressure gauge. If oil pressure **increases** after two to three turns on the fill pump, no air is trapped in the motor; otherwise there is still air present inside the motor.

With air still present, pressure in the motor is **released** by loosening its shipping cap and additional oil is **added** as before until motor pressure builds up after two to three turns on the fill pump. With the trapped air being removed, the fill valve is plugged and running into the well can continue.

7.2.2.2 Protector Installation

With a properly sized clamp installed at the top of the protector, it should be picked up carefully; it must not be allowed to swing or hit any object during this operation. After removing its lower shipping cap the **rotation** is checked with a spline wrench. Remove the motor's shipping cap and **clean** the motor flange and the bolt holes. The protector should be **slowly** lowered onto the motor while properly aligning the shafts and coupling assembly and then the flange bolts are **tightened** using the recommended torque.

7.2.2.3 Pump Installation

A lifting sub should be provided and installed on the pump for picking up the pump and for handling the ESP unit. A tubing sub is installed at the top of the pump and is tightened according to API specifications. The pump should be picked up **carefully** without bending it; during lifting, the pump must not be allowed to swing or hit any object. The pump's lower shipping cap must be removed and its **rotation** is checked with a spline wrench.

After removing the shipping cap from the top of the protector the **rotation** of the motor—protector assembly is checked. The protector's flange and bolt holes are cleaned and the pump is slowly lowered onto the protector flange; bolt holes are cleaned of any oil or debris before the bolts are **tightened** using the recommended torque.

7.2.2.4 ESP Cable Installation

With the cable reel on its support about 75–100 ft from the **wellhead** the cable should run **straight** to the tubing and it must not be **twisted** anywhere between. The reel axle should be at right angles to the wellhead; feed the cable from the **top** of the reel. Use a large cable wheel (48–54 in) at the rig when running the ESP cable. After inserting the motor lead extension (MLE) over the cable wheel raise the wheel over the ESP assembly to a **maximum** height of 10 ft above the motor. The shipping caps of the MLE and the motor connection must be removed and the MLE is positioned over the motor connection.

Now the insulation **resistances** to the ground and in-between phases on the motor and the cable must be checked using a measuring device. New or undamaged cable should read near infinity both phase to phase and phase to ground. If resistances are according to the manufacturer's recommendations, the cable pothead and the motor can be **bolted** together using new O rings. Cable bands should be installed for every 3 ft on the MLE to fix the cable to the equipment. Care should be taken not to damage the cable during installation; about 75% of cable damage is due to mishandling.

7.2.2.5 Final Measures

Using the recommended oil the protector chambers have to be **filled** up completely free of air. Do this **separately** for each chamber until no air bubbles appear at the top of the protector and then **replace** the fill plug.

The correct **direction** of rotation of the pump must be checked to ensure that the ESP unit is connected properly. This is usually done using a “phase rotation meter” that indicates whether the motor is properly connected to the three-phase electrical source. For this the pump shaft is turned in the proper direction by **hand**. If an incorrect rotational direction is indicated, any two **phases** in the motor controller or in the junction box are **changed** to attain the proper direction of rotation.

The direction of motor rotation can be detected by using a two-axial **vibration** sensor [2] that measures the vibrations of the operating motor. The device can be used not only to check for the proper rotational direction but also to find an inoperative motor.

If check and drain valves are used, the drain valve must be installed **above** the check valve in the tubing string. The check valve is normally positioned one stand of tubing **above** the submersible pump's discharge. The distance between the two valves is one stand, too. If no check valve is preferred, then a seating nipple as close to the pump as possible should be run to make pressure testing of tubing possible.

7.2.3 Running Equipment in the Well

Running of ESP equipment in the well must be done **slowly**, the recommended speed is limited to 1,500 ft of tubing per hour. At higher speeds the cable can

become **slack** and may cause problems when pulling the unit in the future. When running large-diameter units or in a **deviated** well, operational speeds lower than the recommended value are advised. When running ESP equipment in the well, always keep the tubing in the **middle** of the casing string.

During running operations, it is extremely important that proper cable **banding** procedures are followed. The recommended number of bands is three bands per tubing stand on #6 and #4 cable and four bands per stand on #2 or larger and all MLE cables. When applying cable bands the manufacturer's recommendations are to be followed, too tight bands **flatten** the cable armor and may lead to early failure. Loose banding, at the other hand, causes **slack** between bands and causes running and pulling problems. Good banding operations require the use of properly functioning banding and crimping **tools**.

The cable should never be **stretched** or jerked during installation because it may reduce the cable life. It is recommended to use a cable **spooler** and to always keep slack in the cable between the spooler and the cable. Never lower the tubing faster than the cable spooler can keep up with because the cable will stretch and become damaged.

The cable must not **twist** or run **crooked** from the cable spooler, through the cable guide wheel, or through the tubing slips. Do not drag the cable on the **ground** or let it become tight between the cable spool and the cable wheel.

To check for cable damage done during running operations, insulation **resistance** should be checked every 30 joints or after 1,000 ft of tubing is run in the well. Phase to ground readings should remain **balanced** on all three phases for an undamaged cable.

If a conventional low-pressure wellhead is used the cable armor needs to be **stripped** off where it passes through the wellhead. The unarmored portion must be wrapped by tape to form a proper seal around the cable in the wellhead. After the packing rubbers are installed the top of the wellhead is bolted or screwed on.

7.2.4 System Startup

Before the first operation of the ESP system the following parameters must be checked:

1. The actual line **voltage** complies with the nameplate voltage of the motor controller; the switchboard fuses are the correct size.
2. Overload and underload settings are properly **adjusted** in the motor controller.
3. All flow valves are in the **proper** positions (opened or closed) and the necessary relays and controls are properly set.

The tubing should be filled up with **fluid** in case a check valve is installed in the tubing. This ensures maintenance of a proper backpressure on the pump so that it can operate within its recommended flow rate range. With an empty

tubing string the pump initially produces a high fluid rate until liquids from the annulus are transferred to the tubing. This initial rate, being outside the pump's recommended operational range, can cause **damage** to the mechanical shaft seals of the protector or the motor, leading to contamination of the motor oil and early failures.

After the unit is started, the starting **amperage**, the pressure at the well-head, and the pumping rate is checked. Values of these parameters indicate if the submersible pump is running in the correct **direction** (If a "phase rotation meter" was previously used to find the proper rotation then the pump should run correctly.) Usually, checking the wellhead **pressure** can give a positive indication of the pump's actual direction of rotation; when the direction is right the wellhead pressure is higher than that in the wrong direction. If an incorrect rotational direction is indicated, any two phases in the motor controller or in the junction box are changed to attain the proper direction of rotation.

After shutdown, the unit should not be started for at least **30 min** if there is no check valve installed to enable the fluid in the tubing to flow back through the pump. The unit can then be restarted and the same checks are performed again to make sure the unit is running in the proper direction.

In wells producing **abrasives** or sand, the unit should not be shut down or the pump may become locked; normally the well cleans up in a short period and can then be shut in. For best results, cleaning sand and abrasives out of the well is recommended **before** running the equipment.

During the first few hours (sometimes a day), many parameters should be recorded: starting amperage, pumping rate, running amperage, and voltage; any changes in these are important. The stabilization of the ESP system sometimes may last for several **days**. Motor overload and underload settings may need to be adjusted during startup and after system stabilization.

In case of a shutdown due to **overload**, it is recommended that the down-hole equipment be checked electrically before trying to restart it; without this check the unit must not be restarted more than once. An underload shutdown normally indicates pumped off or gas locked conditions and necessitates the motor underload value to be adjusted.

Monitoring the operation of the system is best done by using a recording **ammeter**. Initially 24-h charts should be used but after stabilization the use of 7-day charts is sufficient. Additional measurements such as production tests and fluid level shots help determine if the unit is operating correctly and according to the original design.

7.2.5 Pulling ESP Equipment

Before pulling the ESP unit in case of failure collect all information on the equipment in the well and their **performance** in the past. Well test data, electrical parameters, pumping rate, fluid level, etc. should be evaluated.

Before the arrival of the pulling unit **disconnect** power to the cable at the vent box or lock out and tag the switchboard. It is also recommended to check the cable and motor resistance readings from phase to ground and phase to phase.

In case a drain valve is installed in the tubing string, the tubing should be **drained** according to the following procedure. If fluid is observed at the top of the tubing the steel bar to shear the break-off plug of the drain valve should be **dropped** in the tubing. If the tubing is not standing full do not drop the sinker bar until after the tubing is pulled up to where fluid is standing in the tubing. There are two reasons for not dropping the bar until fluid is observed: (1) a tubing **leak** may exist and its place can be found as the tubing is pulled to where the fluid is standing in the tubing and (2) dropping the steel bar in an empty tubing string may damage the **check valve** or the top of the ESP.

For pulling the ESP unit, basically the same procedures are followed as running it. The rig should be **centered** over the wellhead and the cable should be put on a cable reel. The cable is spooled **smoothly** onto the reel and damaged spots are marked.

The cable guide wheel should be positioned between 10 and 30 ft **above** the ground, and the weight of the cable over the wheel must not exceed 100 pounds. No tension must be allowed on the cable, and the cable must not be **dragged** on the ground.

Pulling the equipment must be done **slowly**, stopping enough for the cable bands to be cut using the proper cutting tools. The condition of the bands should be **noted**, and the missing bands must be detected. Care should be taken not to damage the **armor**, the jacket, or the insulation of the cable; any problems (corrosion, bad spots, etc.) should be closely inspected.

After pulled out of the well, all parts of the ESP equipment should be checked and handled carefully using the proper **clamps**. The pump is to be **flushed** with diesel oil before its shipping cap is fixed. The ESP components should be carefully placed in their shipping boxes.

If pulled because of a system failure, the equipment should be transferred to the **repair** shop to determine the cause of failure. A **teardown** procedure must be followed to discover the causes of failure.

7.3 PRODUCTION CONTROL

7.3.1 Introduction

The objective of any artificial lift design is to set up a lift system with a liquid producing capacity that **matches** the inflow rate from the well it is installed in. As the mechanical design of the lifting equipment is only possible in the knowledge of the probable liquid **rate**, the designer needs a precise estimate on the production rate attainable from the given well. Design inaccuracies or improperly assumed well rates can very easily result in a **mismatch** between

the designed and the actually produced liquid volumes. The main cause of discrepancies between these rates, assuming proper design procedures are followed, is the improper estimation of possible well **rates**, i.e., inaccurate data on well inflow performance. Under- or overdesign of artificial lift systems can lead to the following consequences:

- If the artificial lift equipment's capacity is greater than well inflow then the operational **efficiency** of the system cannot reach the designed levels; mechanical **damage** may also occur.
- In case the well's productivity is greater than the capacity of the lifting system, one loses the **profit** of the oil not produced.

Over- and underdesign of artificial lift installations happen in the industry very **often** and professionals know how to deal with them. Some lifting methods such as gas lifting or sucker rod pumping are relatively easy to handle since their lifting capacity can be adjusted in quite **broad** ranges after installation. ESP installations, however, do not tolerate design inaccuracies because any given ESP can only be used in a **specific**, quite restricted range of pumping rates. If used outside its recommended liquid rate range, the hydraulic efficiency of the pump rapidly deteriorates; efficiencies can go down to almost zero. In addition to the loss of energy and the consequent decrease in profitability, the ESP system, when operated under such conditions, soon develops mechanical **problems** that can lead to a complete system failure. The usual outcome is a **workover** job and the necessity of running a newly designed ESP system with the proper lifting capacity.

One common solution for overdesigned ESP systems is the use of production **chokes** at the wellhead. Installation of the choke, because of the high pressure drop that develops through it, limits the well's liquid rate so the ESP is forced to operate in its **recommended** pumping rate range. This solution eliminates the need for running a new ESP system of the proper capacity into the well and saves the costs of pulling and running operations. At the same time, however, the system's power **efficiency** decreases considerably because of the high hydraulic losses occurring across the surface choke.

The following sections investigate the **detrimental** effects of surface chokes on the power efficiency of ESP systems and present remedial actions. The analysis is provided for wells producing negligible amounts of free gas and is based on the application of NODAL analysis principles to describe the operation of the ESP system.

7.3.2 The Effects of Using Wellhead Chokes

7.3.2.1 The Need for Wellhead Chokes

Most ESP installations are designed to operate using electricity at a **fixed** frequency, usually 60 or 50 Hz. This implies that the ESP runs at a **constant** speed and develops different heads for different pumping rates as predicted by

its published performance curve. When designing for a constant production rate, a pump type with the desired rate inside its recommended capacity range is selected. The number of the required pump stages is found from detailed calculations of the required total dynamic head (TDH), i.e., the head required to lift well fluids to the surface at the desired pumping rate.

For an ideal design, when all the necessary parameters of the well and the reservoir are **perfectly** known the pump will produce exactly the design liquid rate because it will work against the design TDH. In this case the head **required** to overcome the pressure losses for moving well fluids to the separator is covered by the head **available** from the pump at the given pumping rate. This perfect situation, however, is seldom achieved; very often inaccuracies or lack of information on well inflow performance cause design errors and the well produces a rate **different** from the initial target.

The problem with the conventional design detailed earlier is that the ESP installation is investigated for a single design rate only and no information is available for cases when well parameters are in **doubt**. All these problems are easily solved if system (NODAL) analysis principles are used to describe the operation of the production system consisting of the well, the tubing, the ESP unit, and the surface equipment. NODAL analysis permits the calculation of the **necessary** pump heads for different possible pumping rates and the determination of the liquid **rate** occurring in the total system. This will be the rate where the **required** head to produce well fluids to the separator is equal to the head **developed** by the ESP run in the well.

Fig. 7.1 shows a schematic comparison of the conventional design with that provided by NODAL analysis. **Conventional** design calculates the TDH at the design rate **only** and selects the type of the ESP pump and the necessary number of stages accordingly. After selecting the rest of the equipment, the ESP unit is run in the well and it is only hoped that actual conditions were properly simulated, resulting in the well output being equal to the design liquid rate. If well inflow performance data were **uncertain** or partly/completely **missing** during the design phase then the ESP system's stabilized liquid rate will be **different** from the design target. **NODAL** calculations, however, can **predict** the required head values for different liquid rates, shown in Fig. 7.1 by the heavy dashed line. The well's actual production rate will be found where the required and the available (provided by the pump) heads are equal, at **point 1** in the figure.

In typical cases the actual liquid production rate is **greater** than the target value. This clearly indicates inaccuracies in the well performance data assumed during the design process. As the well's required production is usually **dictated** by reservoir engineering considerations, production of a greater rate is not allowed. The problem is caused, as shown in Fig. 7.1, by the **actual** head requirement (actual TDH) being much less than the calculated **design** TDH.

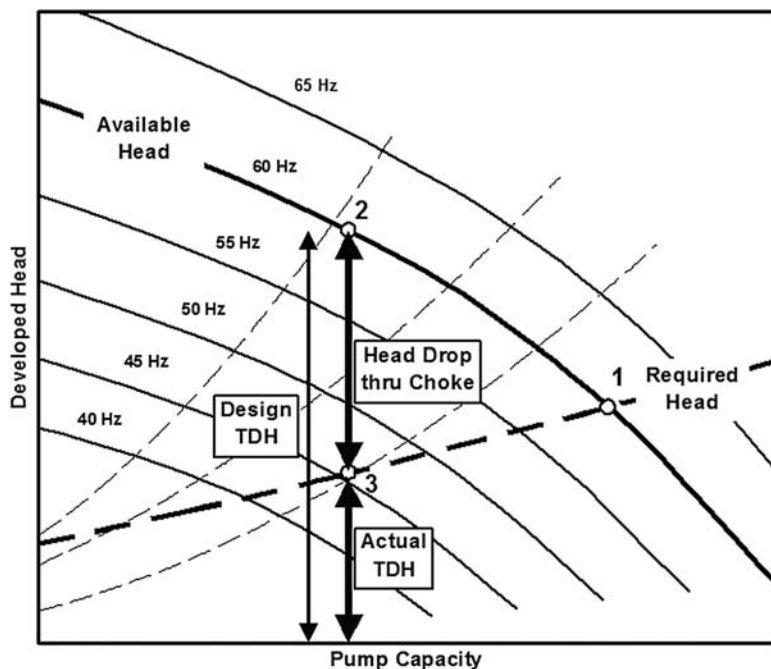


FIGURE 7.1 Explanation of the need of a wellhead choke.

The solution to the problem, if pulling the ESP equipment and replacing it with a properly designed one is not desired, is to place a production **choke** of proper diameter at the wellhead to restrict the liquid rate to the design target. At this rate, however, the installed pump develops the **designed** operating head as shown by point 2. The actual head required for lifting the well fluids to the surface, as found from NODAL calculations, is **less** than this value, so a sufficient head loss across the choke is needed. This head loss, found between **points 2 and 3**, must be sufficient to **supplement** the system's actual TDH to reach the TDH that was used for the original design. As a result the head requirement of the production system is artificially **increased** and the ESP pump is forced to produce the desired liquid rate.

The **detrimental** effect of choking an ESP unit is clearly indicated by the amount of power **wasted** through the surface choke. This can be calculated from the pumping rate and the amount of head loss across the choke, as given in Eq. (6.5) and reproduced here:

$$P_{\text{wasted}} = 7.368 \cdot 10^{-6} q_l \Delta H_{\text{choke}} \gamma_l \quad (7.1)$$

where

P_{wasted} = power wasted across choke, HP

q_l = pumping rate, bpd

ΔH_{choke} = head loss across surface choke, ft

γ_l = specific gravity of the produced liquid—.

The abovementioned power, of course, must be supplied by the electrical motor subjected to a higher than necessary load while driving the submersible pump. Because this power is **wasted**, the ESP system's power **efficiency** as well as the **profitability** of fluid production will decrease.

7.3.2.2 Use of Variable-Speed Drive Units

If, for any reason, the installation design is inaccurate and the ESP system, after installation, produces a higher rate than desired, the use of wellhead chokes is a **common** solution to control the well's production. If a variable-speed drive (VSD) is available, however, the **elimination** of the choke and its associated disadvantages can be accomplished. As shown in the schematic diagram in Fig. 7.1, by reducing the electrical **frequency** driving the ESP system to a level where the head **developed** by the pump is equal to the head **required** to produce the desired rate (**point 3**), the choke is no more needed to adjust the pumping rate.

When a VSD unit is used to control the ESP system's liquid rate the different components of the system behave differently as the driving **frequency** is adjusted. The centrifugal pump will **develop** different head values and will need brake **horsepower** different from the electrical motor. All these changes are described by the pump performance curves valid at variable frequencies usually available from manufacturers. In the absence of such curves, the **affinity laws** and the performance curves at a constant frequency may be used to calculate the required parameters at the corrected frequency: heads, efficiencies, and brake horsepower.

The performance parameters of the ESP motor at **variable** frequency operation are described by two basic formulae that express the change of (1) the nameplate **voltage** and (2) the **power** developed, both with changes in the electrical frequency. Actually, nameplate voltage of the motor is adjusted by the surface VSD unit so that the voltage to frequency ratio is kept **constant**. This is to ensure that the motor becomes a constant-torque variable-speed device. The applicable formula is

$$U_2 = U_1 \left(\frac{f_2}{f_1} \right) \quad (7.2)$$

where

f_1, f_2 = AC frequencies, Hz

U_1, U_2 = output voltages at f_1 and f_2 Hz, V.

The power developed by the ESP motor is linearly proportional to the electrical frequency as shown by the following formula:

$$HP_2 = HP_1 \left(\frac{f_2}{f_1} \right) \quad (7.3)$$

where

f_1, f_2 = AC frequencies, Hz

HP_1, HP_2 = motor powers at f_1 and f_2 Hz, HP.

The use of a VSD unit substantially modifies the **power** conditions of the ESP and the motor. To fully understand the changes and to demonstrate the beneficial effects of removing wellhead chokes, a detailed calculation model is presented in the next example. Energy flow in the ESP system is determined for current conditions when a surface choke is used to control the well's liquid flow rate. Then the use of a VSD unit is investigated and the required operational frequency is determined; the energy conditions of the modified installation are compared to the original case.

7.3.2.3 Sample Calculations

The well with the data provided in the following table is produced through a wellhead choke to restrict the flow rate. Investigate the present power conditions in the ESP system and propose the use of a VSD unit.

Well Data

Depth of Perforations	8,000 ft
Tubing Size	3 ¹ / ₂ "
Static BHP	2,500 psi
Productivity Index	12 STB/psi
Production GOR	82 scf/STB

Production Data

Liquid Rate	2,500 STB/d
Liquid SpGr	0.876
Producing WHP	650 psi
Flowline Pressure	100 psi

ESP Installation Data

Pump Setting Depth	6,000 ft
ESP Type	GN4000
Number of Stages	140
Electrical Frequency	50 Hz
Motor NP Power	104.2 HP
Motor NP Voltage	1,095 V
Motor NP Current	60 A
ESP Cable Size	AWG 2

BHP, bottomhole pressure; *ESP*, electrical submersible pump; *GOR*, gas–oil ratio; *NP*, nameplate; *WHP*, wellhead pressure.

First calculate those parameters that are identical for the original choked condition and for the case without the choke.

The flowing bottomhole pressure (*FBHP*) from the productivity index formula (Eq. 5.1) is found as

$$FBHP = 2,500 - 2,500/12 = 2,292 \text{ psi.}$$

Now the pump intake pressure (*PIP*) is calculated from Eq. (5.2)

$$PIP = 2,292 - 0.433 \cdot 0.876(8,000 - 6,000) = 1,533 \text{ psi.}$$

Knowledge of these parameters permits the calculation of the system's useful hydraulic power from Eq. (6.30):

$$P_{\text{hydr}} = 1.7 \cdot 10^{-5} \cdot 2,500(0.433 \cdot 0.876 \cdot 6,000 - 1,533) = 31.6 \text{ HP} = 23.6 \text{ kW.}$$

At pump suction conditions, there is no free gas present, as can be found from the Standing correlation; the oil's volume factor at the same pressure is found as $B_o = 1.1$ bbl/STB. The total liquid volume to be handled by the ESP is thus

$$q_l = 2,500 \cdot 1.1 = 2,750 \text{ bpd, or } 437 \text{ m}^3/\text{day.}$$

To find the frictional head loss due to the flow of the current liquid rate through the well tubing the **Hazen—Williams formula** or the use of the proper graph gives a head loss of 20 ft/1,000 ft of pipe. The total head loss in the tubing is thus

$$\Delta H_{fr} = 20 \cdot 6,000/1,000 = 121 \text{ ft.}$$

The energy loss corresponding to tubing frictional losses can be calculated from Eq. (6.33):

$$\Delta P_{fr} = 7.368 \cdot 10^{-6} \cdot 2,750 \cdot 121 \cdot 0.876 = 2 \text{ HP} = 1.5 \text{ kW.}$$

Energy Conditions of the Current Installation

This section contains calculations for the original choked condition and evaluates the energy conditions of the current ESP installation.

The hydraulic losses due to the backpressure at the wellhead pressure of 650 psi are calculated from Eq. (6.35) as follows:

$$\Delta P_{bp} = 1.7 \cdot 10^{-5} \cdot 2,750 \cdot 650 = 30.4 \text{ HP} = 22.7 \text{ kW.}$$

From the performance curve of the GN4000 pump operated at 50 Hz, the following parameters of the pump are found at the current liquid rate of 437 m³/day:

$$\text{BHP/Stage} = 0.85 \text{ HP.}$$

$$\text{Pump Efficiency} = 65\%.$$

As there are 140 stages in the pump, the pump's power requirement is

$$BHP = 140 \text{ } 0.85 \text{ SpGr} = 140 \text{ } 0.85 \text{ } 0.876 = 104 \text{ BHP.}$$

Based on these parameters the power lost in the pump can be calculated from Eq. (6.37):

$$\Delta P_{\text{pump}} = 104(1 - 0.65) = 36.4 \text{ HP} = 27.1 \text{ kW.}$$

The existing ESP motor's rated power (see input data) being 104.2 HP, the motor is 100% loaded by the pump power of 104 BHP. From motor performance data, motor efficiency at that loading is 89%. Now the power loss in the electrical motor can be found from Eq. (6.39):

$$\Delta P_{\text{motor}} = 104(1 - 0.89) = 11.4 \text{ HP} = 8.5 \text{ kW.}$$

To find the electrical losses in the downhole cable, first the resistance of the well cable has to be found. The installation uses an AWG 2 size electrical cable with an electrical resistance of 0.17 ohm/1,000 ft. The total resistance of the downhole cable, considering the well temperature, is found as 1.02 ohms.

The actual current flowing through the cable is equal to the motor current found from the motor load and the nameplate current as

$$I = 1.0 \text{ } 60 = 60 \text{ A.}$$

Now the electrical power lost in the cable is calculated from the basic three-phase power formula (Eq. 6.41):

$$\Delta P_{\text{cable}} = \frac{3 \text{ } 60^2 \text{ } 1.02}{1,000} = 11 \text{ kW.}$$

Finally the power lost in the ESP system's surface components is to be found from Eq. (6.43) with a surface efficiency of 97%.

$$\Delta P_{\text{surf}} = 94.4 \frac{(1 - 0.97)}{0.97} = 2.8 \text{ kW.}$$

Energy Conditions of the Modified Installation

This section presents the calculations required to describe the conditions when a VSD unit is used to control the pumping rate instead of choking the well. In this case the operating wellhead pressure is reduced to the flowline intake pressure; this was measured downstream of the wellhead choke as 100 psi. The hydraulic losses due to the backpressure are calculated from Eq. (6.35) as follows:

$$\Delta P_{bp} = 1.7 \text{ } E-5 \text{ } 2,750 \text{ } 100 = 4.7 \text{ HP} = 3.5 \text{ kW.}$$

Next the required electrical frequency is calculated using the head performance curve of the pump. From the 50 Hz head curve the head developed

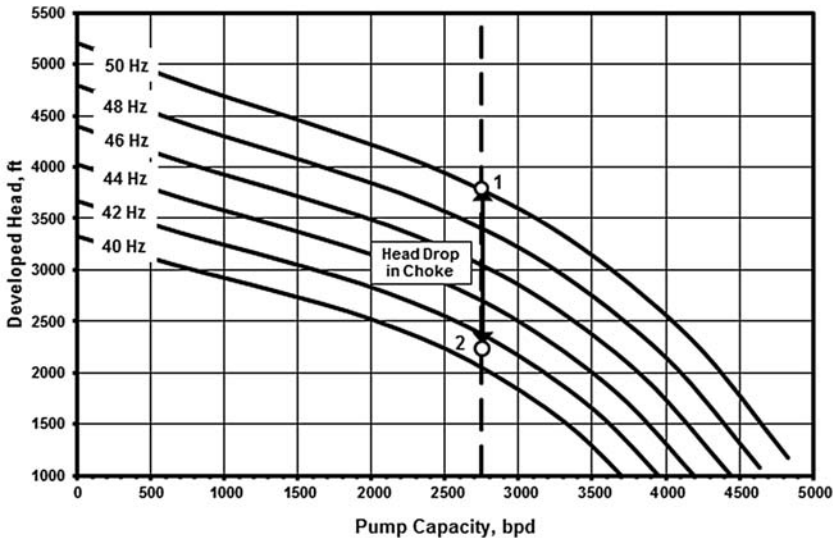


FIGURE 7.2 Finding the required frequency for the sample case.

by the pump at the desired rate of 2,750 bpd is read as 3,800 ft (**point 1**) from the pump performance curve presented in Fig. 7.2. The head drop across the choke is easily found by expressing the pressure drop across the choke in head units, as follows:

$$\text{Drop} = 2.31(WHP - p_{\text{downstream}})/\text{SpGr} = 2.31(650 - 100)/0.876 = 1,450 \text{ ft.}$$

From **point 1**, a vertical is dropped by the calculated distance of 1,450 ft; this defines **point 2**. The frequency valid at that point (41 Hz) is read; this should be used on the VSD unit to drive the ESP motor.

The power wasted across the choke according to Eq. (7.1) is

$$BHP_{\text{wasted}} = 7.368 E-6 \cdot 2,500 \cdot 1,450 \cdot 0.876 = 23.4 \text{ HP} = 17.5 \text{ kW.}$$

Next the operational parameters of the GN4000 pump at 41 Hz service must be determined. As detailed performance curves for this frequency are not available, the published 50-Hz curves and the **affinity laws** should be used.

The required rate of 437 m³/day at 41 Hz operation corresponds to the following rate at 50 Hz, as found from the **affinity laws**:

$$\text{Rate} = 437 \cdot 50/41 = 533 \text{ m}^3/\text{day.}$$

The power requirement and the efficiency of the pump at this rate at 50 Hz operation are read from the 50-Hz performance curves as

$$\text{BHP/stage} = 0.85 \text{ HP/stage}$$

$$\text{Pump Efficiency} = 67\%.$$

From these data and using the affinity laws again, the power requirement of the pump at 41 Hz operation is calculated:

$$\text{BHP/stage} = 0.85(41/50)^3 = 0.47 \text{ HP/stage.}$$

The efficiency of the pump remains at 65%.

Now the power needed to drive the 140 pump stages at an electrical frequency of 41 Hz has decreased to

$$\text{BHP} = 140 \text{ } 0.47 \text{ SpGr} = 140 \text{ } 0.47 \text{ } 0.876 = 58 \text{ HP.}$$

Based on these parameters the power lost in the pump can be calculated from Eq. (6.37):

$$\Delta P_{\text{pump}} = 58(1 - 0.67) = 19 \text{ HP} = 13 \text{ kW.}$$

The operating conditions of the ESP motor change at the modified frequency. Its nameplate power decreases linearly with the electrical frequency according to Eq. (7.3):

$$P_{np} = 104.2(41/50) = 85.4 \text{ HP.}$$

Motor voltage is adjusted by the VSD unit from the nameplate value valid at 50 Hz to the following voltage, according to Eq. (7.2):

$$U_{\text{motor}} = 1,095(41/50) = 898 \text{ V.}$$

The ESP motor's loading is found from the pump power requirement and the modified motor power:

$$\text{Loading} = 58/85.4 = 68\%.$$

The efficiency of the motor at this loading is 86%, as found from motor performance curves. The power loss in the electrical motor can now be found from Eq. (6.39):

$$\Delta P_{\text{motor}} = 58(1 - 0.86) = 8.1 \text{ HP} = 6 \text{ kW.}$$

When finding the electrical losses in the downhole cable, the resistance of the cable is identical to the previous case at 1.02 ohms. Because the nameplate current of the ESP motor at the new frequency does not change, the actual motor current is found from the motor load and nameplate current as

$$I = 0.68 \text{ } 60 = 41 \text{ A.}$$

The power loss in the cable is calculated from Eq. (6.41):

$$\Delta P_{\text{cable}} = \frac{3 \text{ } 41^2 \text{ } 1.02}{1,000} = 5.1 \text{ kW.}$$

Finally the power lost in the ESP system's surface components is found from Eq. (6.43) with a surface efficiency of 97%:

$$\Delta P_{\text{surf}} = 52.7 \frac{(1 - 0.97)}{0.97} = 1.6 \text{ kW}.$$

Conclusions

The power conditions of the two cases are presented in [Table 7.1](#); use of the VSD unit to decrease the motor's speed instead of using a surface production choke has substantially increased the system's power **efficiency**. Similar evaluations in a 10-well field [\[3\]](#) reduced total power requirements to almost one-third of the power used in choked wells. These results clearly prove that using VSDs to control the production rate of ESP wells is a much superior solution than wellhead choking frequently adopted in field practice.

TABLE 7.1 Comparison of Power Conditions of an Example Choked and VSD Installation

Power Component	Choked Case	VSD Case
Useful hydraulic power, kW	23.6	23.6
Backpressure losses, kW	22.7	3.5
Tubing friction, kW	1.5	1.5
ESP pump losses, kW	27.1	13
ESP motor losses, kW	8.5	6
ESP cable losses, kW	11	5.1
Surface losses, kW	2.8	1.6
Total Power, kW	97.2	54.3
System Efficiency, %	24	43

ESP, electrical submersible pump; *VSD*, variable-speed drive.

7.4 MONITORING ESP OPERATIONS

7.4.1 Introduction

Assuming that a correct design and trouble-free installation of ESP equipment has been made, the ESP system's performance and probable run life depend largely on the proper operation of the system. To ensure the most profitable operations, some key parameters must be continuously **monitored** and the

analysis of these parameters allows an assessment of the actual condition of the equipment as well as provides a way to anticipate possible problems.

The operating parameters to be monitored along with their possible diagnostic values are the following.

- **Electrical** properties such as motor current, phase voltages, phase imbalance, frequency, harmonics measured at surface.
 - Any changes in the pump, the well, or the electrical system are indicated in the current drawn by the ESP motor.
 - Pump speed and production are related to motor speed which is function of frequency.
 - Trends in the ESP system's loading are detected.
 - Motor damage caused by electrical or mechanical problems can be noticed.
 - Changes in produced fluid properties can be traced.
- Motor **temperature** measured using downhole gauge.
 - Rising motor temperature is indicative of low flow rate past the motor or deadhead meaning surface valve closed or obstruction in flowline.
- Motor and pump **vibrations** using downhole gauge.
 - Increased vibration is indicative of pump imbalance that could result from significant changes in phases flowing through pump, pump wear, blockage(s) inside pump, impending motor failure, broken shaft, startup, frequent startup—shutdown cycles, sudden motor frequency changes.
- **PIP** calculated from an acoustic survey or downhole gauge.
 - Changes in PIP indicate changes in pump performance, well inflow, or installation integrity.
 - Comparing measured PIPs with design values gives information on the accuracy of installation design.
 - Cyclic operation can be anticipated from PIP records.
- **Pump discharge pressure** (PDP) calculated from surface tubing pressure or measured using downhole gauge.
 - Changes in PDP indicate change in pump performance, water cut or gas—oil ratio (GOR) changes, or pump wear or deadhead condition.
- **Wellhead** tubing and casing pressure measured at the surface.
 - Tubing pressure allows determination of PDP.
 - Changes in casing pressure indicative of change in PIP.
 - Change in production, pump performance, or deadhead condition.
- Liquid production **rate** from surface measurements.
 - It allows the determination of the pump's operating point on the performance curve.
 - It is the first indicator of downhole problems such as equipment wear, leaks.
 - Changing well inflow can easily be detected from the trend of liquid rate variation.

TABLE 7.2 Evolution of ESP Monitoring Procedures

Monitoring Level	Measured Parameters	Benefits/Limitations
Electrical only	Motor current (continuously)	Lowest cost
		Limited value for diagnosis and control
Electrical plus acoustic	Motor current (continuously)	Low cost
	Dynamic liquid level (on demand)	PIP calculated, may be incorrect
Basic DHM	Motor current (continuously)	Accurate PIP values
	PIP (continuously)	Limited diagnostic data
	Temperature (continuously)	
Advanced DHM	Continuous measurements	Permits perfect diagnosis
	Motor current	Allows advanced system control
	PIP	Most expensive, but pays out quickly
	Pump discharge pressure	
	Pump intake temperature	
	Motor temperature	
	System vibrations	
	Downhole fluid rate	

DHM, downhole measurement; *ESP*, electrical submersible pump; *PIP*, pump intake pressure
 After Williams AJ, Cudmore J, Beattie S. ESP monitoring – where is your speedometer? Paper presented at the ESP workshop held in Houston, Texas; April 30–May 2, 2003.

ESP system monitoring has evolved over the years and the different levels of monitoring are given in [Table 7.2](#). As shown, the use of **ammeter** charts to measure and record the current taken by the downhole motor was the earliest and simplest solution to diagnose ESP systems. Even today, most **onshore** installations use only the ammeter and acoustic surveys because of the moderate workover costs. **Offshore** wells, however, rely on sophisticated downhole measurements (DHMs) to a greater and greater extent. Although DHM units may be quite expensive as compared to the rest of the system, typical payout times for moderate liquid rates are days only [4]. These units not only improve the accuracy of the most important production parameters but also allow the application of advanced system **controls** not possible with simple ammeter measurements. DHM units are becoming more and more widespread and an increasing number of new installations are equipped with these devices.

The following sections discuss the applications and advantages of different solutions for monitoring ESP installations.

7.4.2 Acoustic Surveys

The most common method of finding the annular liquid level in an ESP installation is the acoustic well survey, also called **well sounding**. This survey operates on the principles of the propagation and reflection of pressure waves in gases. At the surface an acoustic **impulse** is generated in the casing-tubing annulus that travels in the form of pressure **waves** down the length of the gas column. These pressure waves are echoed (**reflected**) from tubing collars, ESP equipment with diameters greater than that of the tubing, casing liners, well fluids, etc. because of the local changes in cross-sectional area available for pressure wave propagation. The reflected waves are picked up by a **microphone** placed at the surface and the electrical signals are recorded on paper or by electronic means. Proper evaluation of the reflected signals permits the determination of the depth to the annular liquid level.

The acoustic **well sounder** consists of two basic components: (1) the wellhead assembly and (2) the recording and processing unit. The wellhead assembly is easily connected to the casing head by means of a threaded nipple. It contains a mechanism that creates the sound wave and a microphone for picking up the signals. Conventional well sounders utilize blank **cartridges** with black powder fired either manually or by remote control. Modern well sounder units employ the so-called “**gas guns**,” which provide the required pressure impulse by suddenly discharging a small amount of high-pressure gas (CO_2 or N_2) into the annulus. The recording unit filters and amplifies the electrical signals created by the microphone. The processed signals are then recorded on a chart in the function of time. The depth of the liquid level in the annulus is found by the proper interpretation of the acoustic chart.

An example acoustic chart is shown in [Fig. 7.3](#), which contains analog recordings of reflected sound signals on a **strip chart**. As seen, every tubing collar is clearly identified by a local peak of the reflected signal, whereas the liquid level is marked by a much larger reflection. Determination of the liquid level depth is possible either by counting the number of **collar signals** and comparing these data to well records or by finding the reflection time from the liquid level [\[5,6\]](#). Time readings are facilitated using the 1-s timing signals also recorded on the chart.

The accuracy of acoustic liquid level surveys is highly dependent on the accurate knowledge of the **acoustic velocity** of the natural gas valid under the actual conditions. The use of some form of the **equation of state** for the gas can supply these values [\[7\]](#). In wells with uniform gas composition in the casing annulus, reliable acoustic velocities are calculated from this approach. Wells **venting** annulus gas to the flowline, such as ESP installations, contain **uniform** gas columns with a constant composition along their length. The casing

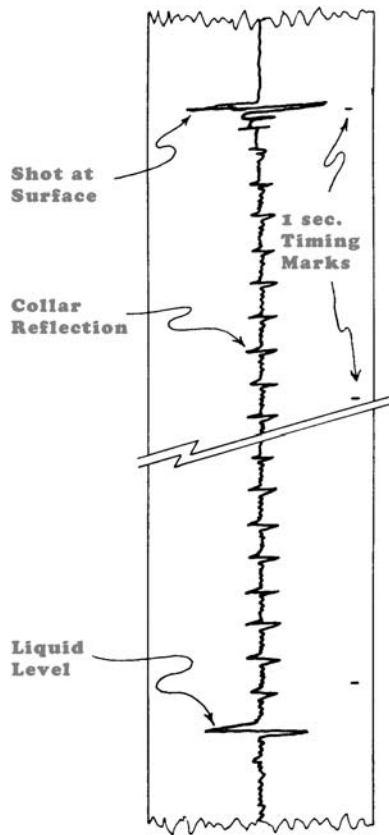


FIGURE 7.3 A typical acoustic chart.

annulus of wells not venting gas or shut in for longer periods contains a gas column whose composition varies with depth. In such cases, great discrepancies in local acoustic velocities may occur.

Acoustic surveys are widely used in the analysis of ESP installations. However, if the annular fluid has a tendency to **foam**, no firm signals from the liquid level can be attained and survey results are unreliable. Provided the liquid level measurements are accurate the PIP and FBHP can be calculated by using the calculation model described in Section 6.3.3.

7.4.3 Downhole Measurements

The application of measurement devices **permanently** installed in the well, a.k.a. **DHM**, along with the ESP unit facilitates monitoring and control of the ESP system at an advanced level. The different transducers send signals to the surface, which can be stored, evaluated, and used for controlling the system's

operation. The following sections describe the available instruments and their main features.

7.4.3.1 *Instruments and Communications*

The basic parameters monitored by downhole devices are pressure, temperature, flow, and vibrations. The devices available to measure these data along with their main features are detailed in the following according to **Berry [8]**.

- **Pressure Transducers.**
 - The Bourdon tube models contain potentiometers and are sensitive to vibrations and mechanical damage. They need proper calibration.
 - Capacitance gauges consist of two or more metal diaphragms that form a capacitor. High pressure ratings require small diaphragms, which are prone to permanent distortion.
 - Torque capacitance gauges combine the use of a Bourdon tube and capacitances and are exceptionally accurate.
 - Strain gauges measure the distortion of a metallic diaphragm and are reliable and rugged.
 - Resonant quartz gauges sense the change in resonant frequency of a quartz crystal.
- **Temperature Transducers.**
 - Thermocouples use dissimilar conductors joined together and generate a voltage differential proportional to temperature.
 - RTDs (resistive thermal devices) consist of a usually platinum wire whose electrical resistance changes with changes in temperature.
 - Thermistors are semiconductors with high electrical resistances that change according to temperature.
- **Flow Rate Transducers.**
 - Turbine meters contain a rotor turning with fluid flow whose speed is proportional to flow rate.
 - The Venturi meters do not have moving parts and contain a (usually wireline retrievable) Venturi tube and two pressure sensors.
- **Vibration Transducers.**
 - The accelerometers used for vibration detection usually measure movement in one or two directions.

One or several of the devices described are run in the well in an instrument **package** that can be situated at two possible **positions**:

1. If the package is at the **bottom** of the motor housing, then its power supply and data transmission via the motor cable are easily accomplished. The disadvantages of this position are (1) the heating effect of the motor on the produced liquid cannot be detected and (2) checking the status of stator windings is impossible.

2. Downhole measuring devices can be run in special “**gauge carriers**” at a short distance above the pump discharge. The need for a dedicated communications wire is the main disadvantage.

Downhole measuring devices can **communicate** to the surface by one of the following ways.

1. The ESP **power cable** may be used to transmit at regular intervals a DC signal from the surface that reads a variable resistance. This resistance is set by the downhole circuitry to indicate the reading from a transducer. Some units can sequentially send signals from several transducers but at longer time cycles. As seen, no transmission equipment is needed, but data transfer rates are limited.
2. With gauge carriers a **dedicated line** is used for power and communications. A small-diameter (maximum $\frac{3}{8}$ " diameter) stainless steel tube contains the centralized signal wire surrounded by some sealing material. This solution's biggest advantages are the increased communications bandwidth and the complete electronic isolation from the ESP unit. Limitations are related to the necessity of the separate line and include increased costs for the line and tubing clamps as well as increased operational costs.

7.4.3.2 Measured Parameters and Their Use

Modern DHM units use some of the transducers described previously to obtain reliable data on the most important operating parameters of the ESP system. The continuous measurement of these parameters enables the operator to control the ESP system's operation; the different solutions are described in the following.

PIP indicates the flowing pressure in the well at the level of the ESP suction. It is the PIP that determines whether **free gas** enters the pump or not. It is also strongly related to the well's FBHP and its inflow rate; greater PIPs meaning lower liquid rates. Because of all these factors, PIP can provide a reliable way to control the operation of the ESP system. Surface control can be set to shut down the unit when PIP falls below a minimum value to prevent **free-gas** production or to prevent **pump-off** conditions.

PDP changes in response to changes in wellhead pressure, in water cut, or in the gas content of the well fluid in the tubing string. When used as a control parameter, its value must not be allowed to rise above a maximum.

Pump differential pressure represents the increase of flowing pressure, Δp_{pump} , exerted by the ESP. For a known driving frequency and fluid density, this is determined by the pump's **Q–H** (head vs. pumping rate) performance curve. If the heads belonging to the two limits of the recommended operational range are converted to pressures then those two pressures can be used to check whether the unit is operating properly. If the actually measured Δp_{pump} is less

than the calculated minimum, the pump should be shut down due to dangerous **upthrust** conditions, while a measured differential pressure greater than the calculated maximum means dangerous **downthrust** operations (For an explanation consult Fig. 3.4.).

Motor temperature may be the measured temperature of the motor windings or that of the motor oil. Motor **winding** temperature is the preferred one because it more rapidly responds to motor problems; the use of embedded thermocouples is the best solution. If used in system control, measured temperature must not be allowed to rise significantly above the motor's rated temperature.

Pump intake temperature provides a check on measured motor temperature and any changes may indicate a changing inflow rate to the well. In system control, it is used similarly to motor temperature.

System vibrations, because of their different possible causes, cannot be directly interpreted; their **trend**, however, indicates the onset of possible problems that may later lead to more severe mechanical or electrical problems. Usually, a limit to the measured system vibrations is set above which an alarm is activated. Normal operations result in accelerations of less than 3 g ($g = 32.2 \text{ ft/s}^2$ being the acceleration of gravity), whereas severe vibrations usually give a reading above 10 g.

7.4.4 Surface Controllers

7.4.4.1 Introduction

Surface ESP controllers play a very important role in that they isolate downhole system from the power supply system. They provide capabilities to control and modify ESP operations. Most of the surface electrical property measurements and controls are performed here. These controllers also interface with and increasingly integrate functionality of a wellhead remote terminal unit (RTU).

7.4.4.1.1 Fixed-Speed Drives

Fixed-speed drives are typically used in onshore, simple, cost-effective situations where reservoir conditions are benign or the cost of surface controller is a consideration. As the name implies, they do not allow speed variations, so from ESP optimization perspective, they have limited utility.

7.4.4.1.2 Variable-Speed Drives

Operators have increasingly begun to realize the importance of operational flexibility and savings afforded by the VSD units. As described in Section 4.7, when an ESP system is operated at varying speed, the same pump can cover fairly wide flow rate range.

7.4.4.2 ESP Controller Functions

An ESP controller is a versatile wellsite optimization tool and performs some or all the following functions depending on the type and built-in capabilities:

- collect and store the following **electrical** parameters
 - current: three phases
 - current imbalance
 - voltage: three phases
 - voltage imbalance
 - power
- provide ESP **monitoring** and shutdown capabilities
- allow frequency modification in VSDs
- provide a human—machine interface, typically a touch-sensitive surface display that supports
 - displaying run status
 - limited visualization of data trends, alarms
 - data entry for trip and alarm settings
 - resetting alarms or trips
 - adjusting control modes and parameters, e.g., switching from variable to fixed frequency or fixed-current mode
- support control algorithms that detect or infer well conditions from downhole and surface measurements and take autonomous action, for example,
 - change the speed in case of gas lock or cavitation or high motor temperature condition
 - adjust the startup speed in case of hard to start or stuck pump conditions
 - alternate motor speed to “shake pump” to prevent or free scale deposits inside pump
- some of them have programming capabilities to provide “programmable logical controller (PLC)” capabilities that allow custom algorithm implementation.
- integrate power conditioning to avoid harmonics problem in variable-speed situation.

7.5 SURVEILLANCE OF ESP OPERATIONS¹

7.5.1 Introduction

Assuming that a correct design and trouble-free installation of ESP equipment has been made, the ESP system’s performance and probable run life depend largely on the proper operation of the system. To ensure the highest availability

1. This section was written by Dr. R. Chokshi from Accutant Solutions.

of the ESP system and thereby the most profitable operations, a list of indicative **goals** for an ESP **surveillance** program could be developed:

1. match pumping rate to inflow,
2. operate pump within manufacturer specified pumping limits,
3. prevent pump upthrust and downthrust conditions that lead to lower efficiencies and higher failures,
4. never run the pump dry,
5. keep production rates high enough to avoid overheating the motor And production flow must cool the ESP motor,
6. keep production rates low enough to avoid excess sand, pump-off, and “unsafe” operation,
7. recognize abnormal situations such as gas locking or cavitation,
8. identify failure conditions such as current fluctuations could be shaft failure or gas ingestion or hole in tubing.

To meet the abovementioned surveillance objectives, some key parameters must be continuously **monitored** and the analysis of these parameters allows an assessment of the actual condition of the equipment as well as provides a way to anticipate possible problems. The operating parameters to be monitored along with their possible diagnostic values are the following.

- **Electrical** properties such as motor current, phase voltages, phase imbalance, frequency, and harmonics measured at surface.
 - Any changes in the pump, the well, or the electrical system are indicated in the current drawn by the ESP motor.
 - Pump speed and production are related to motor speed, which is a function of frequency.
 - Trends in the ESP system’s loading are detected.
 - Motor damage caused by electrical or mechanical problems can be noticed.
 - Changes in produced fluid properties can be traced.
- Motor **temperature** measured using downhole gauge.
 - Rising motor temperature is indicative of low flow rate past the motor or deadhead, meaning surface valve closed or obstruction in flowline.
- Motor and pump **vibrations** using downhole gauge.
 - Increased vibration is indicative of pump imbalance that could result from significant changes in phases flowing through pump, pump wear, blockage(s) inside pump, impending motor failure, broken shaft, startup, frequent startup—shutdown cycles, and sudden motor frequency changes.
- **PIP** calculated from an acoustic survey or downhole gauge.
 - Changes in PIP indicate changes in pump performance, well inflow, or installation integrity.

- Comparing measured PIPs with design values gives information on the accuracy of installation design.
- Cyclic operation can be anticipated from PIP records.
- **PDP** calculated from surface tubing pressure or measured using downhole gauge.
 - Changes in PDP indicate change in pump performance, water cut or GOR changes, or pump wear or deadhead condition.
- Wellhead tubing and casing **pressure** measured at the surface.
 - Tubing pressure allows determination of PDP.
 - Changes in casing pressure are indicative of change in PIP.
 - Change in production, pump performance, or deadhead condition.
- Liquid production **rate** from surface measurements.
 - It allows the determination of the pump's operating point on the performance curve.
 - It is the first indicator of downhole problems such as equipment wear and leaks.
 - Changing well inflow can easily be detected from the trend of liquid rate variation.

7.5.2 Key Challenges and Requirements in Production Operations

Quite commonly the methods, devices, and best practices of surveillance are covered under the header of **digital oilfield** (DOF). To understand what DOF is and why it is needed, first we need to look at the broader issue of challenges and requirements to solve the challenges in production operations.

In highly competitive and low-price environments, operators are constantly challenged to improve productivity and keep costs low. Specifically, the following demands are placed:

1. improve operations efficiency
 - a. minimize nonproductive down time
 - b. minimize operating costs
 - c. minimize the time to collect, filter, and deliver data to multitude of stakeholders
2. maintain well and infrastructure integrity
 - a. improve visibility of equipment availability
 - b. safeguard health, safety, and environment
3. do the above continuously and in real-time with shrinking resources including staff.

These challenging objectives can be addressed with

1. exception-based management
 - a. rank activities and focus on the most value-added tasks
 - b. manage large number of assets

2. early problem diagnosis
 - a. detect impending performance degradation or failure(s)
 - b. implement lessons learned from root cause analysis
3. faster and well-informed decision making
 - a. complex problem solving
 - b. understand and evaluate consequences

DOF is an enabler in meeting these challenges by providing toolsets that aim to satisfy these requirements. First let us address what DOF is.

7.5.3 The Digital Oilfield

DOF concepts have been revised over the years and various implementations are tried. Most of the progress has relied on advancements in microprocessors, communications, and computer-based technologies.

DOF is evolution and convergence of digital control techniques coupled with standardized communications that are overlaid with analytics. Many acronyms are used to describe DOF, and one way to put it is DOF is a melding of operations technology (OT) with information technology (IT) and the Internet of things. It is important to understand that a successful DOF implementation requires careful orchestration and convergence of technology/tools, processes, and properly trained people who apply technology consistently with processes.

Underlying expectations for a DOF implementation are

1. remote data capture from field site and wellsite reliably
2. convert captured data into actionable information to
 - a. diagnose problems remotely without sending personnel to well/field
 - b. prevent problems and failures from occurring
 - c. predict performance
 - d. optimize
3. make the above a sustainable and repeatable process.

According to the BP outlook report to 2050 [9], “in the environment of lower hydrocarbon prices with reduced revenues and capital spending, the digital technologies — including sensors, data analytics and automated systems — stand out as the leading contributors for reducing costs. ...These technologies play important role in enabling and integrating with other technologies while enhancing safety and reliability. The digital technologies have potential to increase oil and gas production by 4% while reducing costs by 13%.”

7.5.3.1 An Overview

A high-level overview of a DOF installation is shown in [Fig. 7.4](#). The following description reviews each numbered segment in the figure as data flows from the section numbered 1 on the right to the end users on the left of the figure.

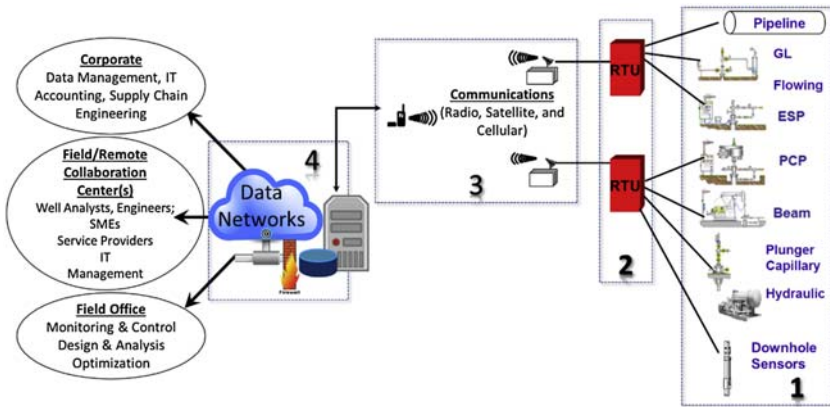


FIGURE 7.4 Digital oilfield overview [10]. ESP, electrical submersible pump; GL, gas lift; IT, information technology; PCP, progressing cavity pumping; RTU, remote terminal unit; SME, subject matter expert. Courtesy: Weatherford. Artificial lift training: production optimization. Class Notes 2015.

- Wellsite Optimization—Sensors:** In the right-most section numbered 1, well and field installations are shown equipped to source data both from surface and downhole locations. Data is captured by a wide variety of sensing devices that measure pressure, temperature, injection and/or production flow rates, fluid levels, vibrations, loads, torques, speed, voltages, currents, differentials, movements, device openings, leakages, pollution levels, etc.
- Wellsite Optimization—Controllers:** The data stream is collected through local RTUs or PLCs or programmable integration differentiation devices at a wellsite or pad or installation level. Such devices usually have some autonomous decision-making capabilities in the sense that they react and modify or even stop the operation if one of the measured or inferred parameters is out of the predefined range. Sensors discussed previously and controllers discussed in this section are collectively referred to as wellsite optimization tools that provide measurement and control capabilities at an individual wellsite.
- Communication Infrastructure:** RTUs or similar devices transmit the data to a localized or cloud-based data storage network via some mode of communication such as radio, satellite, cellular modems, or increasingly dedicated fiber-optic links. To conserve communication bandwidth, RTUs often capture more data and store them locally rather than communicating the entire stream. RTUs transmit data at preset times and on user demand. Security is implemented at each stage through firewalls.

- **Desktop Optimization—Software Applications:** Field data are consumed by various software applications that are deployed on computer server(s) at a field office and at a centralized office location. These are collectively referred to as application servers in the following.
 - a. Application servers are loaded with some form of surveillance software to collect and analyze the data. The application servers interface with a number of field applications that allow for monitoring, control, analysis, production optimization, and design. Field application operators monitor operations from remote locations and at times remotely start or shut down the devices in the field.
 - b. The application servers, through dedicated web- and mobile applications, provide support to artificial lift applications, management, production, and reservoir engineers located away from field. Data analytics-based software applications are being introduced gradually. Industry trends are to eventually develop and deploy deep machine learning algorithms that combine pure data-driven approaches with physics-based traditional models into expert systems. Application servers are typically referred to as desktop optimization tools.
 - c. Enterprise resource planning, supply chain management, accounting, compliance, human resources, and vendor management are some of the corporate groups accessing aggregated and rolled up data.
 - d. Service companies and other partners may be provided access to data through secure links from company networks.

A fully formed DOF requires full support from and collaboration among IT and OT groups.

7.5.3.2 Digital Oilfield: Step-by-Step Progression

Various activities that occur under the DOF umbrella can be broadly classified into four stages listed in progression order: surveillance, analysis, optimization, and innovation, as shown in Fig. 7.5. Resource requirement, sophistication, and the resulting gains increase noticeably, so an organization's maturity level in terms of DOF may be assessed by reviewing its infrastructure and workflows against each of these stages.

1. **Surveillance:** During this stage, sensors are deployed at the wellsite, downhole, and at surface facilities. supervisory control and data acquisition (SCADA) systems provide connectivity from wellsite to field office and remote locations. Focus is on alarming, root cause analysis and reporting type of monitoring activities that are equipment and outflow centric. Typically, activities covered here are tactical in nature, so field personnel are heavily involved and invested.
2. **Analysis:** Once surveillance phase is mastered, activities shift to understanding events in real time or near real time. Engineering diagnostic and

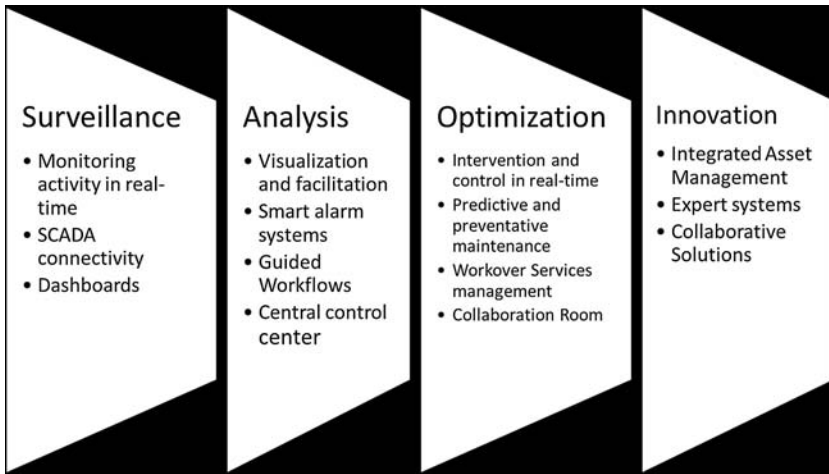


FIGURE 7.5 Progression in digital oilfield implementation [11]. SCADA, supervisory control and data acquisition.

analytical models are interfaced with real-time data to gain understanding of events as they occur rather than performing a postevent analysis. Well and network performance is coupled with reservoir deliverability and phase changes, so this stage has been called “Well Inflow Centric.” Prominent features of this stage are visualization and facilitation, smart alarm systems, guided workflows, and centralized analysis and control centers. This stage involves office-based experts evaluating activities and outcomes in support of the field personnel.

- 3. Optimization:** After the surveillance and analysis phases the optimization phase is about intervention and control in real time. Scope expands to cover wells, network, and the entire asset. Adjustment of production rates from different reservoir segments and wells is considered. Long-term and higher-value activities, such as workover and facility maintenance, are tweaked and scheduled. Predictive and preventative maintenance approaches are deployed in favor of corrective or postfailure maintenance. Expanding focus necessitates forming teams of diverse expertise to work in collaboration rooms—physical or virtual.
- 4. Transformation to Innovation:** This phase is increasingly clubbed with the optimization phase. Herein integrated asset management approaches are applied using next-generation data analysis tools. Expert systems are developed and deployed that look at potential opportunities from many different angles using terabytes of historical data. When historical data is lacking or unavailable, analogs from other domains are applied and machine intelligence is trained gradually.

7.5.4 ESP Data Visualization

Once data is captured from downhole and surface sensors, it is available on the user desktop through interface of a monitoring software. Various views and visualizations are available that are often grouped under workflow headings such as monitoring, surveillance, and analysis.

7.5.4.1 Well Status and Well Group Status

Using a “Well Group Status” view, field personnel are able to review status of all wells in a field or a subset of wells in a tabular grid format where each row represents a dataset for a well. [Fig. 7.6](#) shows an example dashboard for four wells with details captured during the last scan and parameters averaged over the previous 24-h, such as current, voltage, frequency, motor temperature, and others.

Most software packages incorporate **calculated** parameters on the status dashboard for a well. Examples of calculated parameters are

- pump differential pressure,
- PDP from wellhead pressure and the tubing gradient for the last test flow rates,
- PIP from known inflow performance relationship (IPR) and outflow performance,
- motor loading,
- motor amps, horsepower,
- calculated production rates from intake pressure and known IPR.

Often alarms based on calculated parameters are programmed into the software. Such alarms are often referred to as “**intelligent**” or “**inferred**” alarms, whereas the alarms based directly on measured parameters are called “analog” alarms. Some examples of inferred or intelligent alarms are

- gas lock condition detected
- pump intake plugged
- perforation plugged.

Example screenshots of dashboards displaying measured and calculated parameters are given in [Figs. 7.7 and 7.8](#). More examples of alarms and related rules are discussed in [\[13–16\]](#).

Status grid shows trip and alarm status for a particular well. Trips are indicative of ESP-stopping actions taken by the wellsite controller whenever a particular operating parameter (such as voltage, current, pressure, liquid level, flow rate) becomes **dangerously** higher or lower than its safe operating range. Unlike trip settings, alarms are raised in the surveillance software whenever a value of any measured or calculated well parameter falls **outside** one of the four bounds: High-High, Low-High, High-Low, and Low-Low. [Fig. 7.9](#)

ESP Well Group Status (2/4) / ESP_2

Well Alarm History | Note Log | Analog Status | Edit Downtime | RTU Read/Write

Well Name	Out of Scan	Scan Date	Scan Time	Last Good Scan Date	Last Good Scan	Alarm Message	HOA State	Well Status	Today Run Time (hrs)	Yesterday Run Time (hrs)	Last Run Time (hrs)	Starts Today	Total Starts	Avg Amps	Average Volts	Pump Intake Pressure	Vibration	Motor Temperature	Operating Frequency	Well
									hr	hr						PSI	G	DegF	Hz	
ESP_1	0	10/15/2013	15:21	10/15/2013	15:21		?	Run	1277.93	24.00	0.00	0	0	28.00	0.00	967.30	0.06	259.50	66.44	
ESP_2	0	02/18/2013	06:00	02/18/2013	06:00	Scanning RTU	A	Run	6.51	24.00	0.00	0	0	120.00	225.00	1100.00	4.00	137.00	64.60	
ESP_3	0	02/24/2013	01:30	02/24/2013	01:30	Current AMP HI	A	Run	1.99	24.00	0.00	0	0	120.00	222.00	1700.00	5.00	122.00	67.10	
ESP_4	0	02/23/2013	21:00	02/23/2013	21:00		A	Run	21.48	24.00	0.00	0	0	120.00	210.00	1500.00	5.00	136.00	60.90	

FIGURE 7.6 Well group status showing measurements and runtime status [12]. *Courtesy: Weatherford. LOWIS ESP optimization software screenshots. Private Correspondence June 2017.*

ESP System Performance (1/5) / ESP_1 Filter: Rate;												
*Well Name	Test Date	Observed Rate	Daily Power / Volume of Fluid	Head Eff.	Rate Eff.	Lift Perf Indicator	Pump System Eff.	Lift Eff	Lift Cost / Volume of Oil	Lift Cost / Volume of Fluid	Analysis Date	Analysis Time
		stb/d	kWh/bbl	%	%		%	%	\$/bbl	\$/bbl		
ESP_1	09/12/2016	1795.00	1.97	87.22	64.67	56.41	33.65	12.49	2.19	0.20	05/16/2017	11:44
ESP_2	08/03/2016	3000.00	1.20	93.21	82.94	77.31	57.75	28.09	0.36	0.12	05/09/2017	07:28
ESP_3	12/23/2015	2209.20	1.67	92.32	93.89	86.68	31.10	22.56	0.76	0.17	05/19/2017	14:11
ESP_4	05/06/2016	787.00	2.00	84.76	91.96	77.95	25.77	15.53	1.82	0.20	04/24/2017	13:13
ESP_Test	06/02/2016	700.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	04/24/2017	13:13

FIGURE 7.7 Well status grid showing performance analysis [12]. Courtesy: Weatherford. LOWIS ESP optimization software screenshots. Private Correspondence June 2017.

ESP Downhole Performance (1/10)															
*Well Name	Test Date	Observed Rate	Motor Hz	Water Cut	Ideal Rate	Rate % Difference	In-Situ Rate	Observed TDH	Theoretical TDH	TDH % Diff	Observed PDP	Theoretical PDP	PP % Diff	Observed PDP	Theoretical PDP
		stb/d	Hz	%	stb/d	%	stb/d	ft	ft	%	psia	psia	%	psia	psia
ESP_1	09/12/2016	1795.00	45.00	91.00	2775.03	-35.33	1548.00	3408.84	3446.89	-1.10	1195.36	1177.75	1.49	2755.00	2771.01
ESP_2	08/03/2016	3000.00	60.00	66.67	3617.03	-17.06	3096.00	3737.60	5536.45	-32.46	1323.30	537.37	146.26	3021.00	2963.78
ESP_3	12/23/2015	2209.20	45.00	78.00	3292.97	-8.11	2292.00	2705.13	3699.68	0.96	1768.81	1906.96	-1.00	3014.00	3013.66
ESP_4	05/06/2016	787.00	44.00	89.00	855.80	-8.04	828.00	3036.73	2630.86	0.23	1585.36	1597.92	-0.16	0.00	2799.51

FIGURE 7.8 Well status grid showing downhole performance [12]. Courtesy: Weatherford. LOWIS ESP optimization software screenshots. Private Correspondence June 2017.

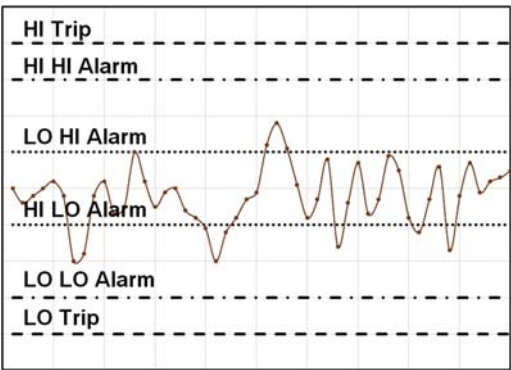


FIGURE 7.9 Typical trips and alarms for measurements.

shows how these bounds are set relative to the “normal” value for a measurement parameter.

Monitoring software color codes the well status grid per the “Management by Exception” principle. Well rows, having a trip status or alarm status values of HI-HI or LO-LO, are shown with red background. When one or more parameters exceed LO-HI or HI-LO values, the corresponding well’s row is shown in

yellow background and wells having all parameters in normal range are shown in green color. This feature helps field personnel **prioritize** activities such that wells in red are addressed first followed by the wells in yellow. Although not shown in color, Fig. 7.10 shows an example dashboard with a variety of alarms. The bottom panel shows a well group status where alarms for multiple wells are shown. The top-left panel shows the status of analog and calculated parameters based on measurements. The top-right panel shows “intelligent” alarms such as detection of gas lock or reduced inflow or pump intake plugging.

Surveillance software increasingly incorporate geographic information systems dashboard wherein wells are shown on physical **maps** and the user can zoom in to the particular well. Icons are superimposed for a well location to display alert status or important key performance indicator for a well. When a well icon is clicked, well details are visualized in a dashboard containing graphical and data grid format. Fig. 7.11 shows an example dashboard for a sample well. Depending on the operator setup and permissions granted, field personnel may be able to scan the well in real time and remotely change control parameters inside the wellsite controller.

7.5.4.2 Analysis Using Data Trend Visualization

Data trend **visualization** is a very important tool for the analysis of ESP installations. Visualizing multiple streams of **real-time** or **historical** data on a single screen allows a well analyst to understand the well’s changing behavior with time as well as to uncover hidden opportunities for optimization or figure out an impending problem. Analysis can be done through evaluating analog or measured data trends. Trend data such as PIP, PDP, amps, and motor

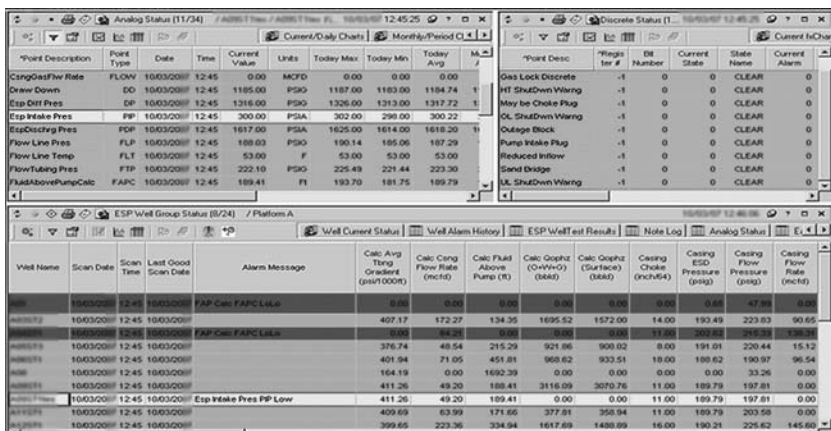


FIGURE 7.10 Dashboard with analog and intelligent alarms [12]. Courtesy: Weatherford. LOWIS ESP optimization software screenshots. Private Correspondence June 2017.

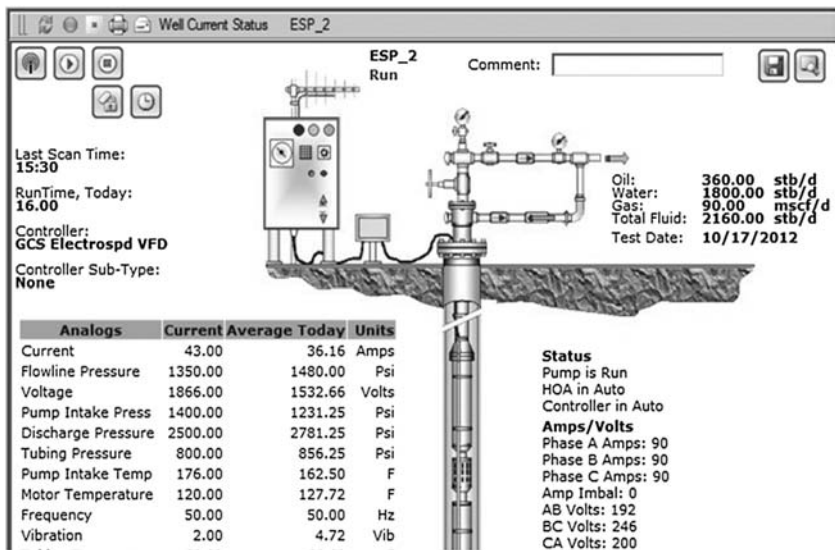


FIGURE 7.11 Well current status and control screen [12]. Courtesy: Weatherford. LOWIS ESP optimization software screenshots. Private Correspondence June 2017.

temperature help identify **abnormal** operating conditions and possible pump and well performance issues. Analysis can also be done using **inferred** variables such as static or FBHP, liquid level above pump, and pressure differential across pump.

Fig. 7.12 shows an example plot of trend analysis with various pump measurements and inferred production rate.

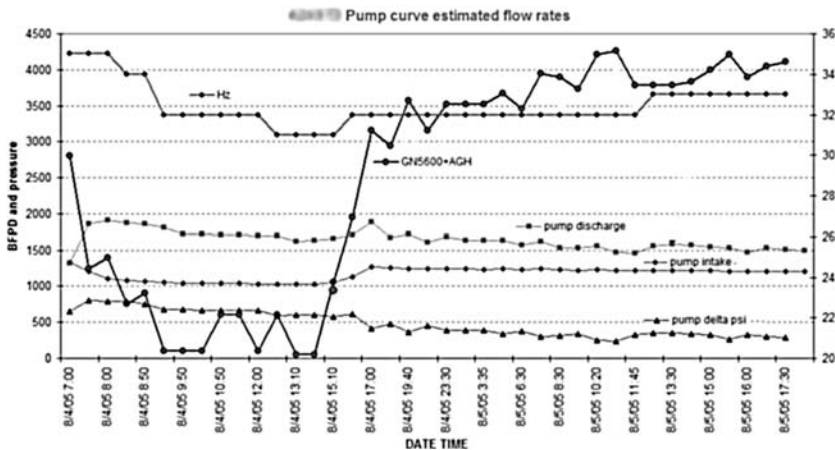


FIGURE 7.12 Trend analysis showing pump operating parameters [11].

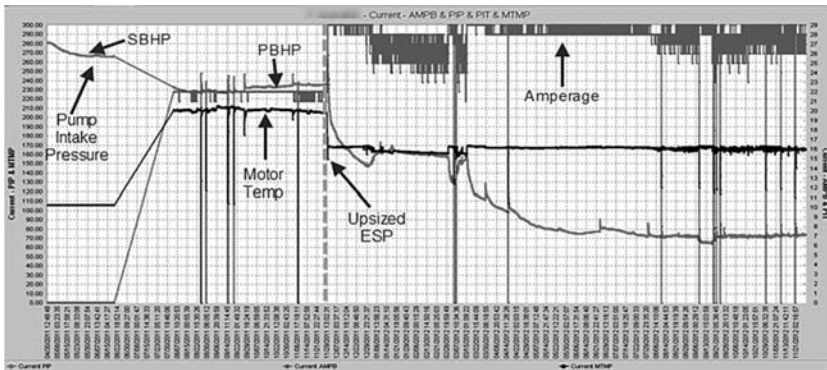


FIGURE 7.13 Trend analysis: effect of upsized ESP [12]. PBHP, producing bottomhole pressure; SBHP, static bottomhole pressure. Courtesy: Weatherford. LOWIS ESP optimization software screenshots. Private Correspondence June 2017.

Fig. 7.13 shows how PIP and motor temperature trends on the left of the dashed line were used to identify pump upsizing opportunity. Using shut-in and flowing PIP values as a proxy for reservoir pressure, **insufficient** drawdown was deduced by the operator. Another indication was higher motor **temperature** resulting from low flow rate. The operator decided to upsize the pump based on these inferences. The right side of the dashed line shows the impact of installing a **larger** pump where a drop in the PIP and the resulting increased drawdown signify higher flow rates. Further drop in motor temperature after installing a larger pump is also observed.

Fig. 7.14 shows another example of **trend** analysis with PIP and motor temperature measurements. Initially PIP is seen to be rising while motor

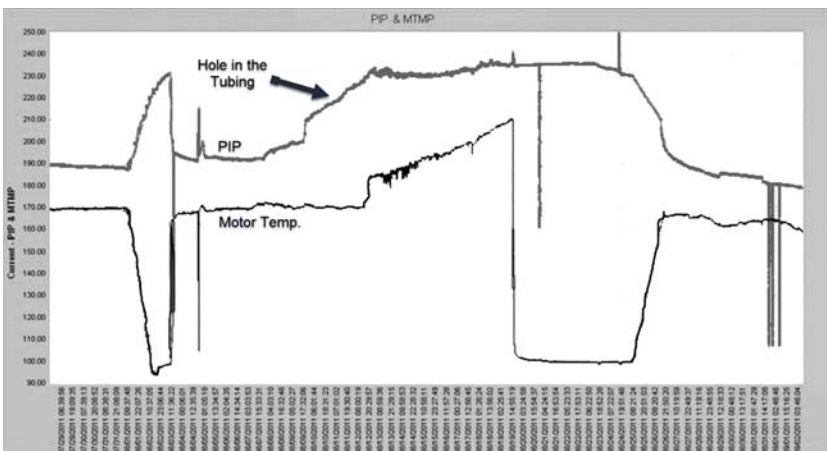


FIGURE 7.14 Trend analysis: hole in the tubing identified [12]. PIP, pump intake pressure. Courtesy: Weatherford. LOWIS ESP optimization software screenshots. Private Correspondence June 2017.

temperature is relatively steady. As PIP begins to stabilize, motor temperature is rising, leading to system **trip** once motor temperature exceeds threshold. Conclusion of the analysis is there is a **hole** in the tubing, and once this fault is corrected and the well is back on production, the motor temperature rises from no-flow geothermal values to flowing temperature while PIP drops closer to the value before the hole in the tubing event.

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Chapter 8

Failures and Troubleshooting

8.1 INTRODUCTION

Although proper monitoring can postpone system **failures**, failures eventually happen; these can be related to many possible general causes described in this chapter. According to their great importance, system **vibrations** are the subject of a separate section. Possible failures are classified according to the place of their occurrence, and typical failures in pumps, motors, etc. are detailed. **Teardown** analysis of failed electrical submersible pumping (ESP) components that helps diagnose the cause of failures experienced is described.

Troubleshooting of ESP installations is covered in the last section where the time-honored method of **ammeter** chart interpretation is described first. Modern methods utilizing **Nodal analysis** principles round up the treatment of troubleshooting practices. Understanding of the principles and practices described in this chapter helps operators achieve a sufficiently long run life of ESP equipment.

8.2 SYSTEM FAILURES

Eventual failures of the ESP system are mostly **electrical** in nature because the electrical system is often the weakest link. These failures, however, often originate from some other, mostly mechanical problem, which is the real **source** of the failure. This is the reason why every failure must be accurately analyzed and its proper cause must be found. Failure analysis, therefore, is a valuable tool in increasing the run life of ESP equipment.

In the following, first, the general causes of equipment failures and then typical failures in system components will be detailed.

8.2.1 General Causes of Failures

Equipment failures may be caused by several factors related to well conditions, design and installation problems, etc. These are classified and described in detail in this section. System **vibrations**, an inherent feature of

high-speed machinery, deserve a detailed treatment because of their detrimental effects.

Improper Installation Design

Proper initial sizing of ESP equipment is the **prime** requisite of a sufficiently long run life. The unit must be operating within the pump's **recommended** liquid flow rate range; pump wear is accelerated leading to premature pump and motor failures if the unit is run outside of this range. Proper design requires reliable data on the well's **productivity** and accurate information on **fluid properties**. It must be noted, however, that design inaccuracies may be corrected with the use of a **VSD** (Variable Speed Drive) unit.

Faulty Equipment and/or Installation

Running of equipment that is not properly checked for performance before installation can mean premature failure due to manufacturing **defects**. Therefore, ESP components must be **tested** for proper operation before running into the well. Moreover, proper **installation** of all components (motor, pump, cable, etc.) is important and following proper procedures is a crucial requirement.

Well Conditions

There are several well conditions that may have a prime effect on the longevity of ESP equipment, as listed in the following:

- **Corrosion** can affect all parts of the system, but corrosion of motor and/or pump housings is the most detrimental for the system. Proper choice of housing **material** or the application of a proper **coating** on outside surfaces can postpone or completely prevent this kind of failure.
- **Scale** formation on metal surfaces causes two kinds of problems:
 1. Scale deposits on motor housings reduce the intensity of heat transfer to the well fluid and cause increased motor winding **temperatures**, causing the motor to eventually burn out. Special **coatings** applied to the outside surface of the motor are the usual solution.
 2. Scale formation in pump stages reduces the liquid rate and can also totally **plug** the electrical submersible pump.
- **Sand** or abrasive production does the most harm to the electrical submersible pump, but rotary gas separators (RTGs) are also affected. Sand or abrasive particles carried by the liquid pumped cause **abrasion** and **erosion** in those parts where high liquid velocities are present. These are not only the pump stages but also radial and axial bearings; damage of these parts eventually leads to total system failure. The proper choice of pump **type** and **materials** can reduce or eliminate the negative effects of abrasive production.
- **High well temperatures** mostly affect the ESP motor, which, under unfavorable conditions, can burn out because of its inside temperature exceeding the rated temperature.

- **Formation gas** causes problems in the pump and the downhole cable. Pump operation may become unstable and the pump may be **gas locked**. Application of RTGs or other means of removing free gas from the well-stream before it enters the pump are the general solutions. In the case of the electric cable, free gas can **migrate** into insulation materials; proper choice of cable type may solve this problem.
- **High-viscosity** well fluids, as a rule, have higher specific gravities, and a greater pump brake horsepower is needed to lift the well. In addition, the efficiency of the centrifugal pump decreases while the frictional pressure loss in the tubing increases; both effects result in higher horsepower load on the motor.

Electrical Problems

Although in the majority of cases the **root** cause of an eventual electrical failure is unrelated to the electrical system, purely electrical problems may also lead to failure. Because the electrical system is series-connected, the weakest component may be the critical factor.

- **Power supply** problems include unbalanced phases, voltage spikes, the presence of harmonics, lightning strikes, etc., and their main effects are the **overheating** of the ESP motor and the cable. Most of these problems can be eliminated by using a “clean” electric supply or a VSD unit with **sinusoidal** output.
- **Motor controllers** usually operate flawlessly but (high/low temperatures, high level of moisture, etc.) may fail in extreme conditions.

8.2.1.1 Vibrations in the ESP System

Because almost all components of the ESP system contain parts rotating at a high speed, **vibrations** in these elements are unavoidable. Vibrations originating from any of the ESP components are promptly transmitted across the system via the shafts connected to each other; failure of any component may follow. Root causes of system vibrations can be grouped, according to **API RP 11S8 [1]**, as given in the following:

- Vibrations caused by mass **imbalance** of the rotating parts may be due to:
 - a nonhomogeneous distribution of material density of some parts,
 - nonsymmetrical configurations of rotating parts, and
 - **eccentricity** in shafts and/or bearings.
- Hydraulically induced vibrations take place in the electrical submersible pump:
 - when it is operated outside of its **recommended** liquid rate range and
 - when pumping of **gassy** or highly **viscous** fluids.

- In lightly loaded journal bearings, **oil whirl** can cause vibrations.
- In improperly secured journal bearings, the bearing may **rotate** with the shaft.
- Mechanical **rubbing** between stationary and rotating parts also induces vibrations.
- Electrical effects such as power instability and changing of the driving frequency can also result in vibrations.

All the effects described here can be identified by observing the **frequency** of the vibrations that occur because those are characteristics of the problem. [Table 8.1](#) contains typical frequencies in the function of the ESP system’s rotational speed.

Vibrations in ESP systems are one of or a combination of the following kinds:

- **Axial** vibrations damage the unit’s thrust bearings in the pump, motor, and protector.
- **Radial** (lateral) vibrations are especially dangerous because the long **slender** shafts of the unit are very sensitive to radial stability. If stability is lost because of any reason, conditions deteriorate very quickly leading to

TABLE 8.1 Typical Vibration Frequencies in Electrical Submersible Pumping (ESP) Systems		
Frequency, Multiple of RPM	Probable Causes of Vibration	ESP Part Involved
1/3 to 1/2	Mechanical rubbing	Any stationary and rotating part
<1/2	Vibration in lightly loaded journal bearings	Any journal bearing
1/2	Journal bearing spinning with shaft	Any journal bearing
1	Hydraulic or mass unbalance	Any rotating part
	Rotor running eccentrically in stator	Motor
1 to 2	Misalignment in shaft bearing and/or coupling	Shaft bearings, couplings
	Bent shaft	Motor rotor, any shaft
>5	Poor lubrication, high friction	Any bearing

an eventual failure. Radial bearings in pumps are the most affected components.

- **Torsional** vibrations occur during start-up and when changing pumping speeds. Shafts with limited strength may be twisted or broken.

Vibration effects in ESP units can be minimized by the following:

- preventing **resonant** conditions;
- reducing the severity of vibrations mainly by
 - using as **straight** shafts as possible and
 - reducing the **clearances** in radial bearings; and
- **balancing** of rotating masses.

8.2.2 Typical Failures of System Components

Shafts

The shafts in ESP components are relatively long and very **slender**; therefore small disturbances in their radial stability can result in severe **vibrations** and an eventual failure. Shafts must be strong enough to transmit the high torque of the motor to the pump. Their torque capacity is proportional to the **cube** of the diameter; therefore a small increase in diameter increases their applicability very substantially.

Typical shaft failures in any component of the ESP system (pump, motor, etc.), according to **Wilson et al. [2]**, can be classified as detailed in the following:

- **Torsional yield** failure occurs when the torque capacity of the shaft is exceeded. Permanent **deformation** or a **break** is inevitable at stress raisers such as grooves and undercuts, somewhere between the top rotor of the motor and the first stage of the pump.
- **Torsional twist** in the shaft **absorbs** the energy during system start-up; the bottom of the motor shaft may lead the top of the pump shaft by two or even more full turns. If the shaft twists more than allowed by a connected part, then that part may be damaged.
- **Bearing wear** mostly occurs under abrasive conditions when clearances in radial bearings increase, the shaft's stability is lost, and severe **vibrations** take place.
- **Galling** of bearings is the consequence of lost **lubrication** in the radial bearings of pumps. Because of the metal-to-metal rubbing, local temperature is rapidly increased and bearing materials **weld** together. Because of the high local temperature, the shaft may be softened and broken.
- **Torsional fatigue** is caused by the repeated torsional vibrations for a longer time and can be identified by the existence of fatigue cracks.

Electrical Submersible Pump

Submersible pumps operate under **harsh** conditions and are subjected to the detrimental effects of the wellstream. Their bearings, in contrast to the submersible motor's bearings running in high-quality oil, are lubricated by the **fluid** pumped and thus may fail much earlier than those in the motor. Typical problems leading to pump failures are listed as follows [3,4]:

- **Upthrust wear** occurs when the pump is operated at flow rates greater than the maximum **recommended** pumping rate. In floating-type pumps the impellers are forced up against the diffusers and the upthrust washers may be overloaded and the stage eventually destroys itself; abrasives in the well fluid accelerate the process.
- **Downthrust wear** in pump stages is typical at rates lower than the minimum recommended rate and is exaggerated by the presence of sand or abrasive solids. Downthrust washers are destroyed first before the stages fail.
- **Radial wear** caused by well fluids loaded with **abrasives** increases the clearances in the pump's journal bearings. This usually leads to increased vibrations of the shaft and a further acceleration of wear in the bearings.
- **Erosion** in pump stages occurs when abrasive-laden fluids are produced and can cut through pump housings.
- **Scale buildup** can plug or even lock pump stages.

ESP Motor

Although most of the eventual failures of ESP motors are **electrical** in nature, many other conditions may become the primary cause of motor failure:

- **Overloading** of the motor leads to excessive heating of the motor's wiring; damage or **burnout** may occur. An overload situation is usually caused by:
 - inaccurate design with limited data, leading to **undersized** motor,
 - unpredictably high **specific gravity** of the well fluid, which can cause the actual Total Dynamic Head (TDH) greater than the designed,
 - pump failures increasing the power requirement of the pump, or
 - **irregular** (high, low, or unbalanced) voltage at the motor terminals.
- **Leaking protectors** allow well fluids to enter the motor causing a gradual **contamination** of the motor oil and an eventual motor burnout.
- **Insufficient cooling** of the motor causes the temperature of motor windings to rise; insulating materials may break down, leading to electrical failure. Cooling is unsatisfactory when:
 - fluid **velocity** around the motor is less than that required for an efficient transfer of the generated heat from the motor or
 - the motor is set **below** the perforations and no provisions are provided for an efficient cooling.

Protector (Seal) Section

The protector (seal) section, in normal circumstances, safeguards the electric motor directly connected to it. Protector failures may be attributed to the following factors.

- Broken or damaged mechanical **seals** leak well fluids first into the protector and then into the motor; this may be caused by the following:
 - vibrations transmitted from a worn-out pump or
 - faulty equipment or improper installation.
- The ESP unit's main thrust bearing situated in the protector may fail when the pump operates in **excessive** upthrust or downthrust mode.
- Labyrinth-type protectors may fail in **deviated** (more than 30 degrees from the vertical) well sections.

Rotary Gas Separator

RTGs may fail because of internal **abrasion** due to greater centrifugal forces acting on the solid particles when sand-laden well fluids are produced. Cutting of the separator housing can happen if no special abrasion-resistant materials are used.

Electric Cable

The downhole electric cable transmits the electric power to the motor; its failure immediately leads to system shutdown. The possible causes of cable failures are as follows:

- **mechanical** damage (crushing, cutting, etc.) during running or pulling operations,
- **corrosion** of cable outside surface,
- insulation **deterioration** due to high temperature or the effect of wellbore gases, or
- greater than design **currents** that can rapidly increase cable temperature leading to insulation breakdown.

8.2.3 ESP Teardown Analysis

8.2.3.1 Introduction

Identification of the failures and discovery of their prime causes are usually conducted after the defective ESP unit is pulled when a comprehensive **tear-down** procedure is accomplished. Failures can usually be detected by systematically **dismantling** each component and investigating the status of its parts.

The main objective of ESP teardown analysis is to **diagnose** the cause of equipment failures. Although many times the exact causes may remain

undetected, data from a teardown will surely indicate the **possible** cause of failures, which can be classified as:

- transportation, handling, installation and pulling problems,
- equipment selection and design problems,
- faulty equipment components,
- problems in the electric power supply,
- severe well conditions, and
- other causes.

Lea and Powers [5] give a pictorial review of the teardown procedure and provide a thorough description of typical failures in ESP components. The **API RP 11S1** [6] publication includes **forms** to be used in reporting failures in pumps, motors, protectors, and gas separators. Processing of data gathered during teardown operations and recorded in these reports is facilitated using a **database** utilizing a systematic coding of failures. Such databases stored in electronic form can later be evaluated and are the prime source of information for troubleshooting. Analysis of such data can be used to identify operational problems and compare the performance of different equipment, all with the final aim of increasing ESP **run lives**.

8.2.3.2 General Considerations

Because of the great equipment costs involved, teardown inspections should be closely monitored by the operating **company's** representative. A teardown takes several hours to conduct and is normally done in the factory premises of the **manufacturer**. The representatives of the two sides must **cooperate** closely because both parties have different expertise that can help finding the causes of failures. Proper documentation is highly important; all **records** must be filed and **photographs** of critical parts must be taken, and **samples** of fluids and solids found must be collected.

Before the actual teardown procedure is started the following kinds of information must be collected and properly documented:

- The different pieces of equipment must be positively **identified**; all nameplate data (serial numbers, operating conditions, etc.) are to be checked and recorded.
- Original installation design has a great impact on the ESP system's actual performance; this is why **design** conditions (tubular configurations, well inflow, fluid parameters, etc.) should be properly recorded.
- Operating conditions of the ESP system, including pumping rates, fluid levels, surface pressures, electrical parameters, etc., must be available for all tests made in the past.
- Run and pull **reports** may provide more insight into some specific failure modes.

Having the information just described available during the whole tear-down procedure increases the reliability of the final conclusions. In the following, the main steps of a typical teardown analysis are given; more details with sample photographs can be found in the publications [5,6] mentioned before.

8.2.3.3 Motor Inspection

During teardown, the motor and other equipment are handled in the horizontal position in special fixtures (**vises**). The **pothead connector** (see Fig. 3.19) should remain with the motor after pulling the motor from the well; this necessitates cutting the motor lead extension (MLE) just above the pothead. This way the integrity of the pothead connection as well as the motor can be properly investigated.

The outside surface of the motor **housing** is checked first for the following conditions:

- In the case of **scale** buildup, its level (light or heavy) must be assessed and scale **samples** should be taken for future chemical analysis.
- If the housing is **corroded**, then the type of pitting, the depth, and number of pits, as well as the affected area, are recorded.
- Mechanical surface **damage**, i.e., dents, scratches, or bending, are identified and reported.

With the pothead connection intact, the **integrity** of the motor housing is checked next, using compressed **air** applied to the inside of the motor. The pressure to be applied is usually 75–100 psi, but the manufacturer's recommendation must be followed. Using a **soap** solution, leaks of O-rings, joints, vent plugs, and shaft seals are detected visually.

After removing the pothead connector from the motor, the condition of the **pothead** is evaluated; the pothead connection can be either (1) **plug-in** type or (2) the older and less popular **tape-in** type (see Figs. 3.18 and 3.19). If a plug-in connection is used, one should check the condition of the seal surfaces, the terminals for burn signs, and the O-rings, which may be hard, cut, or melted due to heat effects. For tape-in connections, check wire **splices** for signs of burns, arcs, etc. The connections to motor windings should also be checked; motor leads should be examined for burns between leads or leads to ground.

The lubricating **oil** drained from the motor must be carefully checked and a sample should be taken for laboratory analysis. Oil **color** could be clear, cloudy, or dark; it may contain water in different forms: as an emulsion, in droplets, or wholly water. The most important physical property of motor oil is its dielectric **strength**, which must be measured in every case, even if the oil does not seem to be contaminated. A standard ASTM dielectric strength **test** should be performed using a tester whose electrodes are immersed in the motor oil to measure the

oil's insulation ability. The tester reads kiloVolts (kV) of direct current; the acceptable values are about 30 kVDC, and a reading of zero means water.

The motor **base** is inspected next; after removing the shipping cap from the bottom of the motor, one should look for leaking O-ring surfaces. The O-rings could be hard due to heat problems, or they can be cracked or cut. Many motors have oil **filters** in their bases; these could be clogged by metal chips or any other substances; the presence of metal indicates wear of the motor bearings. In the case of tandem motors, the motor base contains the plug-in connection to the lower unit; this should be checked for burning, bending, and other damage.

At the motor **head** the spline coupling is inspected for damage such as corrosion, burns, and twist. The shaft should turn freely, but contamination or a burnt motor causes the shaft to seize. The extension of the shaft's end from the motor should be checked against manufacturer specifications; a too big extension from the motor can cause damaging upthrust in the pump. The O-ring seal surfaces are inspected for signs of leakage.

The **thrust bearing** should be removed from the motor and its parts must be investigated closely. The thrust runner is fixed to the shaft but must not be stuck to it; if stuck, then it is the effect of excessive heat. The surface of the **runner** should be in good shape: melted or worn-out conditions are signs of excessive heat or pressure. The runner surface may be damaged by excessive thrust loads that have overloaded the unit's main thrust bearing in the protector. A streaky appearance of the thrust bearing indicates that low amounts of water were present.

Now the **rotor stack** is pulled from the motor; it should pull out easily unless distorted by overheating. Signs of burning, indicated by **discoloration**, should be looked for. Any signs of corrosion or a decrease in rotor outside diameter (OD) indicating wear must be recorded.

Rotor **bearings** (see Fig. 3.17) must be checked for spinning inside the stator, indicated by wear on their ODs. The sizes of bearings and **sleeves** are evaluated with go and no-go gauges looking for excessive wear. Thrust washers in rotor bearings may be worn-out or become brittle from heat or chemicals.

Stator **windings** for each phase should be checked with a continuity tester for a phase to ground contact. The electrical resistance of each coil is found using a measuring instrument and the reading should conform to manufacturer's specifications. Motors in good condition usually produce at least a 20 MOhm reading. Any signs of **burning** due to high amperage should be found in motor leads, end coils, stator slots, and windings.

The motor **shaft** should not show any discoloration or abnormal wear along its length.

8.2.3.4 Protector Inspection

After the protector is pulled from the well but **before** it is laid down for shipment, the presence of free water or emulsion must be checked and its location must be noted. This is to prevent migration of water and/or separation of emulsion that may occur in a horizontal position of the protector. Presence of emulsion indicates clearly that the protector was contaminated while it was running.

In the workshop, with the shipping caps present, the outside surface of the protector **housing** is examined for the following:

- **Scale** buildup that can be classified as light or heavy; samples must be taken for laboratory analysis.
- Presence of **corrosion**; the type of pitting, the depth, and number of pits, as well as the affected area, are recorded.
- Surface mechanical **damage**, i.e., dents, scratches, or bending are identified and reported.

Special care must be taken to detect any signs of **heat** damage on the outside of the protector opposite the main thrust bearing. Usually, if the bearing was heavily loaded, this area is more corroded than the rest; too much corrosion here clearly indicates the need for a thorough investigation of the condition of the thrust bearing inside.

The **integrity** of the protector housing is checked with compressed **air** applied to the inside of the protector; usually a pressure of up to 20 psi is applied or the manufacturer's recommendation is followed. While shipping caps are still on the protector, visual detection of leaks through O-rings, joints, vent plugs, etc. is accomplished using a **soap** solution. After removing the shipping caps, the top **shaft seal** is checked with air pressure of 3–4 psi, applied to the protector body while rotating the shaft by hand, where no leaks around the shaft should be observed.

At the **head** and the **base** of the protector, the spline **couplings** on the shaft are inspected for damage such as corrosion, burns, and twist. The protector **shaft** should turn freely; broken shafts should be reported. The **end play** of the protector shaft should be checked against manufacturer specifications by measuring the axial distances from the shaft end and the top flange face at the two extreme positions of the shaft. The O-ring seal surfaces are inspected for signs of leakage. The protector base usually contains an oil **filter** that may be plugged by foreign material; a sample of the solids is taken for analysis.

After loosening the joints of the housing, the **fluid** drained from the protector is inspected; its color could be clear, dark or discolored, or completely black indicating high temperature operation. Water content of the protector

fluid is determined with a dielectric strength test: the presence of any solids content is investigated. Fluids from all isolation chambers (labyrinth or bag-type) of the protector should be examined and documented.

If the protector contains an expansion bag (bladder)—type chamber (see Fig. 3.42), then the condition of the flexible **bag** is assessed: it could be punctured, ruptured, or damaged otherwise. The bag's integrity is checked by an air pressure test: the bag assembly is inflated to about 5 psi and is immersed in water to check for leaks. The fluid check valve is also tested and should open at about 5 psi.

Protectors normally have **labyrinth-type** expansion chambers (see Fig. 3.39); these should be investigated for broken or corroded **breather tubes** and plugged communication **ports**. When checking the integrity of the protector **housing**, higher air pressures are used than required for protectors with bag-type chambers; the manufacturer's recommendation should be followed.

The protector contains the unit's **main** thrust bearing, which works under very harsh conditions and can be damaged if overloaded. When removed from the protector body, the parts of the thrust bearing (see Fig. 3.37) must be closely inspected similarly to the thrust bearing of the motor. However, the main thrust bearing is under a much higher load during operation than the motor and may thus be more damaged. The thrust **runner** could be melted or completely worn-out, or it may be **discolored** if it has run hot but is not totally worn-out. A streaky appearance of the thrust bearing surface indicates operation in some water. Thrust pads (**shoes**) should be checked similarly to the thrust runner. In both cases the evidence of upthrust or downthrust damage should be documented and the extent of wear (negligible, moderate, severe) should be indicated.

Mechanical seals on the protector shaft (see Fig. 3.43) should be disassembled and the condition of each part is assessed. The stationary and rotating **seal rings** may be broken or worn-out, the metal **spring** may be broken, and the resilient **bellows** can become brittle. The mechanical seal faces on the stationary and rotating rings should be inspected for wear marks. Wear **marks** on properly operating (nonleaking) rings must be **concentric** and have the same **width** on both rings; uneven or nonconcentric patterns indicate extensive vibrations, a bent shaft, etc. All O-rings in the protector are inspected for being hard due to heat, seized, melted, or cut.

More detailed information on protector teardown inspection can be found in **API RP 11S7 [7]**

8.2.3.5 Pump Inspection

The **head** and the **intake** of the pump (see Fig. 3.2) is either threaded or bolted to the pump body. They are visually inspected for erosion, abrasion, and

plugging by foreign material. The **housing** should be checked for scale deposition and any surface damage.

The **coupling** on the pump shaft is inspected for damage such as corrosion, burns, and twist. Using a spline wrench, the pump **shaft** is turned in the proper direction and should rotate **freely**; broken shafts should be reported. For pumps with fixed impellers, the shaft's axial movement is measured and compared with manufacturer specifications; excessive shaft **play** indicates worn-out stages. In floater pumps the shaft is easily pulled out by removing the snap rings holding the stages in place.

With the head and intake removed from the housing, the pump **stages** can be retrieved and arranged on a workbench in the order of their position in the pump. All diffusers and impellers must be investigated, starting with the bottom stages and proceeding upward, always looking for the **variation** in observed damage. General condition of stages could be plugged, eroded, or otherwise mechanically damaged.

The observations on **impellers** (see Figs. 2.10 and 2.12) should include the following:

- Both upthrust and downthrust **washers** are checked for wear that may be slight, moderate, or severe; the washers may be brittle or missing.
- Impeller **shrouds** are checked for signs of wear due to upthrust and/or downthrust load; the damage may be slight, moderate, or severe.
- Radial wear is identified by checking the inside diameter of the impeller **hub** using the proper go—no-go gauge; wear can be slight, moderate, or severe.
- The OD of impeller **skirts** is checked with the proper go—no-go gauge; any damage is indicated by a decreased OD.

Investigations on **diffusers** (see Figs. 2.9 and 2.11) should look for:

- Signs of axial wear due to upthrust and/or downthrust loading on **washer pads**; the damage may be slight, moderate, or severe.
- Radial wear is identified by checking the inside diameter of the diffuser **bore** using the proper go—no-go gauge; wear can be slight, moderate, or severe.
- The **O-rings** on the outside of diffusers may be hard, swollen, seized, or melted.
- Any signs of **spinning** during operation should be checked.
- The inside diameter of diffuser **skirt bores** is checked with the proper go—no-go gauge, any damage is indicated by an increased ID.

After the condition of impellers and diffusers is fully evaluated, individual pieces may be **discarded** if beyond repair or can be **reused** if in working order.

8.2.3.6 Cable Inspection

Visual inspection of the whole length of the electric cable is done by **spooling** the cable to another reel and observing its condition. Usual physical damage or deterioration signs are the following.

- Cable **armor** may be corroded, crushed, eroded.
- The **jacket** could be hardened, softened, or swollen.
- Special **coverings** on conductor insulation, if damaged, may not be repairable, especially the lead sheath used against H₂S and other gases. The damaged section must be cut out to restore cable integrity.
- Integrity of the conductor's **insulation** ensures that the cable passes the electrical tests. Insulation may be discolored because of elevated temperatures or it may be hardened, softened, or swollen.
- **Conductor** damage can be in the form of a cut, nick, break, or stretch; damaged sections must be removed.

The **MLE** is under the harshest operating conditions because of the high temperature at the top of the motor and the size restrictions. Because of these circumstances, this item is never repaired and a new one is run at every installation.

Electrical testing of ESP cables is done in several different ways after the cable is allowed to reach a stabilized temperature and to be completely degassed. **Continuity** checks on the cable armor and all three conductors must be performed to find wire breaks. Conductor **resistance** checks using ohm-meters should give uniform readings in all wires.

Detailed electrical tests on ESP cables are presented in the publication **API RP 11S6** [8].

8.2.3.7 Inspection of Other Equipment

In addition to the components described so far, teardown inspection of other parts is also done including gas separators, downhole measuring devices, etc. The **gravitational gas separator** (see Fig. 3.45) is often used to remove small amounts of free gas from the wellstream before it reaches the pump. Its **shaft** should turn freely and its axial play must be checked against specifications. The separator **intake** holes and screen are investigated for plugging; any foreign material such as asphaltenes or corrosion residue should be identified and sampled. After removing the intake and the head, the **couplings** and the **shaft** are inspected for abrasion and other wear. The condition of the pickup **impeller** is assessed and its wear and/or mechanical damage is documented. Accumulation of scale, abrasives, or debris in the **interior** of the separator should be noted.

In wells with high gas production RGSs (see Fig. 4.18) are used; these work under more severe conditions than gravitational separators and may fail

earlier. Their inspection follows the procedure just described for gravitational separators with the following additional investigations:

- Signs of fluid or sand **cutting** should be checked on the housing at the discharge area.
- The **inducer** section of the separator may be plugged (take a sample of the plugging material) or eroded.
- The **separation** section contains a paddle wheel or a single axial pump stage; these parts may also be plugged or eroded, especially if abrasives are produced.

8.2.4 Failure Tracking

Keeping records on ESP failures makes it possible to **detect** the frequently failing **components** as well as to **analyze** operational conditions; these procedures help increase run lives of equipment and help decide on corrective measures. The joint industry project called **ESP Reliability Information and Failure Tracking System** (ESP-RIFTS) started in 1999 [9] collects and analyzes failure data from all over the world and presently contains more than 100,000 records. The project uses its standard failure nomenclature [10] to record failure events in the database. A short description of the failure data structure is given in the following.

For every reported failure event, in addition to the identification of the well, the equipment used, well and operating conditions, the specific details of the failure are classified as given here:

- **Reason for pull.** It defines why the ESP system was removed from the well. If the system was pulled because of a suspected failure, the general and specific reasons for removing it should be stated. Some general reasons, among others, are as follows:
 - **Electrical.** Indicated by abnormal electrical behavior.
 - **Flow problem.** Failure suspected because of flow rate measurement.
 - **Downhole measurement.** Unusual measurement data received from downhole measuring devices.
- **Failed items.** The system components or subcomponents that failed are specified here; these include normal downhole ESP parts such as motors, protectors, pumps, and cables. The actually failed subcomponent must also be defined; this is normally some part of a primary component.
- **Primary failed item.** This is the component or subcomponent for which initial failure led to the final failure of the ESP system. It is not necessarily the most damaged item and its identification requires detailed investigation, preferably accomplished by a teardown analysis.
- **Failure descriptors.** These are normally observed during equipment pulls or teardown inspections and are the main symptoms of damage to

ESP components. They are classified according to the following main categories, and a detailed description of the failures experienced must also be provided.

- **Electrical.** Description of failures related to the supply and transmission of electric power.
- **External.** Failures typically caused by solids in the wellstream such as paraffin, scale, and sand.
- **Material.** Description of physical condition of materials such as color and hardness.
- **Mechanical.** Details of mechanical failures caused by force, pressure, or torque.
- **Failure causes.** Failures can be caused by a variety of factors that can only be identified by detailed analysis. They can be categorized in general and specific kinds; the main general causes are listed in the following. Specific failure causes give more detailed information on the nature of failures under the main cause headings.
 - **System design.** System failures caused by improper equipment selection or design.
 - **Manufacturing.** Substandard equipment design, manufacturing defects, quality problems.
 - **Installation.** Improper installation of ESP equipment in the well.
 - **Operation.** Improper operating or management procedures.
 - **Reservoir or fluid problems.** Unexpected reservoir performance that leads to system failures.

Member companies of the ESP-RIFTS have the ESP system failure database at their disposal and can utilize the data to analyze and predict the behavior of their equipment with the objective of extending the run life of ESP equipment.

8.2.5 Run-Life Estimation¹

8.2.5.1 Introduction

The ESP system's "**run life**" is defined as the active **operating** time of the ESP system in the well, after installation and **before** it fails. It is an important indicator of operational performance, as it is directly related to production losses, workover, and equipment replacement costs; therefore it is a big influential factor in the overall economics of ESP operations. (Note: Because of the great ambiguity existing in the literature most of the terms used in this section have their definitions as per **ISO 14224** [11].)

For a certain population of ESP systems, the variation in **failure rate** with time is normally described by the "**bathtub curve**" (see **Lastra** [12]) that is representative of any mechanical system's typical failure history. As shown in

1. This section was written in cooperation with Dr. F.J.S. Alhanati (C-FER Technologies).

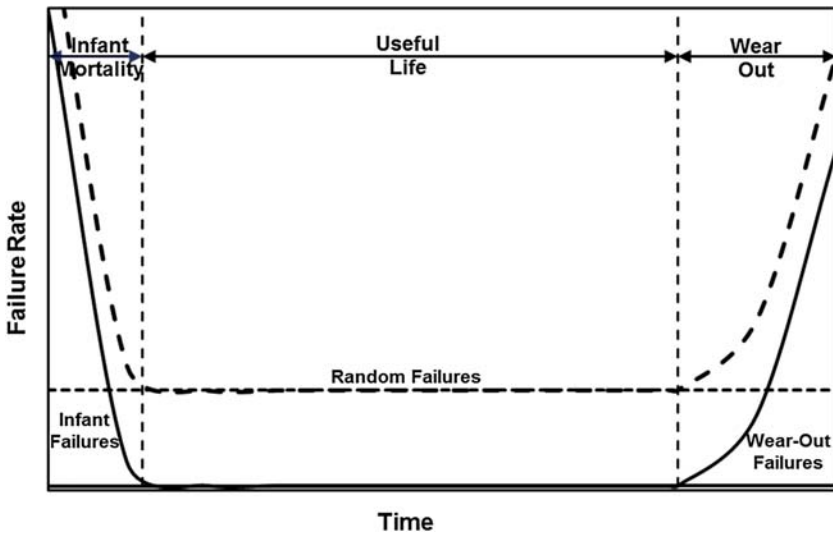


FIGURE 8.1 Typical electrical submersible pumping system failure rates versus time.

Fig. 8.1 failures (shown in dashed curve) happen in three distinct periods in the life of the system:

1. **Infant mortality.** Failure rates are relatively **high** at early times and are due to defects and damages in manufacturing, transportation, handling, and installation. This period is usually **short** (less than a month), compared with the life of the system, and is characterized by **decreasing** failure rates with time.
2. **Useful life.** Failures occur **randomly** at a relatively **constant** rate after the infant mortality period; this is sometimes called the normal or useful life period. This period can be relatively short or quite long, depending on the application and equipment characteristics.
3. **Wear out.** In this final stage the failure rate continuously **increases** because the ESP systems slowly reach the end of their design lives. Natural **wear** of system components due to exposure to well and operating conditions becomes the primary failure mode as different system components start to dysfunction, leading to eventual system breakdowns.

8.2.5.2 Run-Life Measures

Although it is acknowledged that certain failure mechanisms are impacted by the mere exposure of the ESP system to the **downhole** environment (well fluids, pressure, and temperature), it is recommended that (but for exceptional circumstances) only the systems' **running** time, or active operating time, be included as part of the "Exposure Time" in run-life calculations.

When counting failures, it is important to define the boundaries of the system. Usually many workovers are due to tubing failures, but only failures of the main downhole ESP components should be counted as recommended by **Alhanati et al. [13]**. It is also known that failures can occur because of several causes, and many are not related to the quality of the equipment design or manufacturing. Nevertheless, it is recommended that failures due to all causes be counted.

Unfortunately, there is very little consistency in the industry on how to estimate average ESP system run-life for a group of wells. As detailed by **Skoczylas et al. [14]**, while all estimates commonly used in the industry are calculated by summing the “**Exposure Time**” of different systems and dividing the sum by an “**Installation Count**,” very different results are obtained. The results depend on how those two parameters are defined; the possible variations are given in the following.

- **Exposure Time** is the time the ESP system is exposed to the operating environment, i.e., is operational. The most important exposure times, in order of decreasing magnitude, are as follows:
 - **Calendar Time** is calculated between installation time and the time of the next system’s installation.
 - **Time in Hole** is the time between installation and pulling.
 - **Duration** is the difference between the failure and start-up times.
 - **Actual Runtime** is the time during which the system runs.
- **Installation Count** is the number of ESP systems that are considered in calculating run life. Widely used approaches are as follows:
 - **All Systems** is the total number of systems including operating and nonoperative ones.
 - **Running Systems** means the number of operating systems.
 - **Pulled Systems** refers to the number of systems pulled (for any reason) from the wells.
 - **Failed Systems** is the number of systems that have failed.

Obviously, the different variants of the two main parameters introduced previously produce different values and any further calculations using them are inherently affected. Therefore, if run-life performance of different cases is compared, a **consistent** definition of **Exposure Time** and **Installation Count** must be used.

As discussed before, a multitude of run-life measures are obtained by dividing the different exposure times by the different installation counts. Definitions of some of the commonly used such measures are given in [Table 8.2](#).

Most of these statistical parameters are affected by other workovers in the well (for reasons other than an ESP system failure) and are not good

TABLE 8.2 Commonly Used Run-Life Measures

Run-Life Measure	Calculation
Average Runtime of All Systems	Total exposure time of all systems divided by the total number of systems.
Average Runtime of Running Systems	Total exposure time of running systems divided by the total number of running systems.
Average Runtime of Pulled Systems	Total exposure time of pulled systems divided by the total number of pulled systems.
Average Runtime of Failed Systems	Total exposure time of failed systems divided by the total number of failed systems.
MTTF = Mean Time To Failure	Total exposure time of all systems divided by the number of failed systems.

measures of the ESP system reliability [14]. The measure **Mean Time to Failure** (MTTF), however, is a good indicator of the ESP system's reliability; the greater its value, the more reliable is the system. It is calculated by dividing the total exposure time of all systems (**Actual Runtime, Duration**) by the number systems failed (**Failed Systems**); it is a statistical parameter calculated over a large number of units and over a long period of time. According to its definition it represents the **average** operating time of ESP systems **before** failure.

Therefore, it is recommended [14] that MTTF be used as the main **measure** of ESP **run life** when evaluating performance over time, as well as forecasting failure and workover frequencies into the future. Other run-life measures, as those listed above, should be considered “complementary” measures to MTTF, helping one have a good overall picture of a certain operation. Many authors refer to mean time between failures (MTBF). However, the standard definition includes the time required to put the system back (restore) in operation (MTTRes) after a failure; therefore $MTBF = MTTF + MTTRes$ [15]. Therefore, this is not a good measure of system run life, or a good indicator of the system reliability.

Run-life measures can be calculated using a “moving window” in time, considering systems operating/pulled/failed during a defined period (as opposed to all systems installed/pulled/failed up to the current time). This allows for recent changes in the measure to be more apparent. One- or two-year windows are normally used; however, when only few systems are being considered, it may be necessary to increase the window size further, to be able to reduce the noise in the data and identify true trends.

8.2.5.3 Lifetime Distribution

Today it is universally accepted, as first proved by **Patterson** [16], that failures of ESP systems follow the Poisson “**pure death**” theory of exponential life-time distribution. This statistical function describes the **survival** (i.e., operation without failure) of a group of ESP systems and is often called the **Survival Curve**, $S(t)$. It shows the probability of a system surviving (not failing) a certain operating time in a population where the failure rate is **constant** over time. The probability of survival gives the full picture of the ESP system’s **reliability** and is a function of time:

$$S(t) = e^{-\lambda t} \quad (8.1)$$

where:

$S(t)$ = fraction of ESP systems surviving until time t , —,

λ = average failure rate, 1/day, and

t = time, day.

The **Survival Curve** can be empirically established for a group of ESP systems by observing the number of systems still operating at different times during a long period and plotting the data versus time. As proved by many investigators, empirical data can be closely fitted by the probability function given in Eq. (8.1).

It can be easily proved by calculus that the **area** under the Survival Curve (i.e., the integral of Eq. 8.1) equals the reciprocal of the average failure rate, i.e., $1/\lambda$, of the population examined. Because average failure rate and MTTF are inversely proportional, it follows that the **area** under the Survival Curve is **identical** to the **MTTF** value; this feature permits the determination of MTTF from empirical data.

A challenge is that a complete **Survival Curve** is usually not available for the population of interest, as there are usually many systems in an operating state with relatively high exposure times for which the actual failure times are still unknown. A common practice is to fit a certain lifetime distribution to the population and then use the same to estimate MTTF. Using the “**pure death**” distribution the MTTF estimate is equal to the reciprocal of the average failure rate, or equal to the sum of the exposure time for all systems, divided by the number of failed systems.

Fig. 8.2 presents example Survival Curves for different MTTF values, calculated using Eq. (8.1). As seen, the number of operating systems **exponentially** decreases with time, but the reliability of the systems (MTTF) has a big effect also. For example, at MTTF = 300 days half of the ESP units have failed after 200 days of operation. At the same time, however, more reliable systems with an MTTF of 900 days have 80% of the units still operating without failures at the same time.

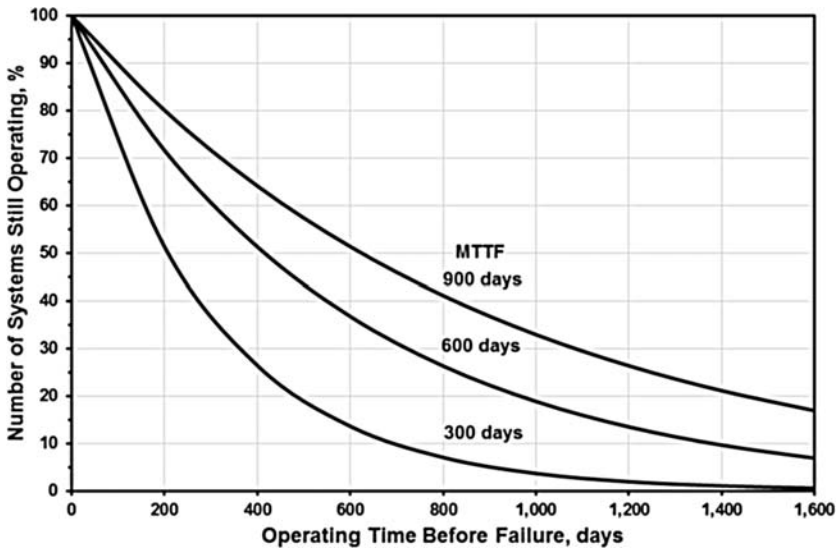


FIGURE 8.2 Example survival curves.

8.3 TROUBLESHOOTING ELECTRICAL SUBMERSIBLE PUMPING INSTALLATIONS

As discussed in Section 7.4, monitoring of ESP operations went through an evolution over the years. Troubleshooting procedures followed the same trend, and the early simple methods have evolved to the highly sophisticated procedures of today. In the following, first the use of ammeter chart technology is detailed, after which the state-of-the-art procedures are described.

8.3.1 Interpretation of Ammeter Charts

Operational problems in an ESP system usually result in an electrical failure. This is the reason why a continuous monitoring of the electric **current** drawn by the ESP motor is a valuable tool for maintaining a sufficiently long equipment run life. Current monitoring is achieved by a **recording ammeter** located in the motor controller or switchboard. It monitors the input amperage to the motor by using a current transformer coupled to one of the electric cable's conductors. The current is recorded in the function of time on a circular **chart** with the proper scale.

The proper interpretation of ammeter chart recordings, especially the evaluation of its variation with time, can provide valuable information for the detection and correction of minor operational problems before they become expensive major ones (see **API RP 11S [17]**).

The following figures (Figs. 8.3–8.16) show representative examples of ammeter charts after Ref. [3] with short descriptions of the causes and possible preventions of typical electrical failures.

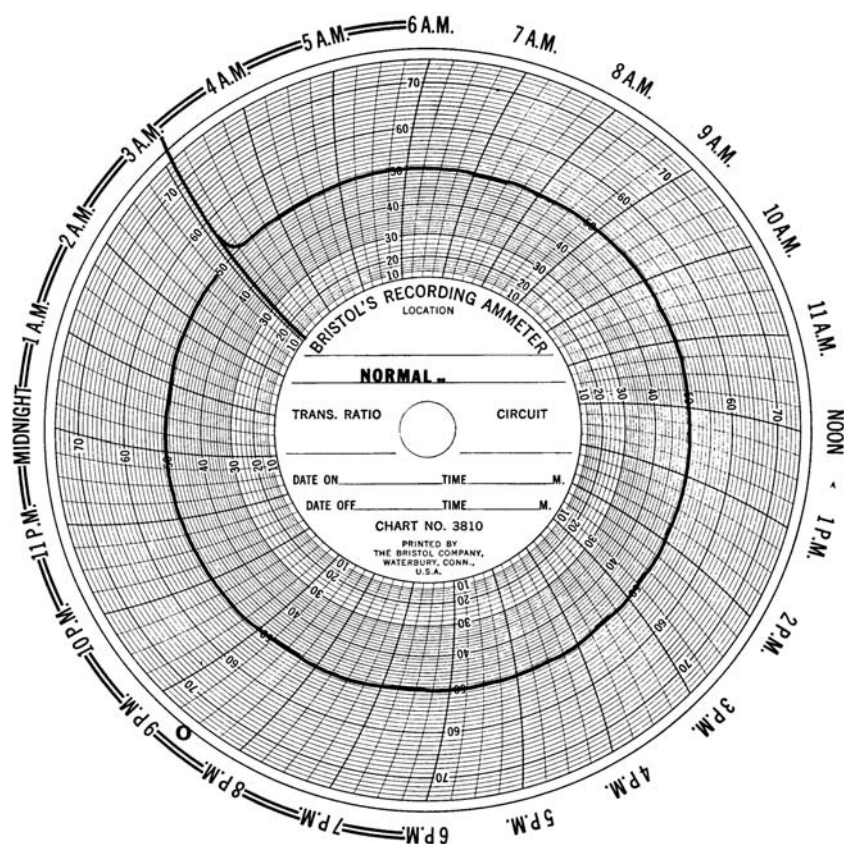


FIGURE 8.3 Example ammeter chart: normal operation.

Discussion

This is a typical chart for a properly designed installation. At start-up (3 a.m.) the motor draws four to five times higher current than during normal operation, but motor current stabilizes instantly at a level close to the nameplate (50 A) current.

In these ideal conditions, the ESP motor develops its nameplate power, which is sufficient to lift the designed liquid rate against the TDH developing in the well tubing. The extremely high currents at start-up are normal and last only a fraction of a second and do not seriously damage the equipment.

Depending on the actual loading of the motor, the stabilized current may be lower or higher than the motor's nameplate rating. Proper operation is indicated by a consistent, smooth curve at a constant current level for long periods.

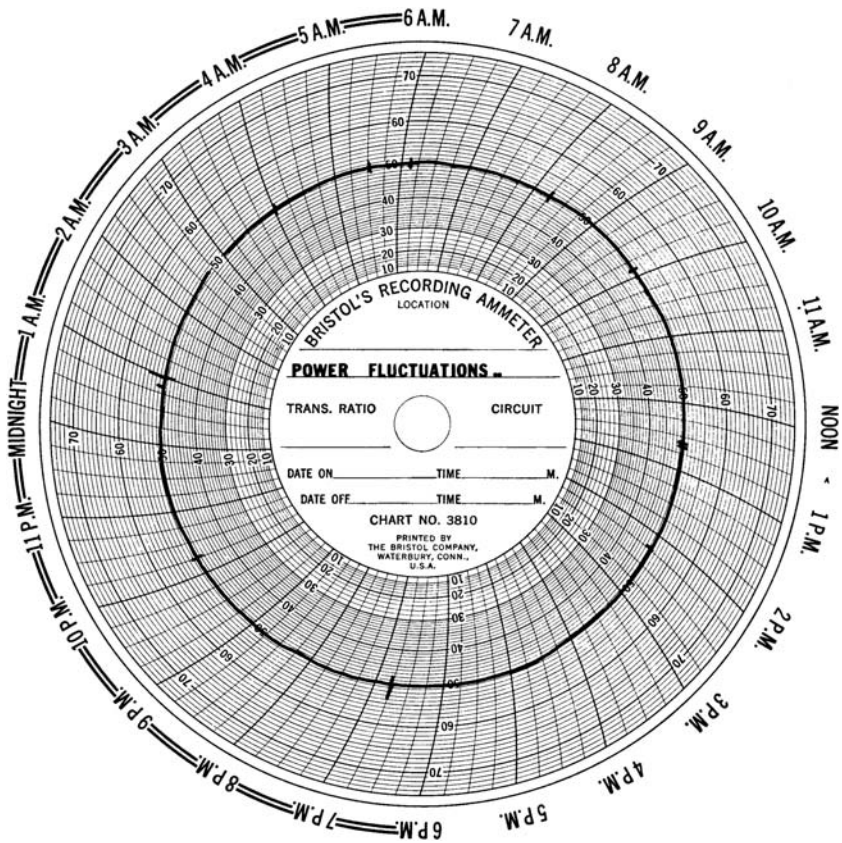


FIGURE 8.4 Example ammeter chart: electric power fluctuations.

Discussion

This is how motor current of a properly designed installation changes for fluctuating power supply voltages.

Because the ESP unit works under ideal conditions, it develops a constant power and draws a constant current. The spikes in motor current indicate the instances when primary voltage in the supply system drops. At those times the motor draws high instantaneous currents to overcome the power requirements of the electrical submersible pump.

Some possible sources of primary voltage fluctuations are (1) sudden heavy loading of the power system, as caused by starting up a high horsepower electrical load, and (2) electrical disturbances, such as a lightning.

Remedies

The supply system should be investigated for extreme loads.

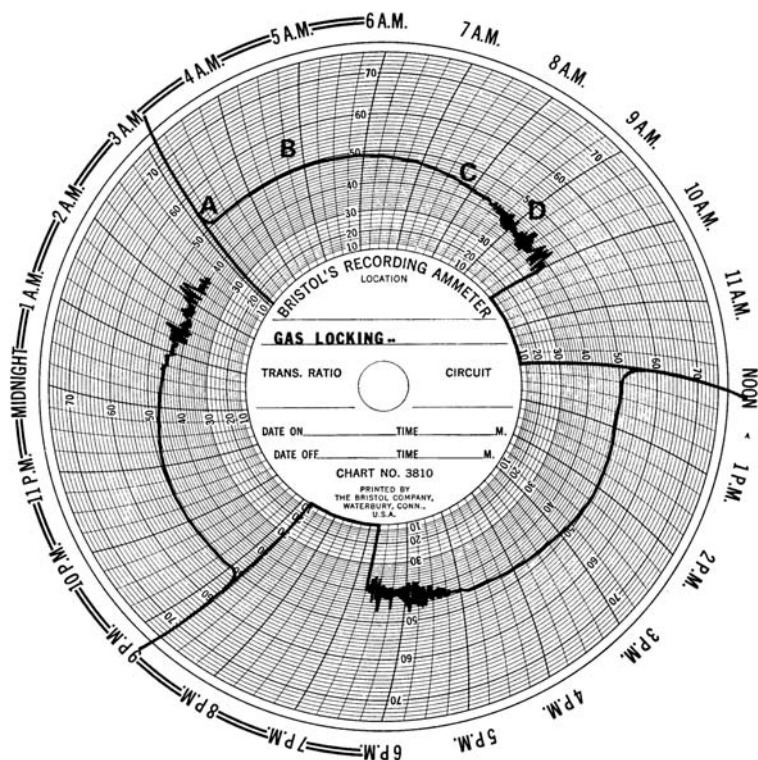


FIGURE 8.5 Example ammeter chart: pump-off with gas interference.

Discussion

Charts similar to this are observed when the capacity of the ESP unit is greater than the inflow to the well and the well produces substantial free gas volumes.

After start-up (Section A) the annular liquid level is high, the pump works against a reduced head and draws a high current. As annular level drops, the pumping rate approaches the design value and motor current decreases to the nameplate value in Section B. Because the pump is oversized, as compared with well inflow rates, annular liquid level drops further in Section C, as indicated by decreasing motor currents. Because of the decreased annulus pressure, solution gas breaks out, making fluctuations in pumping rate and motor current in Section D. As the fluid level approaches the pump intake, cyclic production of free gas and liquid results in erratically low motor currents. Finally, the unit is shut down by the undercurrent protection circuit in the switchboard.

A “gas locked” situation occurs if motor currents in Section D do not drop to a level previously set for undercurrent protection. The unit then continues to operate with the motor drawing a reduced but constant amperage without any liquid production taking place. The electrical submersible pump is now totally gas locked and free gas occupies the eye of the pump’s impeller, preventing the pumping of the liquid phase.

Remedies

Increase pump setting depth or reduce the pumping rate by installing a surface choke.

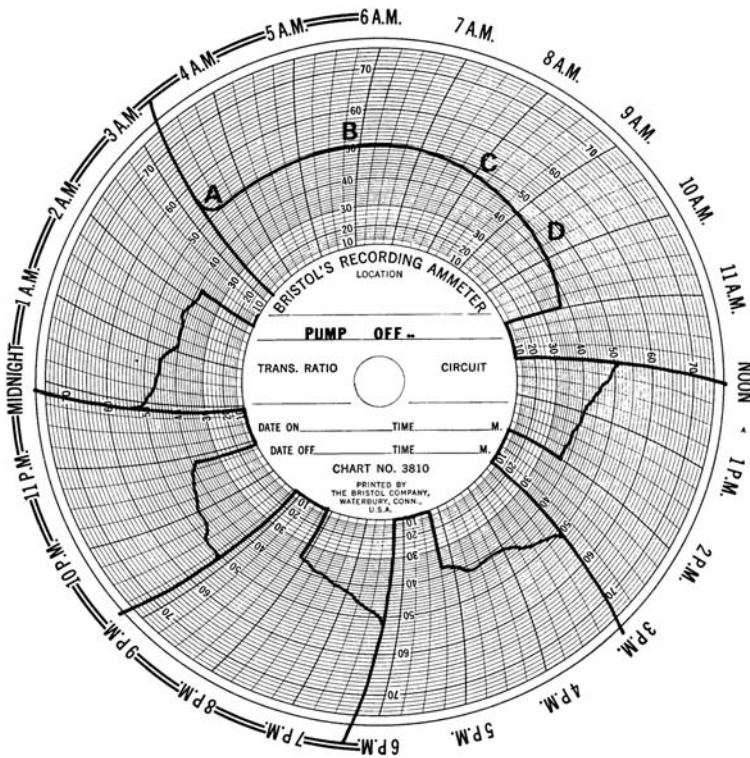


FIGURE 8.6 Example ammeter chart: pump-off condition.

Discussion

This chart illustrates a typical cyclic operation in a pumped-off well with negligible gas production, caused by insufficient well inflow, as compared with the capacity of the ESP equipment.

After start-up, motor current rapidly decreases to the design value because the initial high liquid level in the annulus drops and the head required increases to its design level (Section A). In Section B the pump operates as designed, but the limited inflow of well fluids causes the annulus liquid level to drop further in Section C. In Section D the fluid level approaches the pump intake; pumping rate and motor current decline when the surface switchboard shuts down the unit at a preset undercurrent level. After a preset time delay, automatic restart is triggered and the unit is restarted. By this time, the fluid level in the annulus has risen and a new production cycle begins. As before, the pump rapidly removes the liquid from the annulus, causing the level to drop and the unit to shut down due to the detection of an undercurrent situation. As seen, the second cycle is shorter because the annulus liquid level did not reach its static value.

Remedies

Install another ESP unit with a production capacity that properly matches the inflow rate of the well.

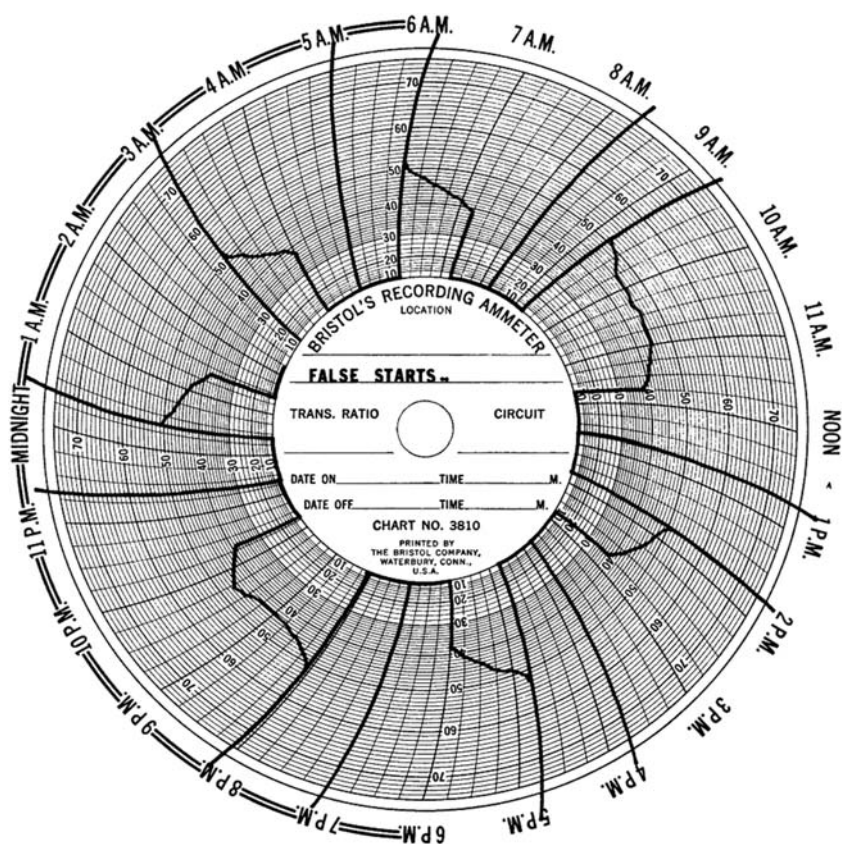


FIGURE 8.7 Example ammeter chart: pump-off with restart failure.

Discussion

Typical cyclic operation caused by limited inflow to the well, negligible gas production, and improper time delay setting.

The explanation of the chart is identical to that describing the pumped-off condition with the only difference that the length of the time delay is insufficiently short for allowing a proper liquid level buildup in the annulus. The unit is shut down again immediately after the automatic start because of the lack of liquid to be pumped.

Remedies

Set the time delay so as to allow for a sufficient buildup of liquid level in the annulus.

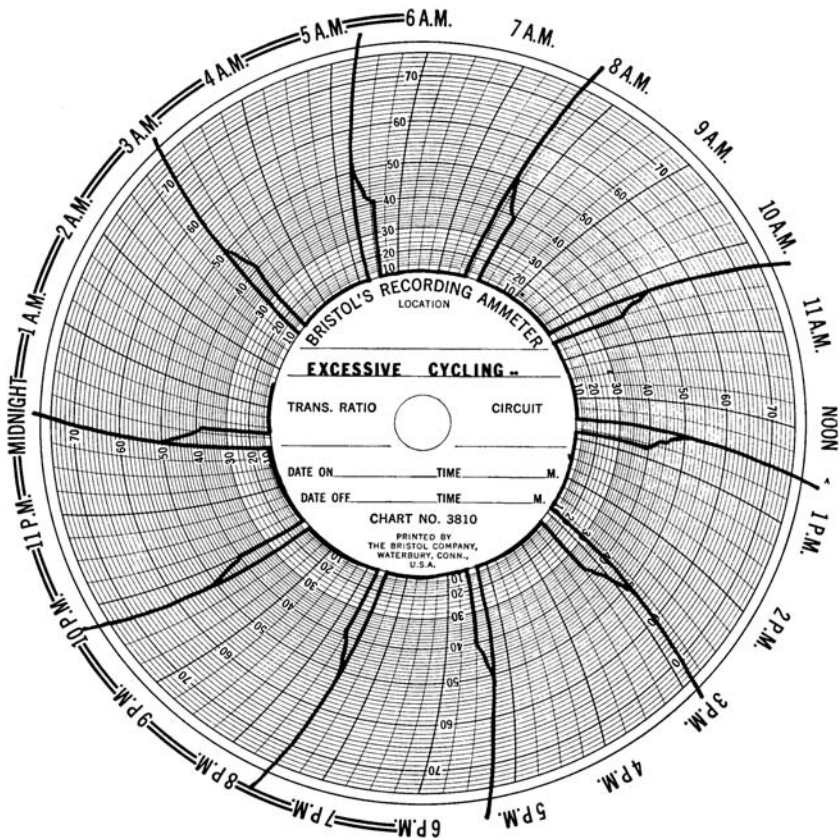


FIGURE 8.8 Example ammeter chart: frequent cycling.

Discussion

Typical chart found on wells with a much lower inflow than the capacity of the ESP equipment run in the well.

The chart is very similar to that described for pumped-off wells, but cycles are more frequent with much shorter running times. Because frequent starts are very detrimental to the longevity of ESP motors because of the extremely high start-up currents, units experiencing frequent cycling must be shut down and investigated as soon as detected.

Remedies

This is a very dangerous situation for ESP motors and usually occurs when (1) pump capacity is too high or (2) the head developed by the pump is insufficient. By shutting in the wellhead for a short time and observing the shut-in pressure, one can find the actual cause. Anyway, redesigning and running the proper ESP unit in the well must be done immediately.

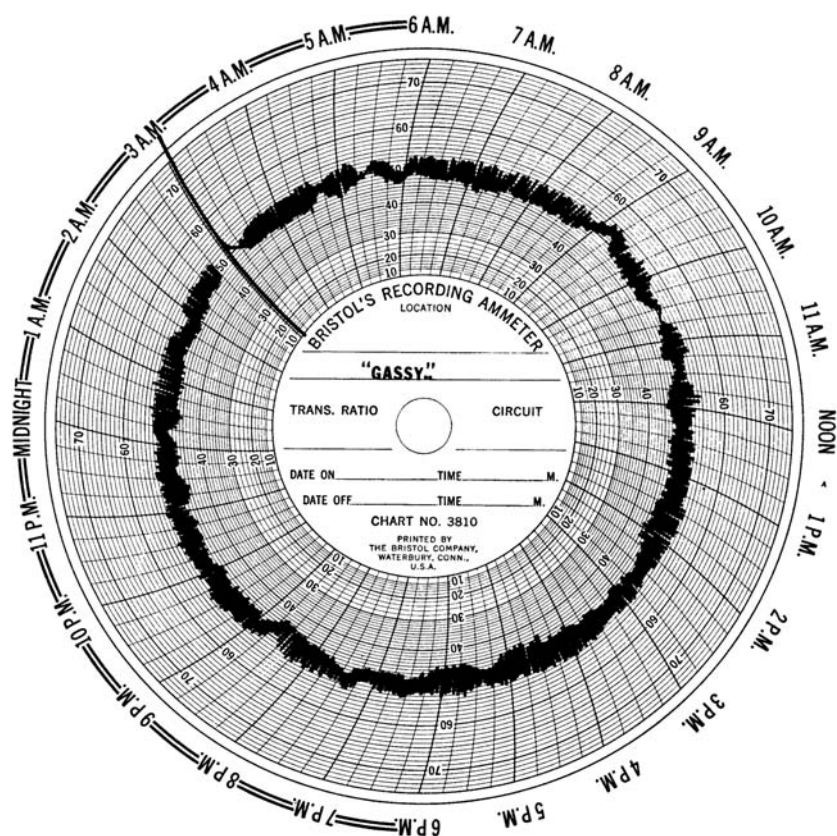


FIGURE 8.9 Example ammeter chart: free gas production.

Discussion

The chart shown here is typical for a properly designed installation producing a gassy fluid.

The frequent and regular fluctuations in motor current are caused by the free gas present at pump suction. Because the gas content of the fluid entering the pump changes rapidly, the power requirement of pumping changes accordingly. This causes the current drawn by the motor to fluctuate, as recorded on the ammeter chart. The general consequence of such operations is the substantial reduction of liquid production rate.

Remedies

Proper adjustment of the casing and/or tubing pressure can reduce this kind of behavior.

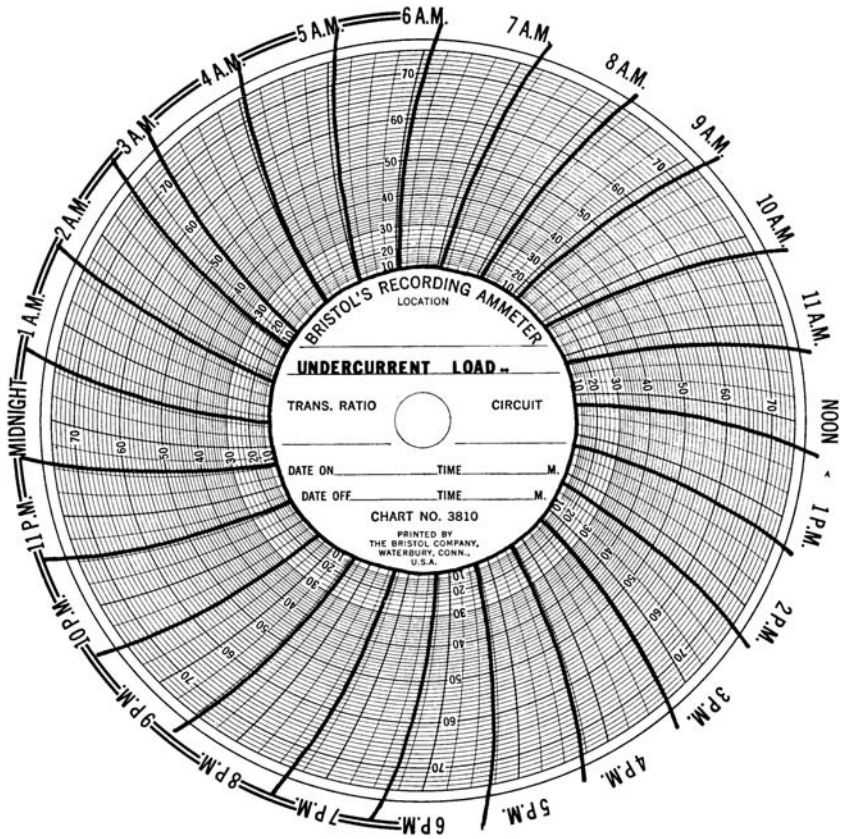


FIGURE 8.10 Example ammeter chart: undercurrent shutdown.

Discussion

This behavior is typical in situations when motor current does not attain the undercurrent setting. Every time the time delay expires the unit is shut down immediately after start-up.

After starting, the motor immediately is shut down by the undercurrent relay because the power required by the pump is too small. This can be caused by a broken unit shaft, a too light well fluid, or a low liquid rate.

Remedies

In case sufficient well inflow is present, lowering the undercurrent setting can work.

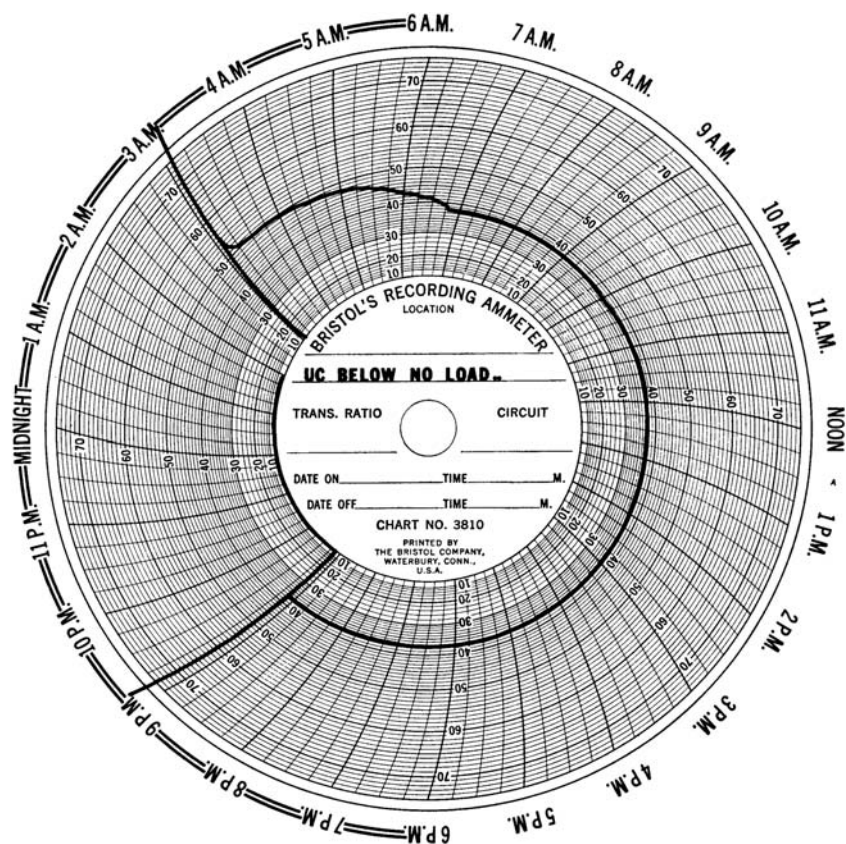


FIGURE 8.11 Example ammeter chart: undercurrent failure.

Discussion

The chart shown here illustrates a failure mode of ESP equipment typical for oversized units with improperly set undercurrent protection.

After start-up the ESP unit pumps the well down and pumping rate decreases to a point where it should be shut down by the undercurrent relay. This, however, was improperly set below the idle amperage of the motor; therefore the motor continues to operate with almost no liquid pumped. Without sufficient cooling, the motor heats up and eventually burns.

Remedies

The ESP unit is oversized and must be changed to a properly designed one. Another problem is the setting of the undercurrent relay; the threshold current must be increased.

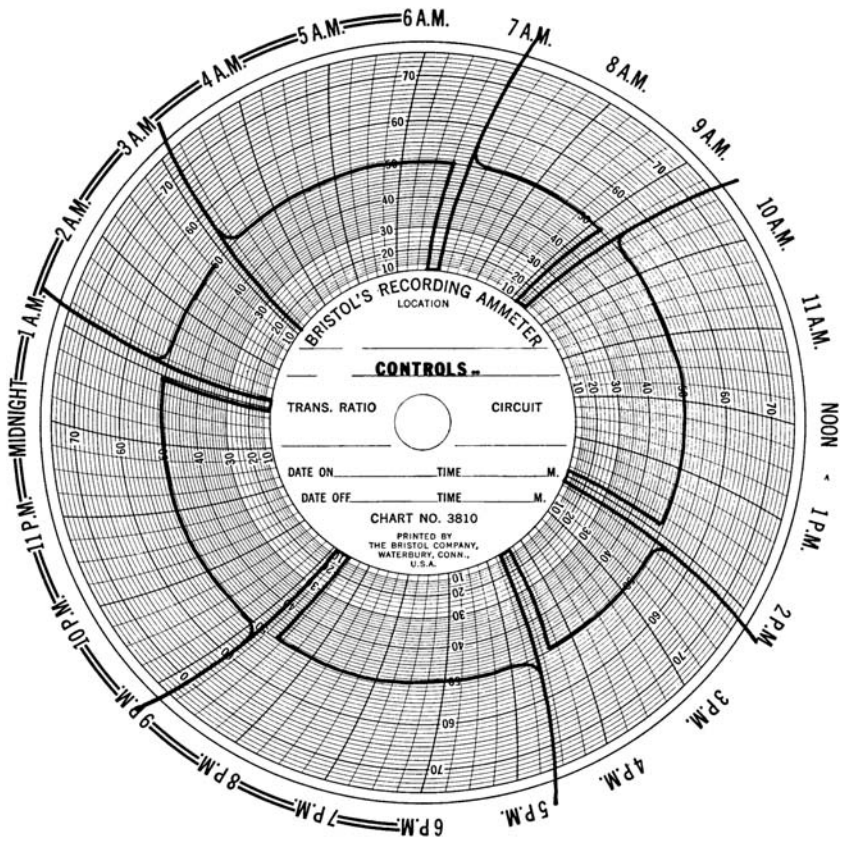


FIGURE 8.12 Example ammeter chart: improper tank level control.

Discussion

This example illustrates an ESP unit controlled by a tank level switch with an improperly set restart delay.

Sometimes the ESP unit is controlled by a tank switch that stops the units when the autorestart counter is activated. The time delay set in this case is too short for the produced fluid to fall back from the tubing. Because the ESP unit is spinning backward while the tubing is emptied, starting it during this period may result in severe damage to ESP components, especially to shafts. To prevent this to happen, a minimum of 30 min of delay is recommended after each shutdown.

Remedies

Set the time delay to a minimum of 30 min.

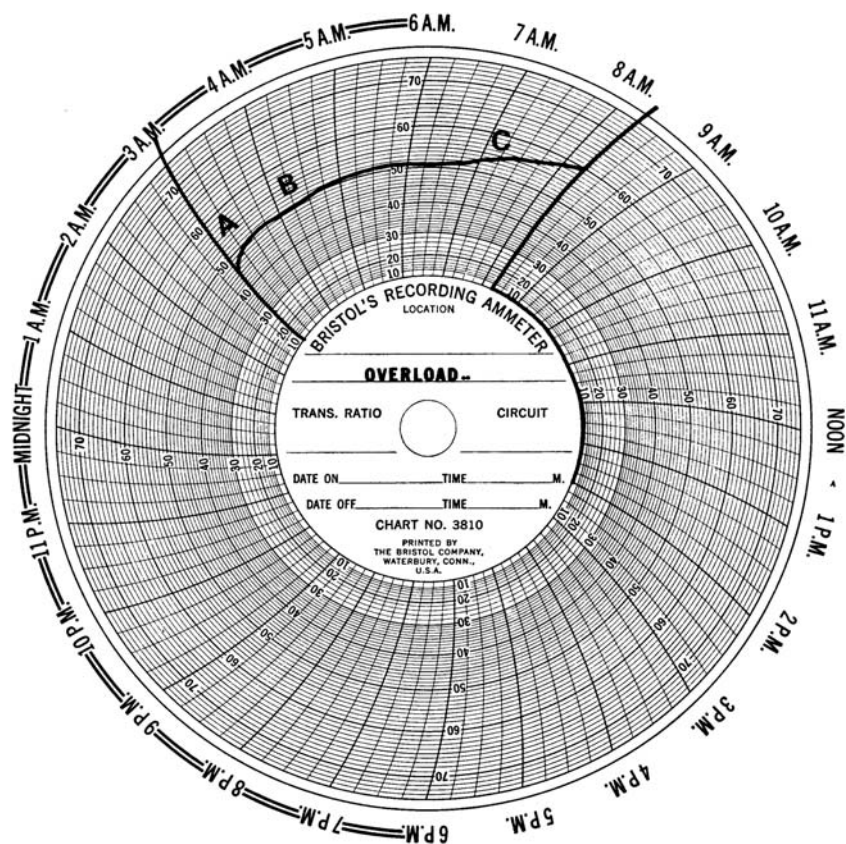


FIGURE 8.13 Example ammeter chart: overload condition.

Discussion

The chart illustrates normal shutdown of ESP equipment due to current overload.

After start-up in Section A, motor current gradually increases to the nameplate value, a normal situation. In Section B the motor runs normally, drawing a constant current. Finally, in Section C, motor current continuously rises until the unit is shut down by the overload protection relay. The cause of the overload cannot be determined from this plot alone, and the unit must not be restarted until the cause of the overload is detected and corrected.

Remedies

A complete analysis of the possible causes of overload must be made. Equipment overload can be attributed to an increase in the density or viscosity of the produced fluid, sand or emulsion production, or mechanical problems such as worn-out equipment.

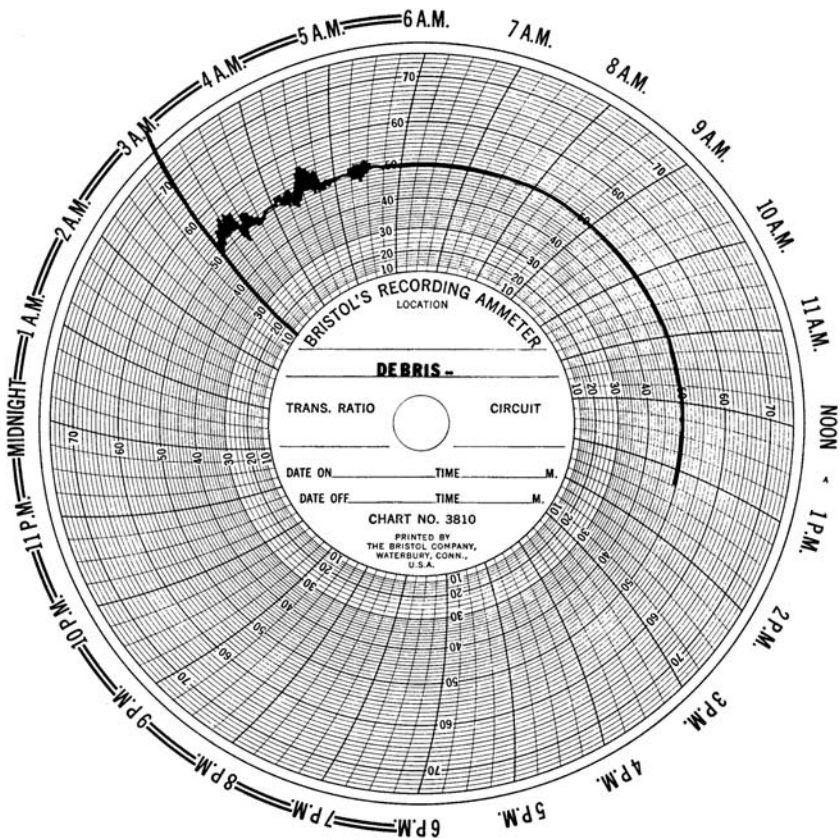


FIGURE 8.14 Example ammeter chart: initial cleanup of the well.

Discussion

This is a normal chart for installations where the proper operation is assumed after the well cleans up.

The fluctuating currents after start-up are caused by changes in the specific gravity of the well fluid. Production of heavy kill fluids or solids may be the main sources of gravity increases. After the well cleans, operations are normal.

Remedies

Make sure the motor is capable to pump the heavy fluids, which may be present in the well after workover operations. If possible, use light fluids for well killing.

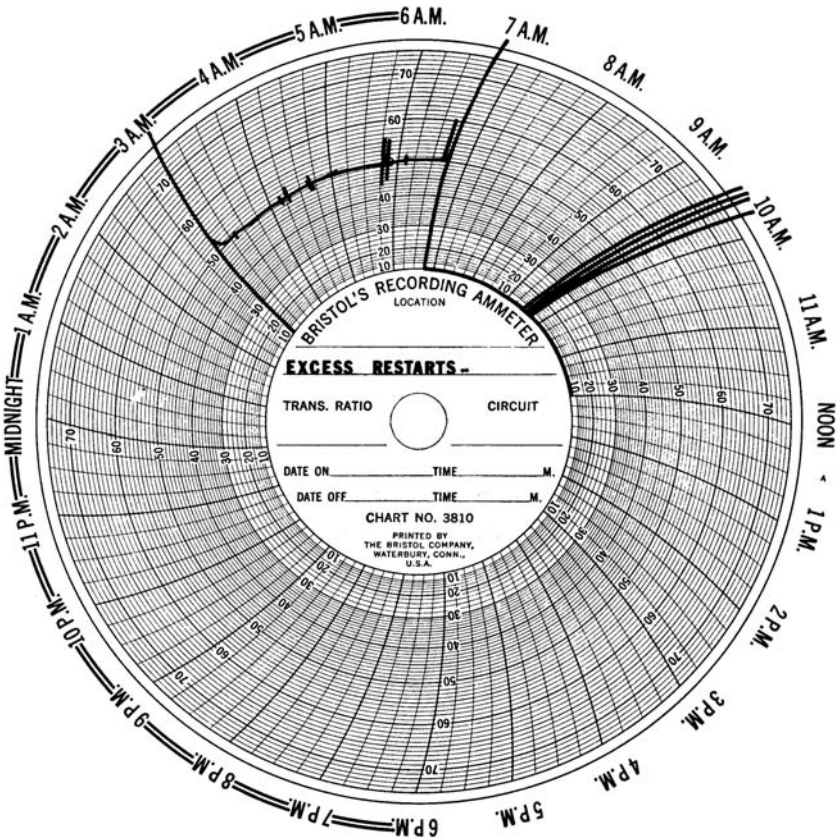


FIGURE 8.15 Example ammeter chart: excessive manual restart attempts.

Discussion

The chart shows several manual attempts to start the ESP unit after it was shut down because of an overload condition.

It is a very bad practice to try manually starting a previously overloaded and automatically shutdown motor without first detecting and eliminating the cause of overload. Several such attempts will eventually destroy and burn out the motor.

Remedies

Always check the ESP unit before attempting a manual restart after an overload shutdown.

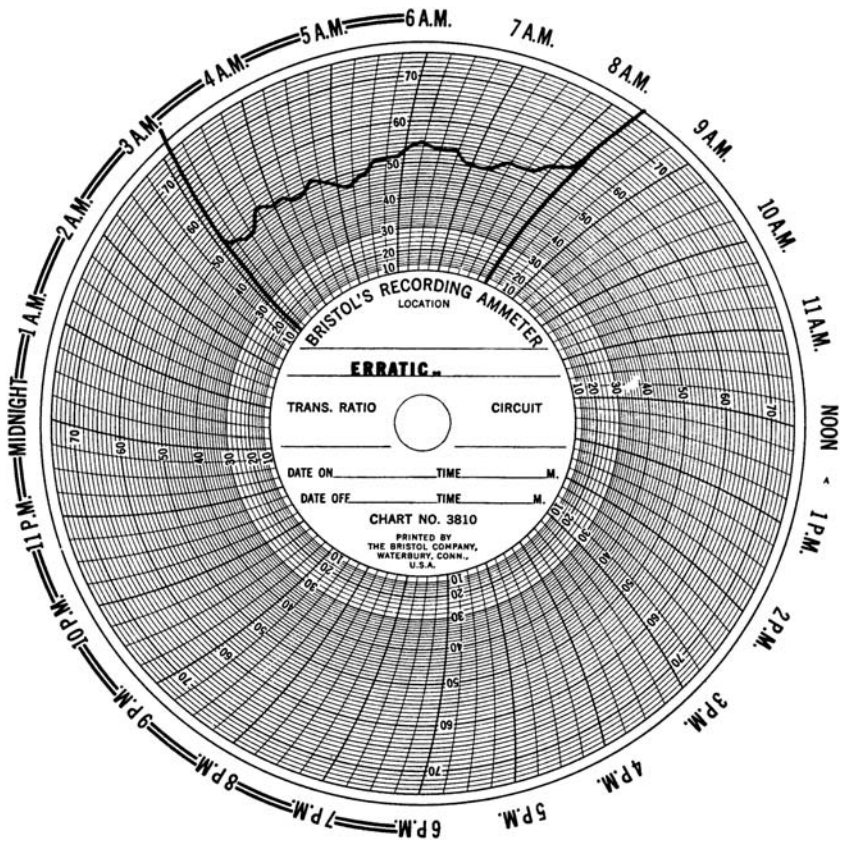


FIGURE 8.16 Example ammeter chart: erratic loading conditions.

Discussion

Before being shut down because of an overload situation, the current drawn by the motor varied very erratically.

The frequently changing motor current can be caused by wide fluctuations in fluid specific gravity and/or surface pressure or mechanical failures in the pump or motor.

Remedies

A thorough analysis of the failure must be made before a manual start is attempted.

8.3.2 Nodal Analysis Techniques

8.3.2.1 Introduction

The use of ammeter charts to troubleshoot ESP installations, although it has been a successful practice for many decades, provides a very “**one-sided**” picture of the ESP unit’s operation. Because it relies on electrical measurements it can positively indicate changes in the **electrical** system but seldom gives enough information to pinpoint the **root cause** of the operating problem. It must be remembered that electrical failures are very often caused by mechanical or other problems, which, over time, finally develop into a failure of electrical nature. The detection of the initial or root failure, therefore, is not an easy task and requires additional information and an experienced analyst. **Williams et al.** [18] very fittingly compare the use of ammeter charts to a case when, instead of looking at the speedometer, one tries to find a car’s speed from its fuel consumption.

Modern ESP controllers use **microprocessors** to provide a greatly improved control and protection of the ESP system’s electric components [19]. Instead of the ammeter chart recorded on paper, they use several other electrical variables than the current drawn by the motor and can **display** and **store** those for immediate or future analysis and for control. Pump-off condition, for example, is much easier detected if the changes of motor power (kW) are analyzed.

The basic shortage of using ammeter charts is that this type of analysis does not and cannot cover all aspects of the ESP system. The ESP installation works as a system consisting of (1) **mechanical**, (2) **hydraulic**, and (3) **electrical** components and, if a comprehensive description of its behavior is required, it must be treated accordingly. **Systems (NODAL) Analysis** is the ultimate way to describe the system’s operation and this approach is the proper one to be followed also for troubleshooting.

8.3.2.2 Applications

In the following, the comprehensive procedure of **Williams** [18,20] is described, which utilizes the **NODAL analysis** principles discussed in detail in Section 6.2.

Let’s assume an ideal case for troubleshooting: the system is equipped with a **DHM** (downhole measurement) unit and the following parameters are accurately measured or known:

- on the surface:
 - wellhead pressure, WHP,
 - production rate, q_l ,
 - dynamic liquid level, L_{dyn} , and
 - driving frequency, f_{actual} .
- by the DHM unit:
 - pump discharge pressure, p_d , and
 - pump intake pressure, PIP.

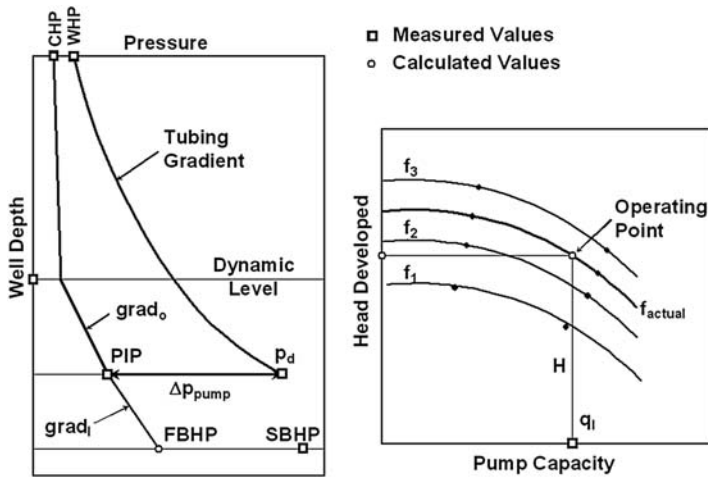


FIGURE 8.17 Troubleshooting plot for electrical submersible pumping systems.

- from well files:
 - static bottom-hole pressure, SBHP, and
 - the well's productivity index, PI.

Based on these data and considering the hydraulic and the mechanical components of the ESP system, two graphs shown in Fig. 8.17 can be constructed, which properly describe the system's behavior. Measured parameters are denoted by small squares; calculated ones by circles. The graph on the left-hand side represents the **cooperation** of the well and the productive formation and shows the pressure **traverses** in the well tubing and in the annulus. (For a detailed description consult Fig. 6.6 and its **Discussion**.) The graph on the other side describes the operation of the electrical submersible **pump** at several driving frequencies including the actual one. On each curve, the two limits of the recommended operating range are indicated by dots.

The actual head developed by the pump can be calculated from the **measured** discharge and intake pressures as follows:

$$H = \frac{p_d - PIP}{0.433 \gamma_l} \quad (8.2)$$

where:

- H = head developed by the pump, ft,
- p_d = pump discharge pressure, psi,
- PIP = pump intake pressure, psi, and
- γ_l = specific gravity of the produced liquid, —.

This head, if plotted versus the measured liquid rate, q_l , defines the **operating** point of the ESP system, which must fall on the pump's head performance curve valid at the actual driving frequency. For properly designed systems, this point lies somewhere between the two limits of the recommended liquid flow rate range, as shown in the figure. This is how an ideal system with all production data accurately measured should look like. To check the integrity of the measured data, one calculates the flowing bottom-hole pressure, *FBHP*, from the measured *PIP* with due consideration to the calculated pressure drop in the casing string. Then using the well's known productivity the inflow rate into the well, Q , can be determined. In the case of accurate input data, this rate must be identical to the measured value, q_l .

If the operating point does not fall on the performance curve the pump may be **worn** out and its head performance curve may have shifted to lower heads. Similarly, if the pump is in good condition but the operating point is outside the recommended range, the original installation design must be checked because the pump operates in **upthrust**, or **downthrust** mode.

The use of the pressure traverses (the left-hand side of Fig. 8.17) allows the **validation** of several important parameters not readily available or suspected to be in error. The required procedures differ for each part of the pressure distribution:

- **Flow in the tubing.** As shown in Fig. 8.17, the measured wellhead and pump discharge pressures give the two extreme points of the flowing pressure traverse in the tubing. Unknown or inaccurate parameters (water cut, GLR, fluid gravity at pump discharge, etc.) can be validated by repeatedly running multiphase pressure drop calculations with several of their assumed values. With trial and error, the proper parameters are found when the calculated and measured pressures at the two extremes of the pressure traverse **match** each other.
- **Electrical submersible pump.** From the measured pump discharge and intake pressures the head developed by the pump is found using Eq. (8.2). Pump head is a function of the driving frequency, flow rate, number of pump stages, fluid gravity, and pump efficiency; the exact values of some of these may be suspect. If pump head plotted versus liquid rate fits the pump performance curve, data are validated. If not, then an analysis of the influencing parameters may give the reason for improper performance.
- **Well section below the pump.** It is an easy task to calculate the FBHP using single or multiphase pressure drop calculations. This value can be checked for accuracy by comparing it with the pressure found from the well's inflow performance relationship.

The continuous **monitoring** of crucial downhole parameters, as detailed in Section 7.4, enables the production engineer to perform the type of analysis just described at several times. Results of such analyses greatly improve the chances of **detecting** mechanical or hydraulic problems long before they lead

to fatal electrical failures. By following the proper procedures, the **run life** of the ESP system can be increased, a never-ending goal of the operator.

8.3.2.3 The Use of Measured Pump Intake and Discharge Pressures

When using novel DHM units the electrical submersible pump's intake and discharge pressures are monitored continuously and their values are available on the surface in function of time. Analysis of the variation of these pressures allows the detection of several operational problems according to a model developed by **Bruijnen** [21], as detailed in the following.

This troubleshooting model is based on **Nodal analysis** calculations performed on an ESP system with a packer above the pump as shown in [Fig. 8.18](#). The analysis is special because two **Solution Nodes** are selected simultaneously and appropriate calculations are performed using both nodes. The two nodes are (1) the intake (point 3) and the (2) discharge (point 4) of the

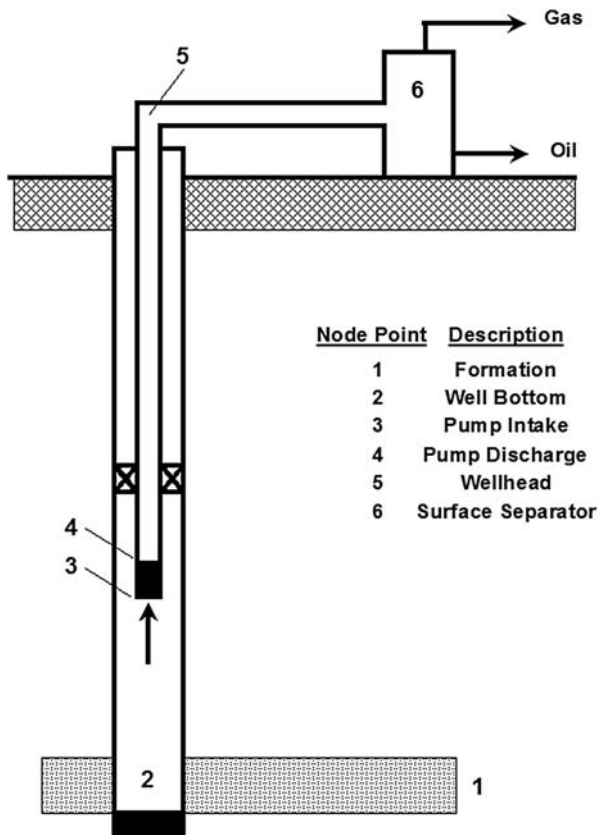


FIGURE 8.18 Nodal system used in the Bruijnen model.

electrical submersible pump. As normally done, at both solution nodes the **inflow** and **outflow** pressures are calculated and equated, according to basic Nodal principles (see Section 2.5.3).

The production system given in Fig. 8.18 can be divided into three subsystems: (1) the formation and part of the casing string (between nodes 1 and 3), (2) the electrical submersible pump (between nodes 3 and 4), and (3) the separator-tubing system (between nodes 4 and 6). The performance of each of these subsystems is easily established using the Nodal theory: starting from a known node point, flowing pressure is determined by considering the pressure drops occurring in the different components. This way the following basic performance curves can be established in function of the liquid rate:

- the well's IPR curve describing the pump intake pressure,
- the electrical submersible pump's pressure performance curve, calculated from its head versus rate performance, and
- the separator—well tubing subsystem's VLP (Vertical Lift Performance) curve, i.e., pressure versus rate at the tubing just above the pump.

Using the performance curves just defined, the inflow and outflow conditions at the two **solution nodes** are as follows:

At the pump intake	Inflow pressure	$p_i = \text{IPR}$
	Outflow pressure	$p_i = \text{VLP} - \text{ESP}$
At the pump discharge	Inflow pressure	$p_d = \text{IPR} + \text{ESP}$
	Outflow pressure	$p_d = \text{VLP}$

Schematic presentation of these performance curves is given in Fig. 8.19, where the heavy lines represent the conditions at the pump **discharge** and

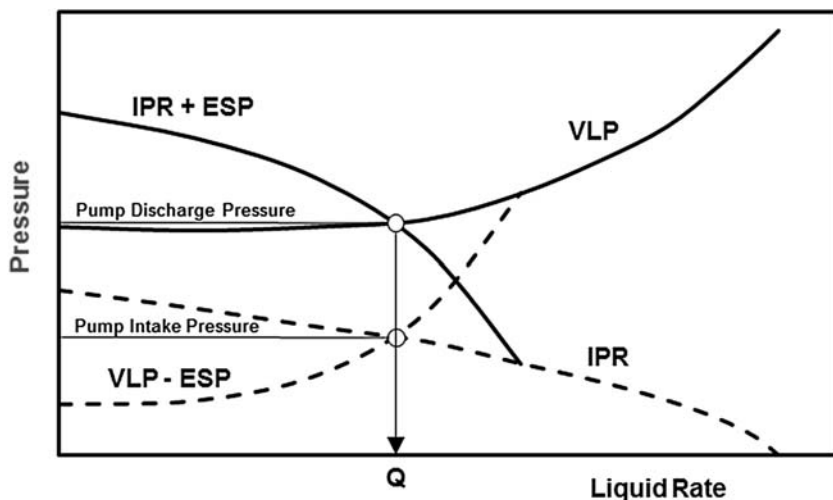


FIGURE 8.19 Nodal analysis using two solution nodes.

dashed lines represent those at the pump **intake**. The intersections of inflow and outflow curves define the actual intake and discharge pressures at the electrical submersible pump as well as the liquid rate, Q , that develops in the system. As seen, the well's liquid rate does **not** depend on the **placement** of the solution node; this is a basic feature of Nodal analysis.

The graphical approach just described is utilized in the troubleshooting model to **identify** the existence of the following basic **problems** in ESP operations:

- possible changes in inflow performance of the well,
- changes in the performance of the electrical submersible pump, or
- changes in tubing performance.

Of the possible scenarios, only the detection of the **deterioration** of electrical submersible pump's performance is detailed in the following. The performance curves for a new and an identical but worn-out electrical submersible pump are shown in Fig. 8.20, plotted in heavy and light lines, respectively. At each solution node one of the curves (IPR or VLP) does not change, only the curves containing the ESP performance are modified. The system's liquid rate using the worn-out pump has decreased because of the lower pressure developed by the pump. Looking at the changes in intake and discharge pressures, one can find that the discharge pressure has **decreased** and the intake pressure has **increased**. The conclusion is that if pump discharge and intake pressures are **monitored** and this behavior is observed, then the system's operation has degraded because of the electrical submersible pump's **deterioration**. It is important to note that knowledge of the well's liquid rate is not necessary to reach this conclusion.

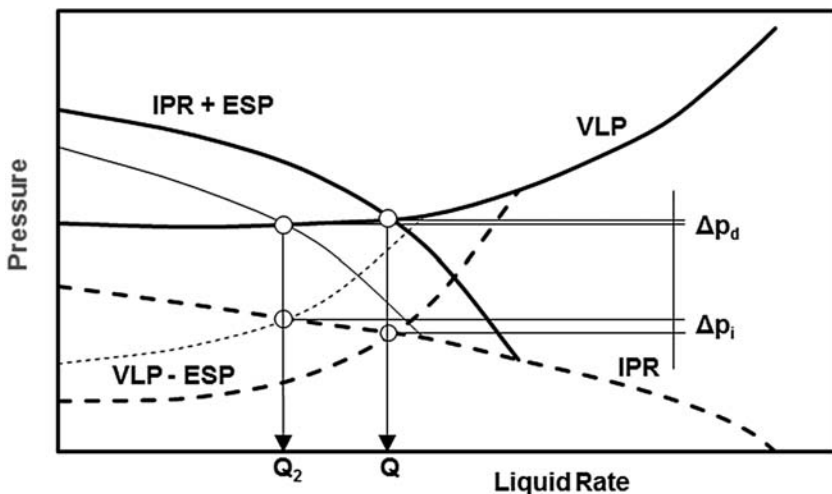


FIGURE 8.20 The effect of deteriorating electrical submersible pump performance on measured intake and discharge pressures.

TABLE 8.3 Decision Table Based on Changes in Pump Intake and Discharge Pressures [21]		
Change in Pump Discharge Pressure	Change in Pump Intake Pressure	
	Negative	Positive
Negative	$ \Delta p_i > \Delta p_d \rightarrow$ Reservoir performance <i>Deteriorated</i>	$\rightarrow$ ESP performance <i>Deteriorated</i>
	$ \Delta p_i < \Delta p_d \rightarrow$ Tubing performance <i>Improved</i>	
Positive	$\rightarrow$ ESP performance <i>Improved</i>	$ \Delta p_i > \Delta p_d \rightarrow$ Reservoir performance <i>Improved</i>
		$ \Delta p_i < \Delta p_d \rightarrow$ Tubing performance <i>Deteriorated</i>

ESP, electrical submersible pumping.

Performance curves similar to those in Fig. 8.20 were constructed by Bruijnen [21] for the remaining operational problems and his conclusions are summarized in Table 8.3. As shown, comparing the **changes** in measured pump intake and discharge pressures permits the detection of several operational problems. It can be found easily whether system changes are due to problems in the pump, in the reservoir, or in the tubing string.

The troubleshooting procedure just described was illustrated by the author through several example problems that proved its usability. The main advantages of the model are as follows:

- it is very simple, no complicated calculations are required to find the cause of changes in well liquid rate,
- no pump performance curves are needed, and
- knowledge of the well’s liquid rate is not needed for analysis.

Limitations include the following: (1) a packer should be installed in the well and (2) the electrical submersible pump’s head performance curve must be concave down (normally valid for most pumps).

Example 8.1

Downhole pressure gauges indicated considerable changes in intake and discharge pressures in a few weeks along with a decrease in the well’s liquid flow rate. Find the possible cause of this behavior if intake pressure decreased from 1,300 to 1,130 psi and discharge pressure decreased from 2,750 to 2,660 psi.

Solution

The changes of the pressures are $\Delta p_i = 1,130 - 1,300 = -170$ psi and $\Delta p_d = 2,660 - 2,750 = -90$ psi. Because both pressures have a negative change, and noting that $|\Delta p_i| > |\Delta p_d|$, the appropriate cell of Table 8.3 indicates that reservoir performance has deteriorated. Further investigations may detect whether reservoir pressure or inflow conditions have changed.

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Chapter 9

Special Installations

9.1 INTRODUCTION

In the previous chapters, mostly conventional electrical submersible pump (ESP) installations were discussed where the ESP system was run in the well at the bottom of the tubing string without a packer. One exception was the family of **shrouded** installations that helps overcome the negative effects on the ESP of free gas entering the wellbore. The majority of other special installations to be reviewed in this chapter can be classified into two big groups: (1) tubing deployed and (2) alternative deployed ESP systems. The first solution uses the well's production tubing string to run the ESP equipment in the well, whereas the other employs either the ESP cable, a coiled tubing (CT), or a wireline string for the same purpose. The horizontal pumping systems (HPSs) used on the surface to move high volumes of liquids are also discussed in this chapter.

In addition to those systems discussed in this chapter, there are even more specialized installations such as those combining ESP operations with other methods of artificial lifting in the same well. Treatment of the emerging technology of ESPPCP (progressing cavity pumping employing a downhole ESP motor drive) also falls outside the scope of this book.

9.2 TUBING DEPLOYED INSTALLATIONS

Tubing deployed installations still follow the original configuration of the ESP system and utilize the well's production tubing string to run and retrieve the necessary downhole equipment. Depending on the number of zones simultaneously or alternatively produced from the well, different installation types are available; the most commonly used solutions are detailed in the following sections.

9.2.1 Producing a Single Zone

9.2.1.1 *Shrouded and Horizontal Well Installations*

As discussed in Chapter 4, free-gas production is very detrimental to the operation of the centrifugal pump used in ESP systems. One way to overcome

the difficulties is to use “**Motor Shrouds**” that act as downhole gas separators and remove the bulk of free gas from the well fluid before it can reach the centrifugal pump. These installations are detailed in Section 4.4.4.1.2, please refer to Figs. 4.11–4.14.

In wells with a **horizontal** well section the following ESP installations are common: [1]

- The ESP unit is set just above the kickoff point in the vertical part of the well.
- The unit could also be run in the horizontal part of the well provided a large enough casing size is available and running into the horizontal section is possible.
- A shrouded installation with a dip tube or “tail pipe” as shown in Fig. 4.13 can also be utilized especially when free-gas production is an issue.

9.2.1.2 *Parallelly Connected Installations*

The use of two complete ESP systems in the same well in such a configuration that the two units work parallel to each other has several applications [2] as detailed in the following. Such **redundant** systems can substantially reduce workover costs, especially in the offshore environment because failure of one of the systems does not necessarily imply a workover. The two units are run simultaneously on initial well completion and may be of identical or different capacity; the latter case takes care of predicted changes in reservoir inflow. As shown in Fig. 9.1, the two ESP systems are connected to the tubing string with two “**Y-tools**,” which makes it possible to select between the two ESP systems at will.

- When operation from the lower unit is desired, a **blanking plug** is set in the lower Y-tool and an **isolation tool** is set in the upper Y-tool, see the left image in Fig. 9.1. To use the upper ESP system, the upper Y-tool is equipped with a blanking plug as well (see the image in the middle). In both cases the lower blanking plug is used to run any instrument or tool down to the well bottom.
- The parallel installation can also be used in wells where surface or downhole restrictions (small casing size, low allowed torque on shafts, etc.) severely restrict available flow rates. The **simultaneous** operation of the two units can significantly increase the production rate from the well for the same TDH (total dynamic head) value. An added advantage is that failure of one ESP unit does not result in total production loss because the other unit may still be producing. As shown in the right image in Fig. 9.1, removal of the upper blanking plug is the only operation needed if running of both pumps is desired.

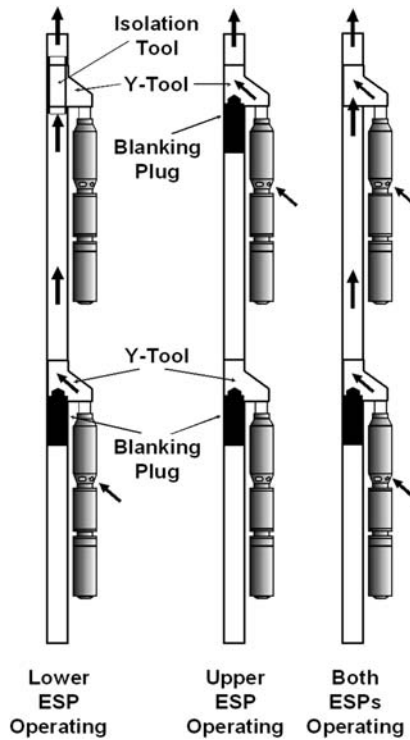


FIGURE 9.1 Parallel installation of two ESP units.

9.2.1.3 Series-Connected Installations

Two ESP units connected in series provide a **booster-pump** setup and as compared to a single-unit installation can achieve the following:

- increase the lifting **depth** for the same production rate or
- increase the total liquid **rate** for the same TDH.

Such systems are usually deployed when

- no motor with the required **power** is available for the given casing size because of a great lifting depth and/or a high liquid rate,
- the well's inflow rate decreases **rapidly** and the capacity of the ESP unit must be decreased after an initial production period,
- **collapse** of the casing string is anticipated because of an extensive pressure drawdown required to produce the target liquid rate.

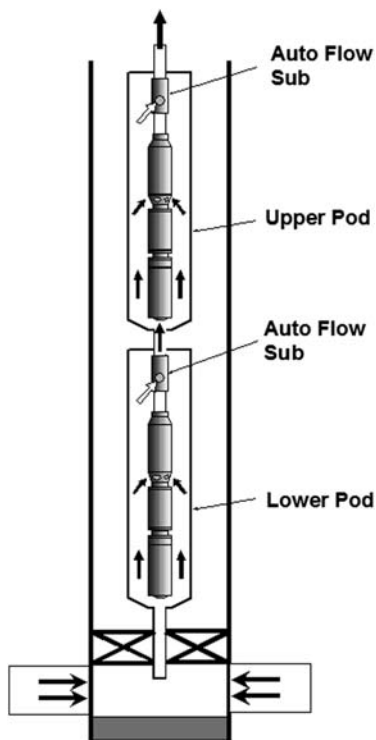


FIGURE 9.2 Series installation of two ESP units.

One version of the series-connected installation is shown in Fig. 9.2. The two ESP units are run above a **packer** that seals reservoir pressure from the casing annulus and prevents the collapse of the casing string should the required pressure drawdown be extensive. Both the ESP units are encapsulated in short sections of pipes called “**pods**” that are special capsules encasing and sealing the ESP units from the annulus. The pod system **protects** the casing string in harsh environments when the annulus can be filled up with completion fluid to prevent corrosion. The integrity of the casing string against overpressure, corrosion, or erosion is highly improved by using such pods.

As shown, on top of both the ESP units a so-called “**Auto-Flow Sub**” valve [3] is installed. These valves work fully automatically without any outside control, so no control lines to the surface are needed; they allow two operating modes:

1. As long as the ESP below the valve operates, the valve allows the produced fluids to freely pass through its open flapper valve upward and to enter the upper ESP unit or the tubing string.

2. As soon as the ESP unit is switched off and fluid flow from the pump ceases the flapper valve in the “**Auto-Flow Sub**” closes the tubing and opens a bypass path from the pod’s annulus, as shown by the hollow arrows in Fig. 9.2. Then the well fluids simply bypass the ESP unit and rise to the upper ESP or to the tubing string.

The operation of the “**Auto-Flow Sub**” valves permits the following operating modes of the series-connected ESP installation:

- When both the units are running, the advantages of a booster-pump configuration can be fully utilized: (1) the same liquid rate can be produced from deeper wells or (2) the production rate from the same depth can be **increased**.
- By switching off any one of the ESP units or if one of them fails, production can still continue. This feature can decrease production losses associated with unit failures and also allows the operator to change the well’s liquid rate by switching to the proper ESP unit.
- Wells otherwise able to sustain flowing production will continue to produce their normal rates with both ESP units shut down due to failure.

The “**Auto-Flow Sub**” valve’s advantages include a low resistance to flow in its bypass mode. This effectively reduces pressure losses and allows almost **unrestricted natural** flow, as compared to the case when well fluids pass through an inoperative ESP. The valve also reduces damage due to sand settling out from the fluid column in the tubing when the ESP is switched off because it diverts backflowing fluid from the tubing into the pod’s annulus. It also prevents pulling a wet tubing string for the same reason.

9.2.2 Dual Zone Installations

9.2.2.1 Production Commingling

Although the recommended technique of producing two zones from the same well is the utilization of two tubing strings, **commingling** of well fluids, i.e., production through a common tubing string, can also be employed.

The installation shown in Fig. 9.3, after Narvaez [4], utilizes one ESP unit to produce two zones from the same well. The two formations are isolated by a packer and a surface-controlled **isolation valve** set in the tubing string just above the packer. A perforated pipe section allows for commingling of the well fluids from the two formations and the ESP unit is encapsulated in a **pod**. Commingled production of the two formations rises to the surface in a single tubing string.

Closing the isolation valve permits to conduct a production test on the upper formation and an estimation of the lower formation’s capacity. The basic **disadvantage** of this installation type is that individual fine-tuning and optimization of each zone’s production is difficult, if not even impossible.

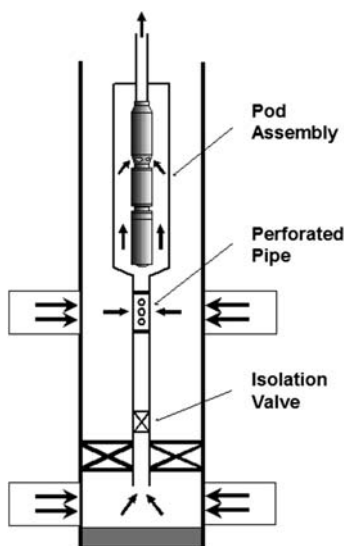


FIGURE 9.3 Commingled production of two zones with one ESP unit.

The dual installation introduced in Fig. 9.4 [5] can be used to solve two different problems:

1. the alternating production of the individual zones and
2. the simultaneous commingled production of two zones.

It utilizes two ESP units operating entirely separated from each other and having different characteristic features. The lower ESP unit is attached to the bottom of the tubing string and is encapsulated in a **pod**; it pumps fluids coming from the lower formation. The upper unit is fixed to a Y-tool and transfers the production of the upper zone into the common tubing string.

For producing the lower zone, an isolation tool (see Fig. 9.1) is inserted in the Y-tool and the upper ESP is switched off. Alternatively, running a blanking plug into the **Y-tool** allows the production of the upper zone.

If used for the simultaneous **commingled** production of the two zones, this installation has the following advantages over the solution previously described:

- The use of two ESPs allows the selection of the equipment according to the individual features of the zones.
- Proper well testing of the two zones is possible because the zones can be produced isolated from each other, as described previously.

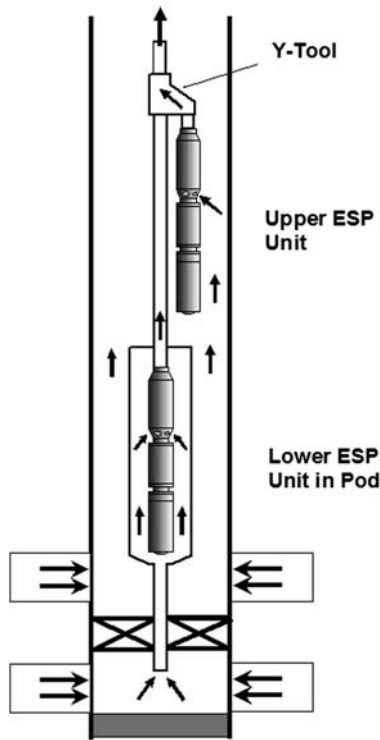


FIGURE 9.4 Commingled production of two zones with two ESPs.

9.2.2.2 Selective Production

The proper way to produce two zones from a single well is to completely **isolate** their operation; this normally requires the use of two tubing strings. Well fluids coming from the two formations are thus **not commingled** and their flow rates can properly be measured and accounted, a basic requirement for reservoir management. Both zones have their own ESP units, which can be properly **designed** and **optimized** to meet production requirements.

A common solution for selective production is shown in Fig. 9.5 where two packers are used to isolate the productive formations. The lower zone is produced through the annulus, a solution only allowed if well fluids do not damage the casing string. The upper zone is produced through the short **tubing** string. A very similar version utilizes two full tubing strings, with the long string producing the lower zone.

In addition to these installation types, solutions using concentric tubing strings and combinations of tubing and CT strings are also known [4].

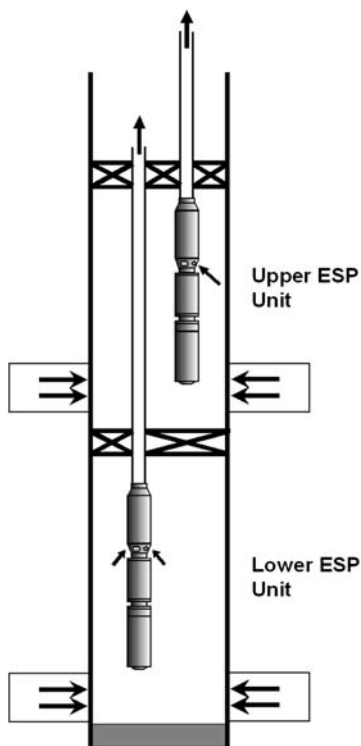


FIGURE 9.5 Selective production of two zones.

9.3 ALTERNATIVE DEPLOYED INSTALLATIONS

The tubing deployed installations all have the common drawback that running and retrieving the ESP equipment involves manipulation of the heavy tubing string. This can only be done with a drilling rig or a relatively high-capacity **workover** unit. As these are not always available and offer their services at elevated prices, workovers on conventional installations severely increase the costs of ESP operations. The number and **frequency** of workover operations are relatively high for ESP installations, as compared to other types of artificial lift. Following are the main problems necessitating interventions:

- The ESP units often fail in **infancy** because of different adverse effects such as pumping out kill fluid and fracturing sand.
- The first unit run in the well is usually **not optimal** because of insufficient reservoir data, or poor original design.
- Frequent equipment **failures** are to be expected.
- During the life of the system, well conditions may **change** and the ESP system may turn out to be inefficient.

All these typical features of ESP operations inherently lead to frequent equipment changes and the associated high production deferments; both increase production costs. This is the reason why decreasing the workover costs is of prime importance in any field; production of high-productivity offshore wells is especially sensitive to such effects.

The industry has long recognized these facts and tried to reduce the number and frequency of well interventions to ESP wells. The earliest solution was running and retrieving the ESP unit on its electrical cable, later a CT string was used. Today there are several different models available that partly or completely eliminate the use of a heavy tubing string and require workover rigs of much smaller load capacity and service fee to reduce operational costs. The following sections describe present-day solutions; an excellent historical overview on alternate system deployments is found in [6].

9.3.1 Cable Suspended Units

ESP units running on their electrical cables were introduced in the early 1970s. The solution patented by Arutunoff [7] is shown in Fig. 9.6. A conventional

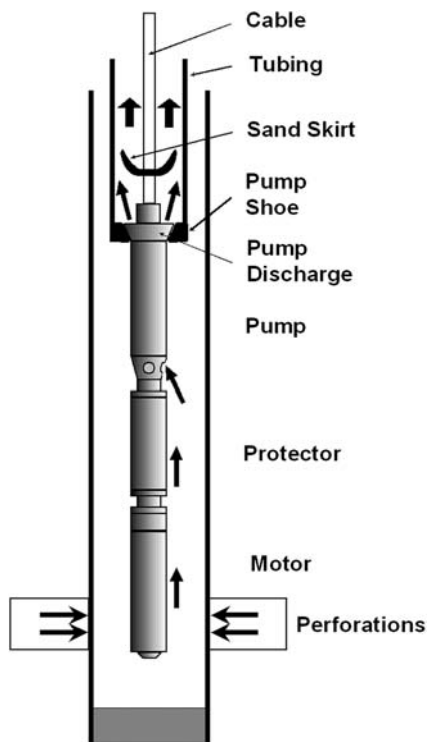


FIGURE 9.6 Arutunoff's cable suspended ESP installation.

ESP unit is run at the end of a special reinforced electrical cable capable of withstanding the stress arising from the weight of the unit. The ESP equipment is landed inside the tubing string at a pump **shoe** that carries the weight of the unit as well as provides the necessary seal between the suction and discharge sides of the ESP.

The ESP pump discharges well fluids into the tubing and production to the surface takes place in the tubing string. To prevent sand from accumulating above the discharge head during system shutdowns, flexible “**sand skirts**” are installed just above the cable connector.

The advantage of the cable-suspended ESP unit is that it does not require a **heavy** workover unit; the cable must not be banded so running and pulling operations are quicker. The main limitation is that high forces may develop in the cable during running and pulling operations and can lead to deterioration of its electrical properties. Such problems are less pronounced if a **dual-cable** system is used where a support cable carries the mechanical loads. Some successful case studies are described by **Caycedo and Diebold** [8].

9.3.2 Coiled Tubing Installations

The use of cable-suspended ESP units did not prove to be a breakthrough for several reasons:

- The special high-strength cables required special tools and handling.
- Lengthening of the cable (**splicing**) was more complicated because of the cable’s two functions (load and electricity carrying).
- Cables were **expensive** to manufacture.

A much better solution to the original problem, i.e., the lowering of the high workover costs associated with tubing installations, is the utilization of the CT technology. The first such system, installed in Oklahoma in 1992, employed a CT string instead of the conventional, jointed tubing string and effectively reduced the running/pulling, hence all workover costs [9,10].

9.3.2.1 Cable Led Outside the Coiled Tubing String

Originally the ESP cable was banded to the outside of the CT string by special cable clamps as shown in the left image in Fig. 9.7. This installation is identical to the **conventional** ESP installation with the motor at the bottom. The main difference is that instead of the jointed tubing string a continuous CT string is used to produce well fluids to the surface. This system required, instead of a heavy-duty workover rig, the use of a CT injector rig, resulting in shortened rig times and much lower operating costs. Running and pulling times are relatively low because of the elimination of tubing connections and the operating costs of CT rigs are also lower than those of workover units.

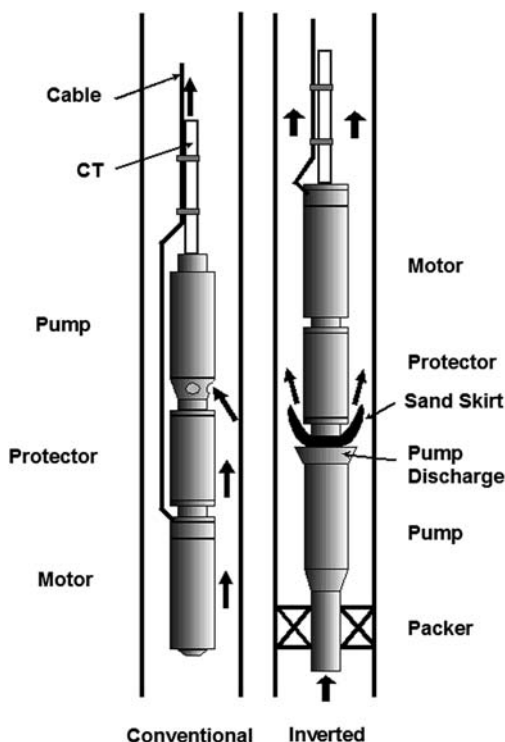


FIGURE 9.7 Coiled tubing (CT) installations using external cable.

Disadvantages of this installation type include the following:

- The smaller size of the CT string induces higher **frictional** pressure losses and the required motor power increases.
- The well's liquid production rate may be **limited** by the smaller size of the CT string, as compared to the full-size tubing string.
- Both the CT string and the ESP cable must be spooled during workovers.
- The well must be killed before running/pulling of the ESP system.
- Running/pulling speeds are limited because of the need to band the cable to the CT string.

In contrast to the conventional arrangement (motor at the bottom, pump at the top), **inverted** ESP units have the electrical motor at the top; see the right image in Fig. 9.7. This configuration eliminates the need for a **motor lead extension (MLE)**, i.e., a flat cable section running down to the pothead on the motor, the most vulnerable part of the ESP cable. The electrical cable can thus be connected directly to the ESP motor, a safe and more reliable solution.

The centrifugal pump in an inverted unit has a modified **discharge head** that directs well fluids into the **annulus** between the casing and the CT string.

The pump's suction and discharge sides must be isolated; this is usually accomplished by a retrievable packer. Note the **sand skirt** preventing sand buildup above the packer and around the ESP unit during system shutdowns.

9.3.2.2 Cable Led Inside the Coiled Tubing String

Technological development has led to the introduction of the integrated CT—cable system where the ESP cable, instead of being clamped to the outside of the CT string runs inside the CT string. The cross section of one major ESP manufacturer's integrated CT/ESP cable [11] is depicted in Fig. 9.8. The $2\frac{3}{8}$ " CT with a wall thickness of 0.204" contains, in addition to the standard ESP electrical cable, an instrument wire and three hydraulic lines, all installed at the manufacturing site. The hydraulic lines may be used for different purposes: to operate subsurface safety valves, for setting packers, etc. The self-supported power cable is held in place by friction, the space between the CT and the cable is filled up with an inhibited fluid. Available CT sizes are 2", $2\frac{3}{8}$ ", and $2\frac{5}{8}$ ".

To decrease friction losses and to increase the production capacity of the installation, the downhole systems shown in Fig. 9.9 employ **annular flow** to the surface. Additional advantageous features of the integrated CT—cable system are the following:

- The electrical cable is **protected** inside the CT string against mechanical and other damages that may occur downhole; its service life may increase.
- The **maximum** allowed running/pulling speeds of the CT string can be used because cable banding is not needed.
- The ESP unit can be run/retrieved **under pressure** because sealing on the CT string is possible. Expensive well-killing operations are thus eliminated.
- Conventional ESP equipment is used.

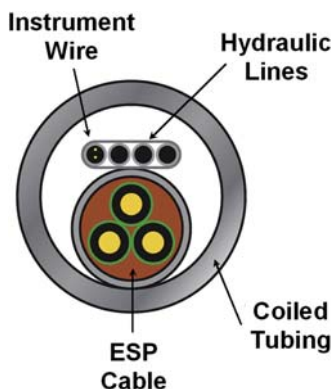


FIGURE 9.8 Integrated CT—ESP cable [11].

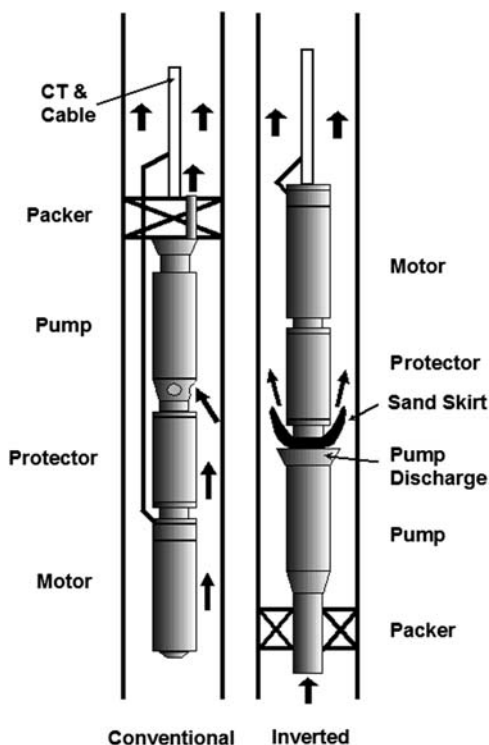


FIGURE 9.9 Coiled tubing (CT) installations using internal cable.

Disadvantages include the relatively high **weight** of the CT/cable system that may limit its application as well as the CT string's high corrosion susceptibility, being usually made from carbon steel.

In the **conventional** arrangement (left image in Fig. 9.9) the production packer contains a crossover and the fluid leaving the ESP is led into the casing-CT annulus. If the well produces some gas, free gas can collect below the packer and may cause well control problems. Using a **sliding sleeve** or a vent valve can eliminate this problem. If more gas is produced a centrifugal pump capable of handling larger volumes of gas must be used.

In **inverted** ESP installations the electrical motor is at the top of the system; see the right image in Fig. 9.9. As mentioned before, no MLE is needed and the cable is connected directly to the ESP motor. The centrifugal pump's modified **discharge head** directs well fluids into the **annulus**. A packer and a **sand skirt** are also installed.

Because of its many advantages the integral CT/cable system has gained momentum and several manufacturers have developed different sizes of CT with integrated ESP electrical cables [12–14]. The **inverted ESP—integrated**

CT installations provide an excellent solution for alternative deployed ESP applications. Their main uses are in **remote** and offshore areas where workover rig availability and production loss due to downtime are the main concerns. Running and retrieving of the ESP equipment requires the use of a standard CT surface unit instead of a full-size workover rig. Other benefits include

- significant reduction in operating **expenses** because of the decrease in workover costs,
- almost perfect elimination of electrical cable **failures** due to mechanical and chemical attacks,
- reduction of friction losses due to **annular** flow, important in high-rate and high-viscosity applications,
- reduced running and pulling time requirements because of the use of standard CT surface equipment.

9.3.3 Thru-Tubing Deployed Systems

Tubing or CT-conveyed ESP systems both require a **workover** rig for running and pulling the unit, which involves high intervention costs and possible production losses. Another limitation is the inability to **access** the reservoir without pulling the existing tubing string. In offshore areas and places where workover rig availability is limited, these factors are taken very seriously and eliminating the necessity of rig operations pays out enormously, especially in high-rate wells. The following sections present the available methods that partly eliminate the necessity of using high-cost workover rigs when running or pulling ESP equipment.

9.3.3.1 Retrievable ESP Pumps

In fields where high concentrations of sand or other abrasives are produced the centrifugal pump fails **first** while the other components may still be in good working order. In such cases, changing the pump inevitably means pulling the **entire** ESP system from the well, which involves high costs. Originally developed in the North Slope of Alaska by a major operator, the installation presented in [Fig. 9.10](#) allows the change of a defective pump by **wireline** operations.

The installation has two main components: (1) a **permanent** system installed at the bottom of the tubing string and (2) the **retrievable** assembly containing the centrifugal pump. The permanent assembly contains the ESP motor and the protector, both fixed to a special **crossover** situated at the bottom of the tubing string. The tubing crossover has two basic functions: it allows the well fluids to enter the tubing and provides the means to attach the ESP to the protector shaft. All these pieces, along with the electrical cable attached to the tubing string, are run in the well using a traditional workover rig before production can commence.

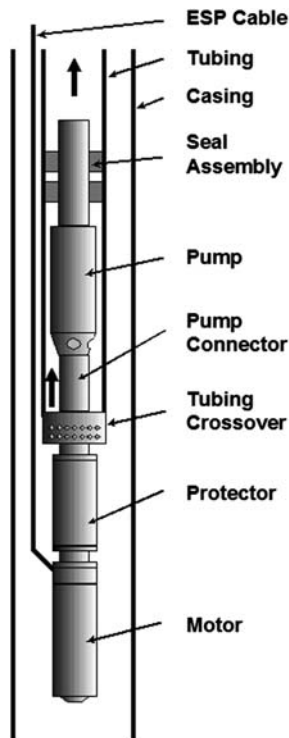


FIGURE 9.10 Thru-tubing installation using a retrievable pump.

The main part of the **retrievable** assembly is a slightly modified centrifugal pump that is run on wireline (slickline) to the tubing crossover and attaches to the protector's shaft. To separate the pump's suction and discharge sides, a **seal** assembly is run above the pump so well production takes place through the tubing string. Any time the pump needs to be changed, the use of only a wireline truck is needed; this normally involves only 5%–10% of the costs of a full-size workover unit's service fees [15].

The advantages of this installation type are manifold. If the permanent part of the system is installed, pulling and running the pump is an **easy** and cheap process. Finding the optimum pump size by frequent pump changes can also be accomplished. As the retrievable pump normally fits the regular lubricators, pump changes can be made on a **live** well, killing operations are not necessary. Limitations include the need for a heavy workover rig to run the permanent assembly in the well; any problems developing in the permanent part also require a rig's intervention. The permanently installed equipment makes it impossible to run any **tools** below the tubing string and formation treatment and other downhole operations are thus excluded.

9.3.3.2 Retrievable ESP Systems

The latest “**rigless**” ESP conveyance system discussed next also uses the time-proven and readily available wireline technology as those already discussed, with the difference that the **complete** ESP unit is retrievable, not only its pump [16,17]. The two basic components of the installation are (1) permanently installed parts such as the tubing string, the ESP cable, and a **docking station** and (2) the **retrievable** ESP components such as the motor, protector, and pump, see Fig. 9.11.

To accommodate standard ESP equipment the required tubing size is relatively big, normally a minimum of $4\frac{1}{2}$ " is used. The **docking station** is attached to the bottom of the production tubing string and its most important part is the **wet** electrical connector with its three **male** electrical contacts. The docking station also provides mechanical support to the ESP equipment and prevents any rotation of the motor. A lubricator valve provides reservoir access after removal of the ESP system from the well. These pieces of equipment are permanently installed in the well using a heavy workover rig.

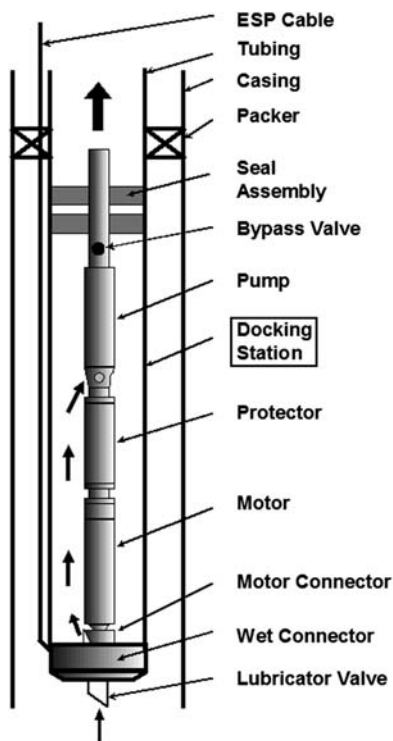


FIGURE 9.11 Main parts of a wireline retrievable ESP system.

The **retrievable** components are run on a standard wireline and are inserted through the lubricator attached to the wellhead. Pump, protector, motor, and all auxiliary equipment are run inside the tubing to the desired depth. At the bottom of the motor a connector is installed (see Fig. 9.12) that has three **female** wet connectors, an orienting sleeve, and a **throughbore** [18]. When reaching the docking station's bottom, the connectors of the motor and the docking station are mated and power is supplied to the motor. Just above the pump a **seal** assembly is installed to isolate the pump's suction and discharge sides. A bypass valve is also provided that is used when the system is retrieved.

After the **permanent** assembly is installed, running and pulling the complete ESP unit is an easy and quick job using a **wireline** truck, or a CT unit. A wireline running or pulling tool is used and after latching it onto the ESP unit, the unit can be run or retrieved through the tubing string in a few hours.

The advantages of the thru-tubing retrievable ESP system include the following, according to [19]:

- highly reduced well intervention costs,
- wireline services, in contrary to workover rigs, are readily available,
- running and pulling operations take only hours,
- production loss is minimized because of limited system downtimes,
- the ability to use any ESP equipment available,
- elimination of the pothead and the MLE that very frequently fail in conventional installations,
- full bore access to the formation is possible after retrieving the ESP unit,
- interventions on live wells are possible, no killing of the well is necessary.

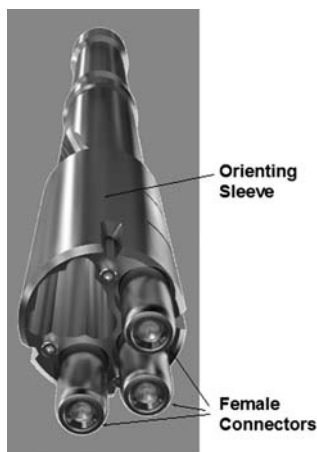


FIGURE 9.12 The ZEiTECS motor connector [18].

A recently developed system follows the same approach as described before with some important modifications [20,21]. To keep the total length and weight of the retrievable components in those ranges that wireline units can handle, permanent magnet motors (**PMMs**) are used. These motors are, as compared to conventional induction motors, lighter and can therefore be used with wireline units, and because they are shorter, they can be accommodated in standard lubricators. Their added benefits are greater power **efficiencies** and lower energy costs than those of induction motors; single PMMs can be used instead of tandem or triple induction motors. The other components of the ESP system [pump, protector, variable speed drive (VSD), etc.] can be selected from any major manufacturer's equipment. The complete retrievable system can also be run and pulled using a CT string; this extends the use of this technology to highly deviated wells where slickline operations are impossible.

The other modification is the way the motor is connected to the cable. For this a special **side-pocket mandrel** is installed in the tubing at the ESP unit's setting depth. The side-pocket mandrel contains a male **wet** electrical connector attached to the ESP cable's three conductors. The retrievable component has a female electrical connector that, while entering the side-pocket mandrel, is oriented and latched to the permanent male connector to provide electrical connection between the retrievable ESP assembly and the power cable. The permanent completion also provides mechanical support to the retrievable ESP system. When the ESP is retrieved the use of a side-pocket mandrel provides full-bore access through the well for conducting any remedial work at the formation.

The advantage of using PMMs is obvious, as their weight and length, according to one manufacturer's data [20], are about one fifth of that of induction motors, as given in Table 9.1. A typical wireline deployed ESP system offered by

TABLE 9.1 Comparison of Length and Weight of PMM and Induction Motors					
Size (in)	Power (HP)	PMM		Induction Motor	
		Length (ft)	Weight (lbs)	Length (ft)	Weight (lbs)
3.75	130	9.1	235	51.2	1,980
4.5	400	16.4	611	47.2	2,200
4.5	600	23.7	884	70.0	3,200
PMM, permanent magnet motor.					

the same manufacturer uses 4½" tubing string for 3.75" ESP equipment, with a maximum power of 380 HP that can produce about 6,000 bpd of liquid.

9.3.4 Conclusions

The broad overview just presented of the different ESP installations used in present-day practice clearly indicates the conventional ESP installation's biggest weakness, i.e., the tubing string must be pulled every time some part or the entire ESP system fails. The intervention costs are especially high in deep wells, in **offshore** conditions, and in fields with **frequent** equipment failures. The different ways to reduce these costs all aim at solutions that do not require the use of full-size workover rigs.

Some important features of the available ESP deployment systems are compared in Table 9.2. The table contains the type of flow (tubing or annular) made possible by the given arrangement as well as the type of **rig** needed to initially **install** and to **service** the installations. Finally, the need for **killing** the well during interventions is indicated.

The costs associated with each system have different structures and their capital expenditure (CAPEX) and operational expenditure (OPEX) changes differently. In principle, the initial investments (CAPEX) are similar for the first four systems and increase for the two wireline retrievable versions. On the other hand, operational costs (OPEX) are at a minimum for the wireline retrievable systems and are the highest for the conventional tubing deployed variant.

TABLE 9.2 Comparison of Different ESP Deployment Systems

Deployment	Flow Path	Initial Rig	Service Rig	Killing
Conventional	Tubing	Drilling/workover	Drilling/workover	Yes
Cable deployed	Annulus	Drilling/workover	Drawworks	Yes
CT Cable outside	CT	CT	CT	Yes
CT Cable integral	Annulus	CT	CT	No
Retrievable ESP Pump	Annulus	Drilling/workover	Slickline, CT	No
Retrievable ESP system	Annulus	Drilling/workover	Slickline, CT	No

CT, coiled tubing; ESP, electrical submersible pump.
Modified after [6].

9.4 HORIZONTAL PUMPING SYSTEMS

9.4.1 Available Configurations

The many beneficial features of the centrifugal pumps developed for ESP applications permit their use on the surface to move high volumes of liquids. The multistage centrifugal pumps providing high discharge pressures and high flow rates can be a good choice for a wide range of applications in the petroleum, mining, and other industries where the reliability of ESPs is highly valued. The HPSs offered by most major ESP manufacturers are mounted on heavy-duty skids and contain

- a horizontally mounted multistage centrifugal **pump** of a standard or special design,
- a **thrust chamber** (seal) to balance the axial forces arising in the pump,
- special **intake** and **discharge** manifolds to direct liquid flow to and from the pump,
- a **prime mover** that provides the energy for pumping.

According to the arrangement of the HPS components, there are three basic versions available.

- **Conventional direct drive system.** The system is driven by a low-maintenance electrical motor directly coupled to the pump shaft. The motor usually runs at 3,600 RPM but a VSD can also be used.
- **Belt-driven system.** The most flexible version of the HPS requires the least space for installation and can use practically any electrical motor mounted in the offset or side position. This feature greatly reduces investment costs, as the user can utilize its own motors. An additional benefit is that the system's overall length is at a minimum. At constant speed operations, the pumping rate is easily adjusted by changing the sheave size of the belt drive.
- **Gas- or diesel-driven system.** In remote locations with no electricity, this is the ideal arrangement where the pump is driven by internal combustion engines running on natural gas, propane, or diesel fuel. Variable speed operation is simply accomplished by regulating the speed of the engine; no expensive VSD unit is required. The pump can be belt-driven, or a gearbox may be used to increase the pumping speed.

Fig. 9.13 shows a schematic drawing of a conventional **HPS** unit where the main components of the system are indicated. All parts are mounted on a heavy-duty skid to form a complete system, and strong pump **supports** ensure vibration-free operations. Any submersible pump can be installed and coupled to the **thrust chamber** that takes up axial pump loads and provides radial stability. The prime mover (an electrical **motor**) is connected to the thrust chamber by a flexible coupling. Liquid is entering the pump through an integrated **intake** section and leaves at the **discharge** connected to the pump. The unit has proper safety features: excessive vibrations as well as high and low pressures are monitored.

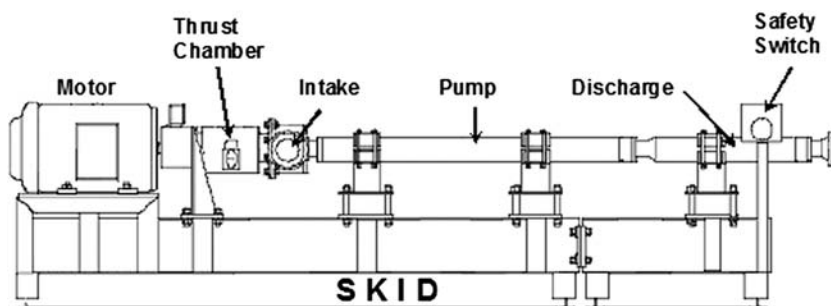


FIGURE 9.13 Main components of a conventional horizontal pumping system.

HPSs offered by different manufacturers [22,23] can pump liquid rates up to 40,000 bpd at discharge pressures up to 5,000 psi. The units usually have sophisticated control systems and several safety functions.

9.4.2 Typical Applications

HPS units provide a viable alternative to positive displacement (PD) and other pump types when transfer of high liquid rates is necessary at the surface. Although capital costs are comparable, maintenance and repair costs of PD pumps are much higher. HPS units excel in flexibility because the pumps are easily changed in their skids. All these features make them a prime choice for many applications in the oil field. Main applications, among others, include the following:

- disposal of produced water,
- pumping of water in waterflood projects,
- crude oil transfer,
- can be used as a pipeline booster pump,
- provides the required pressure for hydraulic fracturing of wells.

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Appendices

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Appendix A

DESCRIPTION

Fig. A.1 contains a diagram to estimate frictional head losses versus pumping rate in standard American Petroleum Institute (API) tubing and casing.

The heavy line displays values for new pipes; the other, for used ones.

EXAMPLE PROBLEM

Find the frictional head loss in a 4,500-ft-deep well with an old tubing of 1.995" inside diameter at a liquid flow rate of 1,000 bpd.

SOLUTION

At a 1,000 bpd rate, the specific head loss is read from the diagram as

$$\Delta h_{\text{fr}} = 41 \text{ psi/1,000 ft.}$$

The total head loss in the tubing string is thus

$$\Delta H_{\text{fr}} = 41 \times 4,500/1,000 = 185 \text{ psi.}$$

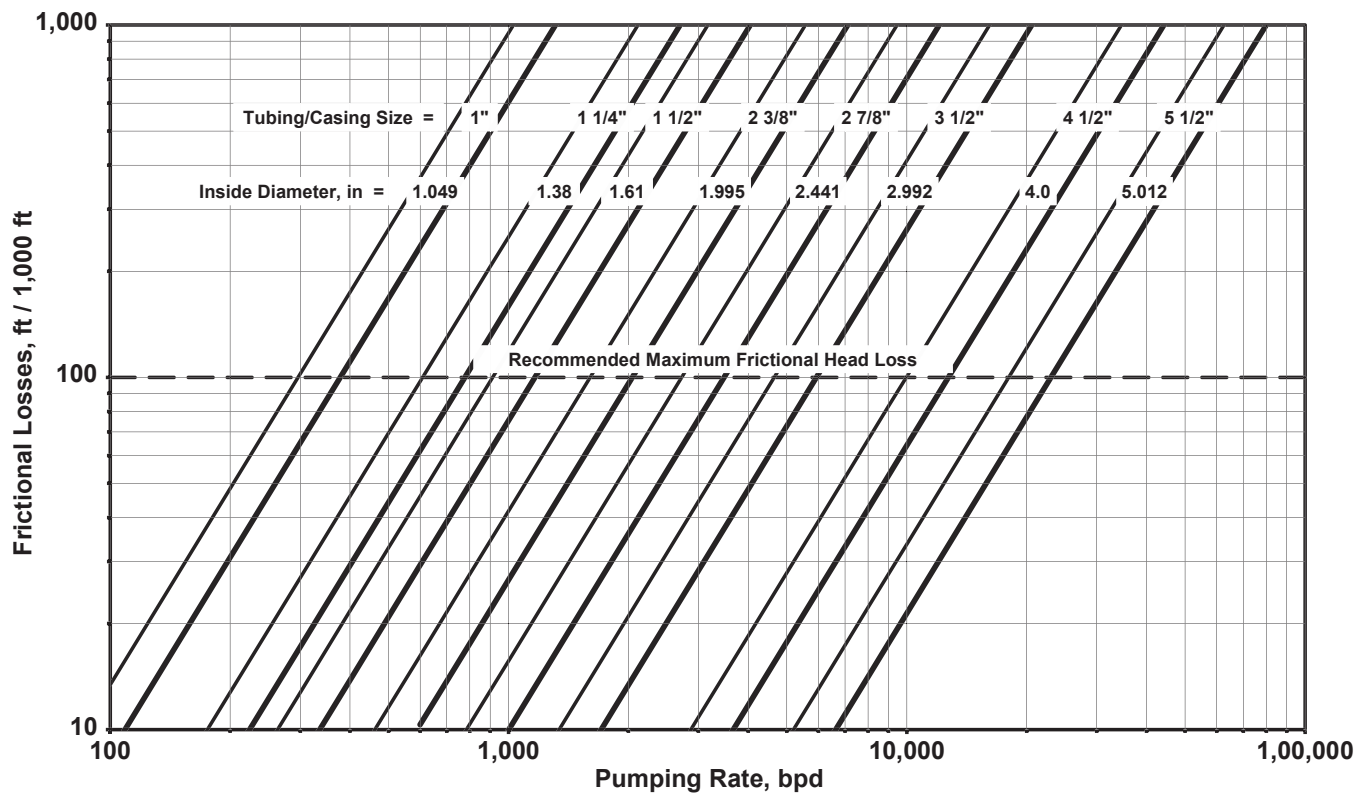


FIGURE A.1 Diagram to estimate frictional head losses versus pumping rate in standard American Petroleum Institute (API) tubing and casing.

Appendix B

DESCRIPTION

Table B.1 contains dimensional data of standard American Petroleum Institute (API) tubing and casing.

TABLE B.1 Main Dimensions of API Tubing and Casing						
Size (in)		Weight	Diameter (in)			Coupling
Nominal	OD	lb/ft	OD	ID	Drift	OD (in)
API NUE Tubing						
1½		2.75	1.900	1.610	1.516	2.200
2	2¾	4.00	2.375	2.041	1.947	2.875
2½	2⅞	6.40	2.875	2.441	2.347	3.500
3	3½	7.70	3.500	3.068	2.943	4.250
3½	4	9.50	4.000	3.548	3.423	4.750
4	4½	12.60	4.500	3.958	3.833	5.200
API EUE Tubing						
1½		2.90	1.900	1.610	1.516	2.500
2	2¾	4.70	2.375	1.995	1.901	3.063
2½	2⅞	6.50	2.875	2.441	2.347	3.668
3	3½	9.30	3.500	2.992	2.867	4.500
3½	4	11.00	4.000	3.476	3.351	5.000
4	4½	12.75	4.500	3.958	3.833	5.563

Continued

TABLE B.1 Main Dimensions of API Tubing and Casing—cont'd

Size (in)		Weight	Diameter (in)			Coupling
Nominal	OD	lb/ft	OD	ID	Drift	OD (in)
Regular API Casing						
	4½	9.50	4.500	4.090	3.965	5.000
		11.60	4.500	4.000	3.875	5.000
		13.50	4.500	3.920	3.795	5.000
	5½	14.00	5.500	5.012	4.887	6.050
		15.50	5.500	4.940	4.825	6.050
		17.00	5.500	4.892	4.767	6.050
		20.00	5.500	4.778	4.653	6.050
		23.00	5.500	4.670	4.545	6.050
	6⅝	17.00	6.625	6.135	6.010	7.390
		24.00	6.625	5.921	5.796	7.390
	7	20.00	7.000	6.456	6.331	7.656
		23.00	7.000	6.366	6.241	7.656
		26.00	7.000	6.276	6.151	7.656
		29.00	7.000	6.184	6.059	7.656
		32.00	7.000	6.094	5.969	7.656
	8⅝	28.00	8.625	8.017	7.892	9.625
		36.00	8.625	7.825	7.700	9.625
		40.00	8.625	7.725	7.600	9.625
		44.00	8.625	7.625	7.500	9.625
	9⅝	36.00	9.625	8.921	8.765	10.625
		40.00	9.625	8.835	8.679	10.625
		13.50	9.625	8.755	8.599	10.625
		47.00	9.625	8.681	8.525	10.625
	10¾	40.50	10.750	10.050	9.894	11.750
		55.50	10.750	9.760	9.604	11.750
	13⅜	48.00	13.375	12.715	12.559	14.375
		68.00	13.375	12.415	12.259	14.375

EUE, external upset end; *ID*, inside diameter; *NUE*, nonupset end; *OD*, outside diameter.

Appendix C

DESCRIPTION

Table C.1 contains dimensional data and recommended capacities for REDA submersible centrifugal pumps.

**TABLE C.1 Recommended Capacities of REDA Submersible Centrifugal Pumps
60 Hz @ 3,500 RPM**

Pump Series	Nominal Outer Diameter (in)	Minimum Casing Size (in)	Pump Designation	Recommended Liquid Capacity (bpd)	Shaft HP Limit		Shaft Diameter (in)
					Std (HP)	High Str. (HP)	
338	3.38	4½	A400	200–500	94	150	0.625
			AN550	400–700	94	150	0.625
			AN900	700–1,060	94	150	0.625
			A1200	670–1,380	78	125	0.625
			AN1200	800–1,650	94	150	0.625
			A1500	1,000–2,000	125	200	0.687
			AN1500	1,000–2,000	125	200	0.687
			A2700N	1,800–3,400	125	200	0.687
400	4	5½	D400	200–550	94	150	0.625
			D475N	200–625	94	150	0.625
			DN525	300–625	94	150	0.625
			D725N	350–925	94	150	0.625
			D950	600–1,150	125	200	0.688
			DN1000	600–1,350	125	200	0.688
			D1400N	960–1,700	125	200	0.688
			DN1750	1,200–2,050	125	200	0.688
			DN1800	1,200–2,400	125	200	0.688
			DN2150	1,300–2,600	125	200	0.688
			D2400N	1,500–3,200	162	245	0.688

Continued

**TABLE C.1 Recommended Capacities of REDA Submersible Centrifugal Pumps
60 Hz @ 3,500 RPM—cont'd**

Pump Series	Nominal Outer Diameter (in)	Minimum Casing Size (in)	Pump Designation	Recommended Liquid Capacity (bpd)	Shaft HP Limit		Shaft Diameter (in)
					Std (HP)	High Str. (HP)	
			DN3000	2,100–3,700	256	410	0.875
			DN3100	2,100–3,900	256	410	0.875
			D3500N	2,400–4,500	316	492	0.870
			D4300N	3,500–5,400	256	410	0.875
			D5800N	4,400–7,000	256	410	0.875
540	5.13	6 ⁵ / ₈	GN1600	1,000–2,150	256	410	0.875
			GN2100	1,650–2,700	256	410	0.875
			G2700	2,000–3,400	256	410	0.875
			GN3200	2,200–4,100	256	410	0.875
			GN4000	3,200–4,800	375	600	1.000
			GN5200	3,900–6,600	375	600	1.000
			GN5600	4,000–7,500	375	600	1.000
			GN7000	5,000–9,000	375	600	1.000
			GN10000	7,000–12,000	637	1,019	1.188
538	5.38	7	SN2600	1,600–3,200	256	410	0.875
			SN3600	2,400–4,600	256	410	0.875
			S5000N	4,000–5,400	375	600	1.000
			S6000N	3,500–7,800	463	720	1.000
			S8000N	3,500–10,500	463	720	1.000
			SN8500	6,000–11,000	375	600	1.000
562	5.63	7	HN13500	5,000–18,000	375	600	1.000
			H15500N	11,000–20,000	637	1,019	1.188
			HN21000	17,500–24,000	637	1,019	1.188
			H28000N	28,000–36,000	637	1,019	1.188
675	6.75	8 ⁵ / ₈	J7000N	4,500–9,000	637	1,019	1.188
			J12000N	8,000–18,500	637	1,019	1.188
			JN16000	12,800–19,500	637	1,019	1.188
			JN21000	16,000–25,000	637	1,019	1.188
725	7.25	9 ⁵ / ₈	L16000N	11,000–20,000	637	1,019	1.188
			L43000N	36,000–54,000	1,000	1,600	1.500

**TABLE C.1 Recommended Capacities of REDA Submersible Centrifugal Pumps
60 Hz @ 3,500 RPM—cont'd**

Pump Series	Nominal Outer Diameter (in)	Minimum Casing Size (in)	Pump Designation	Recommended Liquid Capacity (bpd)	Shaft HP Limit		Shaft Diameter (in)
					Std (HP)	High Str. (HP)	
862	8.63	10 ³ / ₄	M520A	12,000–24,000	637	1,019	1.188
			M520B	12,000–23,000	637	1,019	1.188
			M520C	12,000–22,000	637	1,019	1.188
			M675A	19,000–32,500	637	1,019	1.188
			M675B	19,000–29,000	637	1,019	1.188
			M675C	19,000–28,000	637	1,019	1.188
1,000	10.00	11 ³ / ₄	N1050	35,000–64,000	1,500	2,400	1.188
			N1400NA	35,000–64,000	1,500	2,400	1.750
			N1400NB	35,000–60,000	1,500	2,400	1.750

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Appendix D

DESCRIPTION

Table D.1 contains operational data of selected REDA electric submersible motors operating at 60 Hz and 3,500 RPM.

TABLE D.1 Operational Data of Selected REDA Electric Submersible Motors Operating at 60 Hz and 3,500 RPM

Motor Series	Nominal Outer Diameter (in)	Motor Type	Motor Power (HP)	Voltage (V)	Amperage (A)
456	4.56	Dominator	24	439	35.0
				682	22.5
			36	415	55.5
				780	29.5
				901	25.5
			48	472	65.0
				877	35.0
				1,039	29.5
				1,363	22.5
			50	894	43.0
				995	38.5
				1,097	35.0
				1,198	32.0
				1,400	27.5
			72	951	48.6
				1,437	32.0
				2,288	20.0

Continued

TABLE D.1 Operational Data of Selected REDA Electric Submersible Motors Operating at 60 Hz and 3,500 RPM—cont'd					
Motor Series	Nominal Outer Diameter (in)	Motor Type	Motor Power (HP)	Voltage (V)	Amperage (A)
			84	968	55.5
				1,252	43.0
				1,394	38.5
				2,102	25.5
			96	945	65.0
				1,430	43.0
				2,402	25.5
			108	881	78.5
				1,427	48.5
				2,520	27.5
			120	1,181	65.0
				1,586	48.5
				2,194	35.0
				2,598	29.5
			132	1,076	78.5
				1,299	65.0
				2,413	35.0
				2,635	32.0
			144	931	99.0
				1,174	78.5
				2,145	43.0
				2,389	38.5
				2,631	35.0
			156	1,272	78.5
				1,535	65.0
				2,588	38.5

TABLE D.1 Operational Data of Selected REDA Electric Submersible Motors Operating at 60 Hz and 3,500 RPM—cont'd

Motor Series	Nominal Outer Diameter (in)	Motor Type	Motor Power (HP)	Voltage (V)	Amperage (A)
			168	1,086	99.0
				2,503	43.0
			180	2,682	43.0
			192	1,890	65.0
				2,537	48.5
			204	2,008	65.0
				2,695	48.5
			216	1,397	99.0
				1,762	78.5
				2,125	65.0
				2,490	55.5
562	5.62	Dominator	30	460	39.5
				745	24.5
			60	495	73.5
				850	43.0
				1,205	30.0
			90	850	64.0
				1,275	43.0
				1,380	39.5
				1,490	36.5
				2,125	35.5
			120	850	85.5
				990	73.5
				1,275	57.0
				1,415	51.5
				2,550	28.5
				4,160	17.0

Continued

TABLE D.1 Operational Data of Selected REDA Electric Submersible Motors Operating at 60 Hz and 3,500 RPM—cont'd					
Motor Series	Nominal Outer Diameter (in)	Motor Type	Motor Power (HP)	Voltage (V)	Amperage (A)
			150	885	102.5
				1,240	73.5
				1,415	64.0
				2,655	34.0
				3,190	28.5
				4,160	21.5
			180	1,275	85.5
				2,550	43.0
				3,190	34.0
			210	1,240	102.5
				2,490	51.5
				3,720	34.0
			240	1,415	102.5
				2,555	57.0
				4,250	34.0
			270	1,275	128.5
				1,595	102.5
				2,545	64.0
			300	1,235	147.5
				1,415	128.5
				1,775	102.5
				2,480	73.5
				4,250	43.0
			330	1,360	147.5
				1,560	128.5
				1,950	102.5
				2,725	73.5
				3,505	57.0
				3,895	51.5

TABLE D.1 Operational Data of Selected REDA Electric Submersible Motors Operating at 60 Hz and 3,500 RPM—cont'd

Motor Series	Nominal Outer Diameter (in)	Motor Type	Motor Power (HP)	Voltage (V)	Amperage (A)
			360	1,700	129.0
				2,125	102.5
				2,550	85.5
				3,395	64.0
				4,250	51.5
			390	1,385	171.0
				1,840	129.0
				2,080	114.0
				2,300	102.5
				2,765	85.5
				3,685	64.0
			420	2,480	102.5
				2,975	85.5
				3,470	73.5
				3,965	64.0
			450	1,860	147.0
				2,125	128.0
				2,655	102.5
				3,720	73.5
				4,250	64.0

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Appendix E

DESCRIPTION

Tables E.1 and E.2 contain operational data of selected permanent magnet (PM) motors from Borets International Ltd. (Artificial lift technologies catalog. Moscow, Russia: Borets International Ltd; 2015.)

Table E.3 contains operational data of selected PM motors from Novomet. (NOVOMET product catalog. Your ESP encyclopedia. Perm, Russia: Novomet; 2016.)

TABLE E.1 Borets Permanent Magnet Motors, 456 Series 4-Pole Motors, Operating Range 500–3,900 RPM, All Data at 60-Hz (3,600 RPM) Operation

Motor Power (HP)	Required Voltage (V)	Motor Current (A)	Motor Length (ft)	Motor Weight (lb)
40	672	35	5	238
60	1,022	35	6	298
80	1,356	35	8	359
100	1,698	35	9	419
120	2,028	35	10	481
140	2,370	35	12	540
160	2,167	43	13	600
180	2,541	44	14	662
200	2,028	58	15	721
220	2,264	58	17	783
240	2,424	54	18	842
280	2,830	57	20	961
320	2,180	80	26	1,083
340	2,286	79	24	1,138
360	2,384	88	25	1,197
400	2,646	88	28	1,319

TABLE E.2 Borets Permanent Magnet Motors, 512 Series 4-Pole Motors, Operating Range 500–3,900 RPM. All Data at 60-Hz (3,600 RPM) Operation				
Motor Power (HP)	Required Voltage (V)	Motor Current (A)	Motor Length (ft)	Motor Weight (lb)
80	1,370	44	6	331
120	2,019	44	8	423
160	2,283	51	9	520
200	2,846	50	11	617
240	2,679	60	13	712
280	3,111	60	14	809
320	3,531	60	16	904
360	3,176	76	18	999
400	3,511	75	19	1,094
440	2,897	100	21	1,189
480	3,157	101	23	1,283
520	3,405	101	24	1,378
560	3,665	101	26	1,473
600	3,926	100	28	1,568
640	2,791	152	29	1,608
680	2,948	151	31	1,645
720	3,106	151	33	1,735
760	3,251	150	34	1,824

TABLE E.3 Novomet Permanent Magnet Motors, 406 Series 4-Pole Motors

Rated Power @ 3,000 RPM (HP)	Driving Frequency (Hz)	Rot. Speed (RPM)	Motor Power (HP)	Required Voltage (V)	Motor Current (A)	Motor Efficiency (%)	Motor Length (ft)	Motor Weight (lb)
8	200	6,000	16	560	14.3	91.0	4.5	229
	183	5,500	15	520	14.2	91.0		
	167	5,000	13	480	14.2	91.0		
	150	4,500	12	430	14.1	91.9		
	133	4,000	11	390	14.0	90.7		
16	200	6,000	32	870	18.6	92.1	5.5	267
	183	5,500	29	800	18.6	92.0		
	167	5,000	27	730	18.5	91.9		
	150	4,500	24	650	18.4	91.7		
	133	4,000	21	600	18.4	91.4		
21	200	6,000	43	1,150	18.6	92.5	6.5	304
	183	5,500	40	1,060	18.5	92.4		
	167	5,000	36	970	18.5	92.3		
	150	4,500	32	880	18.5	92.2		
	133	4,000	29	790	18.4	91.9		

Continued

TABLE E.3 Novomet Permanent Magnet Motors, 406 Series 4-Pole Motors—cont'd

Rated Power @ 3,000 RPM (HP)	Driving Frequency (Hz)	Rot. Speed (RPM)	Motor Power (HP)	Required Voltage (V)	Motor Current (A)	Motor Efficiency (%)	Motor Length (ft)	Motor Weight (lb)
27	200	6,000	54	1,430	18.6	92.7	7.5	342
	183	5,500	50	1,320	18.5	92.7		
	167	5,000	46	1,210	18.5	92.6		
	150	4,500	40	1,090	18.4	92.4		
	133	4,000	36	980	18.3	92.2		
34	200	6,000	67	1,720	19.3	92.9	8.4	379
	183	5,500	62	1,590	19.2	92.9		
	167	5,000	56	1,450	19.2	92.8		
	150	4,500	51	1,320	19.1	92.6		
	133	4,000	46	1,180	19.1	92.3		
40	200	6,000	80	2,020	19.8	93.0	9.4	417
	183	5,500	74	1,860	19.8	93.0		
	167	5,000	67	1,700	19.7	92.9		
	150	4,500	60	1,540	19.6	92.7		
	133	4,000	54	1,380	19.6	92.4		

47	200	6,000	94	2,310	20.2	93.1	10.4	454
	183	5,500	87	2,130	20.1	93.1		
	167	5,000	79	1,950	20.1	93.0		
	150	4,500	71	1,760	20.0	92.8		
	133	4,000	63	1,580	20.0	92.5		
54	200	6,000	107	2,600	20.5	93.2	11.3	492
	183	5,500	99	2,400	20.5	93.1		
	167	5,000	90	2,190	20.4	93.0		
	150	4,500	80	1,990	20.3	92.8		
	133	4,000	72	1,780	20.3	92.5		
60	200	6,000	121	2,900	20.8	93.2	12.3	529
	183	5,500	111	2,670	20.7	93.2		
	167	5,000	101	2,440	20.6	93.1		
	150	4,500	91	2,210	20.6	92.9		
	133	4,000	80	1,980	20.5	92.6		
67	200	6,000	134	3,190	21.0	93.3	13.3	567
	183	5,500	123	2,940	20.9	93.2		
	167	5,000	113	2,690	20.8	93.1		
	150	4,500	98	2,430	20.8	92.9		
	133	4,000	90	2,180	20.7	92.6		

Continued

TABLE E.3 Novomet Permanent Magnet Motors, 406 Series 4-Pole Motors—cont'd

Rated Power @ 3,000 RPM (HP)	Driving Frequency (Hz)	Rot. Speed (RPM)	Motor Power (HP)	Required Voltage (V)	Motor Current (A)	Motor Efficiency (%)	Motor Length (ft)	Motor Weight (lb)
74	200	6,000	147	3,490	21.1	93.3	14.3	604
	183	5,500	135	3,210	21.1	93.3		
	167	5,000	123	2,930	21.0	93.2		
	150	4,500	111	2,660	20.9	93.0		
	133	4,000	99	2,380	20.9	92.6		
80	200	6,000	161	3,780	21.3	93.4	15.2	642
	183	5,500	147	3,480	21.2	93.3		
	167	5,000	134	3,180	21.1	93.2		
	150	4,500	121	2,880	21.1	93.0		
	133	4,000	107	2,580	21.0	92.7		
87	200	6,000	174	4,070	21.4	93.4	16.2	679
	183	5,500	161	3,750	21.3	93.3		
	167	5,000	146	3,430	21.3	93.2		
	150	4,500	131	3,110	21.2	93.0		
	133	4,000	117	2,780	21.1	92.7		

94	200	6,000	188	3,630	25.8	93.5	17.2	717
	183	5,500	173	3,340	25.8	93.4		
	167	5,000	157	3,050	25.7	93.3		
	150	4,500	141	2,770	25.6	93.1		
	133	4,000	126	2,480	25.5	92.8		
101	200	6,000	201	3,870	25.9	93.5	18.2	754
	183	5,500	185	3,560	25.9	93.5		
	167	5,000	168	3,260	25.8	93.4		
	150	4,500	151	2,950	25.7	93.2		
	133	4,000	134	2,640	25.7	92.9		
114	200	6,000	228	3,750	30.7	93.6	19.1	792
	183	5,500	209	3,450	30.6	93.5		
	167	5,000	190	3,150	30.5	93.3		
	150	4,500	172	2,860	30.5	93.1		
	133	4,000	153	2,560	30.4	92.8		

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Appendix F

DESCRIPTION

Fig. F.1 presents a widely used correlation to calculate the voltage drop in usual electrical submersible pump cables.

EXAMPLE PROBLEM

Find the three-phase voltage drop in a 5,000-ft-long AWG #2 size submersible cable with copper conductors if the motor current is 80 A and the average cable temperature is 200°F.

SOLUTION

At a current of 80 A and AWG #2 cable size the specific voltage drop is read from the chart as

$$\Delta V/1,000 \text{ ft} = 23 \text{ V}/1,000 \text{ ft}.$$

The total voltage drop at 77°F is found next:

$$\Delta V = 23 \times 5,000/1,000 = 115 \text{ V}.$$

The correction factor for the actual cable temperature of 200°F is found from the table on the chart as

$$\text{Correction} = 1.27.$$

The total voltage drop across the cable at the operating temperature is calculated as

$$\Delta V_{\text{corr}} = 1.27 \times 115 = 146 \text{ V}.$$

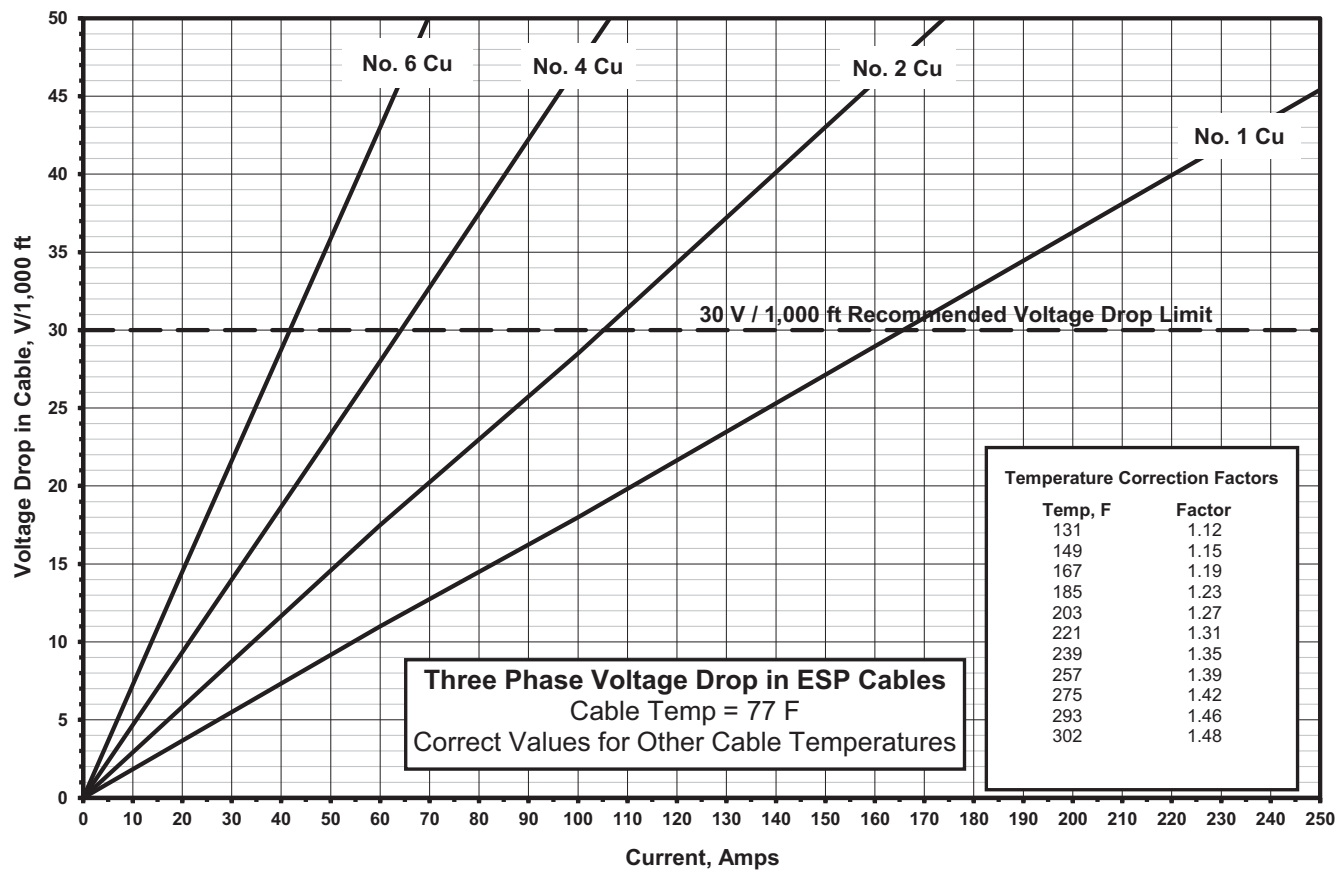


FIGURE F.1 Typical values of voltage drops in copper conductors. *ESP*, electrical submersible pump.

Class Problems

ESP Design and Analysis

Gabor Takacs, PhD
Instructor

CLASS PROBLEM ASSIGNMENT

Subject: Determination of a Well's Composite IPR Curve	Class Problem #1
Name of the Trainee:	Date:

PROBLEM STATEMENT

Develop the composite IPR curve for a well with the following parameters. The well was tested at 500 bpd at FBHP = 1,500 psi; its SBHP = 2,000 psi. The well produced with a water cut of 50%; the bubble-point pressure is 1,300 psi.

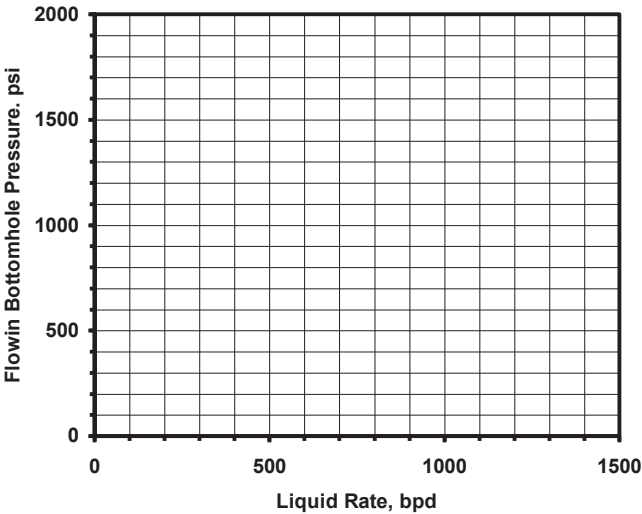
INSTRUCTIONS

First determine the relation between SBHP, FBHP, and bubble-point pressure and accordingly calculate the PI, the rate at bubble-point pressure, the maximum oil rate, the slope, and the well's maximum liquid rate.

Then calculate the FBHPs belonging to the rates given in the table, using Eqs. (2.4)–(2.6) in your Manual. Finally, plot the IPR curve on the chart given.

SOLUTION

Liquid Rate (bpd)	FBHP (psi)
0	2,000
	1,300
	0



ESP Design and Analysis**Gabor Takacs, PhD
Instructor****CLASS PROBLEM ASSIGNMENT**

Subject: Tubing Loss Calculations	Class Problem #2
Name of the Trainee:	Date:

PROBLEM STATEMENT

Find the head to be developed by an ESP pump operating at a depth of 7,000 ft and producing 1,500 bpd of specific gravity (SG) = 0.9 liquid. The liquid level is at the pump, the wellhead pressure equals 350 psi, and the tubing is $2\frac{3}{8}$ " (new pipe).

INSTRUCTIONS

First find the frictional head loss from Fig. A.1 in the Appendix, and calculate the total loss. Then find the head to overcome the wellhead pressure.

SOLUTION

CLASS PROBLEM ASSIGNMENT

Subject: Use of the Affinity Laws	Class Problem #3
Name of the Trainee:	Date:

PROBLEM STATEMENT

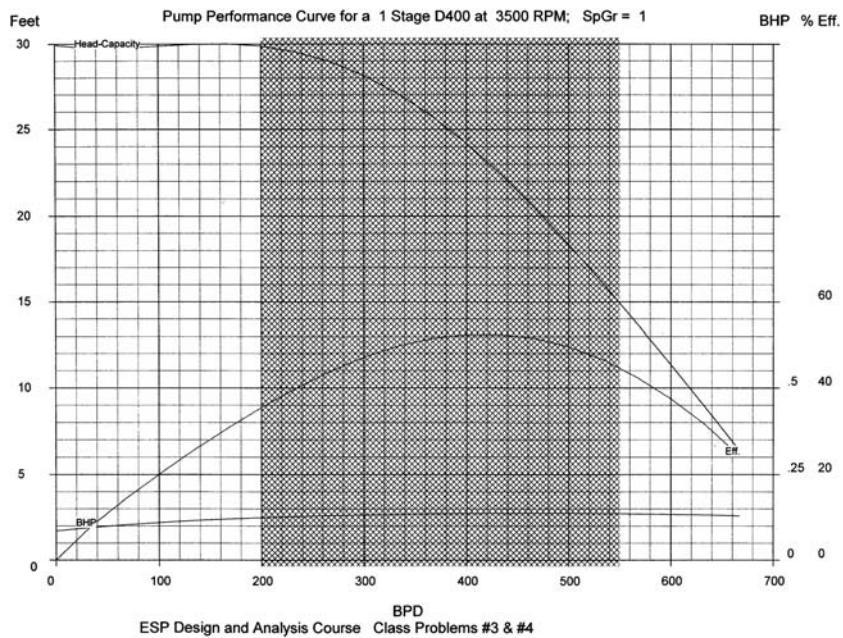
Based on the ESP pump performance curve (60 Hz, 3,500 RPM operation) attached, convert three points of the head, BHP, and efficiency curves to 50 Hz operation, where the pump’s speed is 2,917 RPM. Take the BEP point and the two limiting rates of the recommended operation range. Plot your results on the chart provided.

INSTRUCTIONS

Use the following table.

SOLUTION

	Read from Chart at 60 Hz				Calculated at 50 Hz			
	Rate (bpd)	Head (ft)	BHP (HP)	Efficiency (%)	Rate (bpd)	Head (ft)	BHP (HP)	Efficiency (%)
Minimum								
BEP								
Maximum								



CLASS PROBLEM ASSIGNMENT

Subject: Viscosity Correction of ESP Pump Performance	Class Problem #4
Name of the Trainee:	Date:

PROBLEM STATEMENT

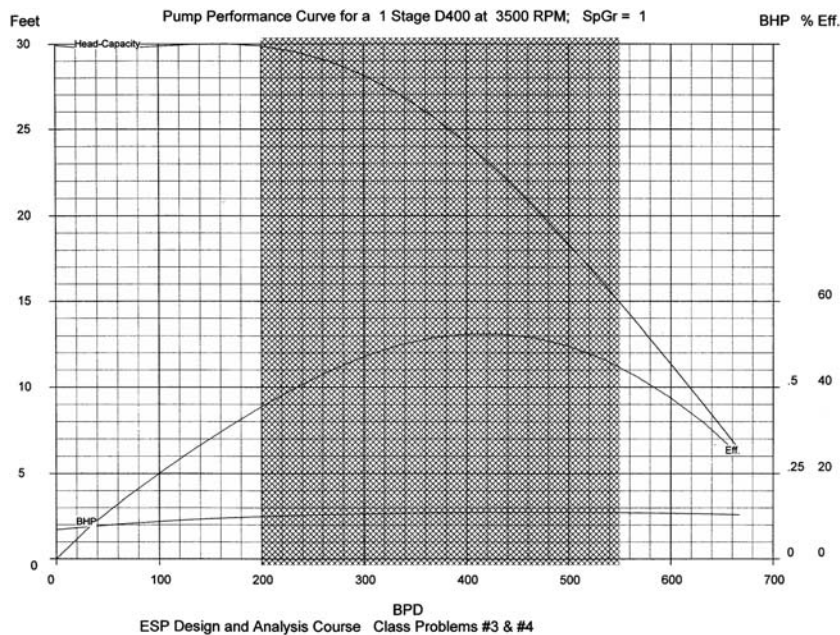
Correct the pump performance curves provided to a liquid of SG = 0.78 and a viscosity of 88 cSt. Use the BEP point and the two limiting rates of the recommended operation range. Plot your results on the chart provided.

INSTRUCTIONS

First convert the viscosity into SSU, and then find the correction factors recommended in Table 4.1.

SOLUTION

	Read From Chart				Corrected for Viscosity			
	Rate (bpd)	Head (ft)	BHP (HP)	Efficiency (%)	Rate (bpd)	Head (ft)	BHP (HP)	Efficiency (%)
Minimum								
BEP								
Maximum								



CLASS PROBLEM ASSIGNMENT

Subject: Free Gas Volume Calculation at Suction Conditions	Class Problem #5
Name of the Trainee:	Date:

PROBLEM STATEMENT

Using the **Turpin correlation**, find the severity of gas interference at the ESP pump’s suction for the following data.

- Oil API degree = 40

Gas SG = 0.6

Suction temperature = 130 F
- PIP = 500 psi

Oil rate = 1,500 bpd

Production GOR = 250 scf/bbl

WOR = 3

INSTRUCTIONS

First, find the solution gas-oil ratio, then the volume factor of the gas, and then the free gas volume.

The oil volume factor is calculated next, then the liquid volumetric rate, and finally the total fluid volume handled by the pump.

To find the severity of the gas interference, use the **Turpin correlation** as given in Eq. (4.30).

Oil SG = 141.5/(131.5 + API)

SOLUTION

CLASS PROBLEM ASSIGNMENT

Subject: ESP Motor Selection for VSD Service	Class Problem #6
Name of the Trainee:	Date:

PROBLEM STATEMENT

Select the proper ESP motor for 100-stage DN3000 pump running at 50 Hz and producing 2,400 bpd of a SG = 0.85 oil. A rotary gas separator is also used and takes 5 HP of power at 60 Hz operation. A one-stage performance curve sheet (60 Hz operation) is attached.

INSTRUCTIONS

First, find the rate corresponding to 2,400 bpd at 60 Hz operation, using the affinity law. Then read the performance parameters at this rate from the chart attached.

SOLUTION

$Q_{60} =$ (Eq. 4.32)
 From performance curves: $BHP_{60}/stage$ at $Q_{60} =$ $\times SG =$ $BHP/stage$
 From affinity law (Eq. 4.34): $BHP_{50}/stage =$
 BHP required for 100 stages at two frequencies:
 $BHP_{50} =$ $BHP_{60} =$
 Rotary separator's BHP requirement at 50 Hz (Eq. 4.34):
 BHP_{sep} at 50 Hz =
 Total required BHP at 50 Hz =
 Motor power at 60 Hz (Eq. 4.40) =

Motor selected from Table D.1:

NPHP =

Voltage =

Amps =

Check for maximum allowed frequency (Eq. 4.41):

$f_{\max}$ =

Calculated motor load (Eq. 4.43):

Load (%) =

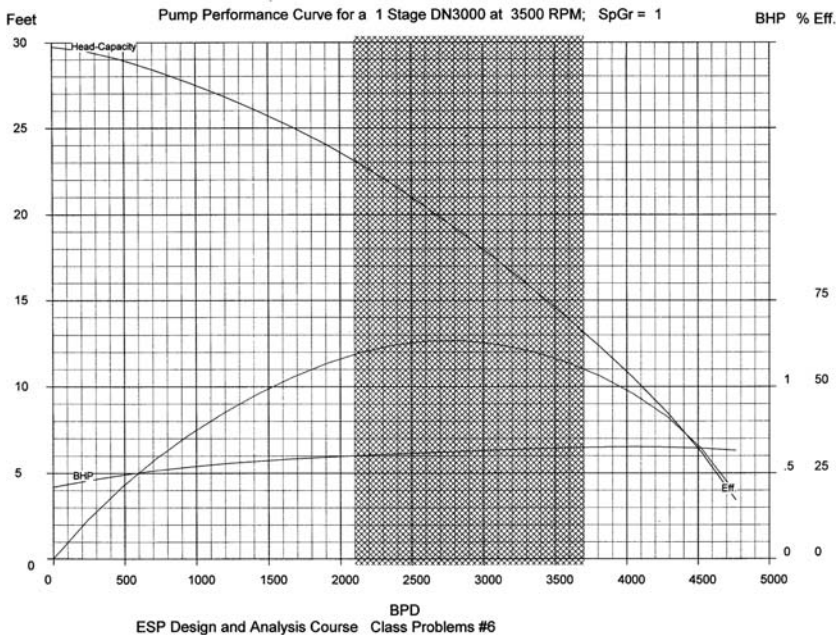
Check pump shaft strength:

Allowed power at 60 Hz from Table C.1 =

Converted to 50 Hz (Eq. 4.42) = Ok Overload

Calculate required VSD voltage (Eq. 4.39):

Voltage =



ESP Design and Analysis**Gabor Takacs, PhD
Instructor****CLASS PROBLEM ASSIGNMENT**

Subject: Determination of Total Dynamic Head	Class Problem #7
Name of the Trainee:	Date:

PROBLEM STATEMENT

The basic data for an ESP installation are as follows:

Pump setting depth = 5,000 ft	Depth of perforations = 6,000 ft
Desired pumping rate = 4,500 bpd	Liquid SG = 0.95
Tubing size = 3½" (old pipe)	Wellhead pressure = 200 psi

Calculate the TDH for three different cases:

- Case A:** The dynamic liquid level is at 4,500 ft
Case B: The static liquid level is at 1,000 ft
The well's PI is 3 bpd/psi
Case C: SBHP = 2,200 psi
PI = 2.8 bpd/psi

INSTRUCTIONS

First, calculate the dynamic liquid level, then use Fig. A.1 in Appendix A to find frictional losses.

SOLUTION

ESP Design and Analysis

Gabor Takacs, PhD
Instructor

CLASS PROBLEM ASSIGNMENT

Subject: Design of a Conventional ESP Installation	Class Problem #8
Name of the Trainee:	Date:

PROBLEM STATEMENT

Select the main components of an ESP installation running at 60 Hz in a well with negligible gas production.

Pump setting depth = 4,500 ft	WHP = 100 psi	SG of oil = 0.85
Perforations at 5,000 ft	CHP = 10 psi	SG of water = 1.0
Tubing size 2 3/8" new	Liquid rate = 1,700 STB/day	SG of gas = 0.60
Casing size 6 5/8"	Static liquid level = 1,820 ft	PI = 2 bpd/psi
Bottom-hole temperature = 200F	Water cut = 80%	Frequency = 60 Hz

INSTRUCTIONS

Follow the steps outlined on the following sheets.

SOLUTION

Inflow Calculations

- Calculate liquid SG from water cut, $SG_l =$
- Calculate SBHP = $0.433 (L_{perf} - L_{stat}) SG_l =$ psi
- Calculate FBHP = $SBHP - Q/PI =$ psi
- Calculate pump intake pressure (Eq. 5.2):
 PIP = $FBHP - 0.433 (L_{perf} - L_{pump}) SG_l =$ psi

Find dynamic liquid level from Eq. (5.6) (use $\text{grad}_g = 0$):

$$L_{\text{dyn}} = \quad \text{ft}$$

Find solution GOR from Eq. (5.3):

$$\text{Oil API} = 141.5/\text{SG Oil} - 131.5 =$$

$$y =$$

$$R_s = \quad \text{scf/STB}$$

Calculate oil volume factor at PIP from Eq. (5.5):

$$F =$$

$$B_o =$$

Calculate liquid rate at pump intake from Eq. (5.4):

$$Q = \quad \text{bpd}$$

Calculate TDH

Find frictional head loss from Fig. A.1: Δh_{fr} (ft/1,000 ft) =

$$\Delta H_{\text{fr}} = \Delta h_{\text{fr}} \times \text{Setting depth}/1,000 = \quad \text{ft}$$

$$h_0 = 2.31 \text{ WHP}/\text{SG} = \quad \text{ft}$$

$$\text{TDH} = L_{\text{dyn}} + \Delta H_{\text{fr}} + h_0 = \quad \text{ft}$$

Select Pump Type Using Table C.1

Series = Pump type =

Performance parameters at the desired rate from attached performance curve:

Head/stage	ft/stage
BHP/stage	HP/stage
Shut-in head/stage	ft/stage
Allowed shaft power	HP
Shaft diameter	In
Housing burst pressure	Psi

Calculate number of stages required

$$\text{Number of stages} = \text{TDH}/(\text{ft/stage}) = \quad \text{stages}$$

Pump housing # with stages (from the table attached)

Check Pump for Mechanical Strength

$$\text{Pump BHP} = \text{Number of stages} \times \text{BHP/stage} \times \text{SG}_1 = \quad \text{HP}$$

Shaft checks OK not OK

$$\text{Maximum, internal housing pressure} = \text{Shut-in head/stage} \times \text{Stages} \times 0.433 \times \text{SG}_1 =$$

$$= \quad \text{psi}$$

Housing checks OK not OK

Select Protector

Downthrust from pump = lbs (Eq. 5.14)

Series selected =

Type selected =

Load rating at BHT = lbs OK not OK

Shaft power rating = HP OK not OK

Select Motor

Motor series considered: Series _____

Check flow velocity around motor (Eq. 5.18)

v_1 = ft/s OK not OK

Required motor HP = Pump BHP = BHP

Select motor from Table D.1:

Power = HP

Voltage = V

Current = A

Motor loading = %

Actual motor current = A

Motor efficiency = %

Motor power factor =

Select Cable

Cable current = Actual motor amps = A

Cable length = Pump depth + 100 ft = ft

Cable type selected =

Select cable size in **Class Problem #19**

Selected cable size = AWG # _____

Calculate voltage drop in cable, V/1,000 ft

Drop read from graph = V/1,000 ft from Appendix F

Adjusted drop:

$V_{adj} = V_{graph} (1 + 0.00214 (BHT - 77)) =$

= V/1,000 ft

Voltage drop in cable: $V_{adj} \times \text{Cable length}/1,000 =$

= Volts

Find power lost in cable

Total resistance of cable

Ohms/1,000 ft for AWG #6 = _____

$$R_T = \text{Length}/1,000 \times (1 + 0.00214(\text{BHT} - 77)) = \text{Ohms}$$

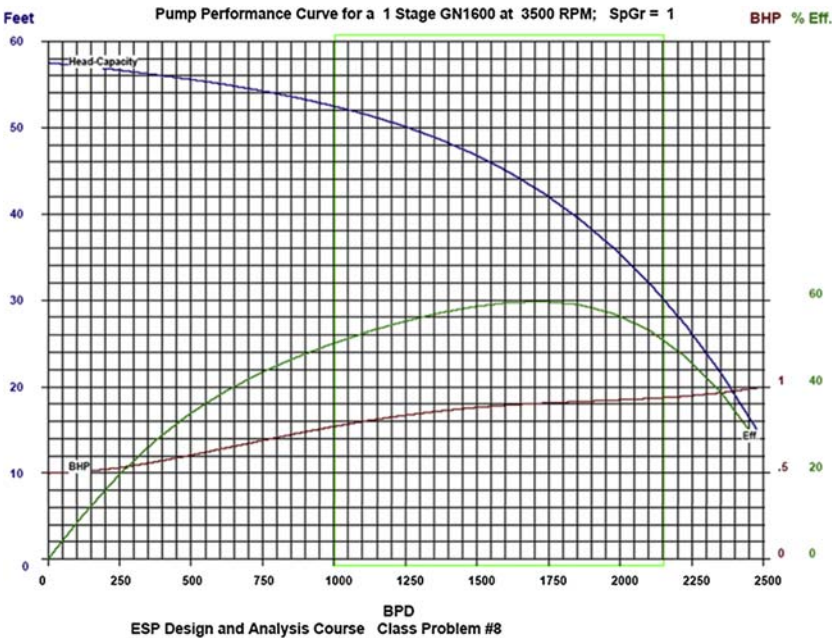
$$\text{Power loss} = 3 \text{ Current}^2 R_T / 1,000 = \text{ kW}$$

Calculate Surface Voltage, Power

Surface voltage = Motor voltage + Drop in cable = Volts

Surface kVA = $1.732 \times \text{Surface voltage} \times \text{Cable current}/1000 = \text{ kVA}$

Predicted power requirement = kVA $\times$ Power factor = kW



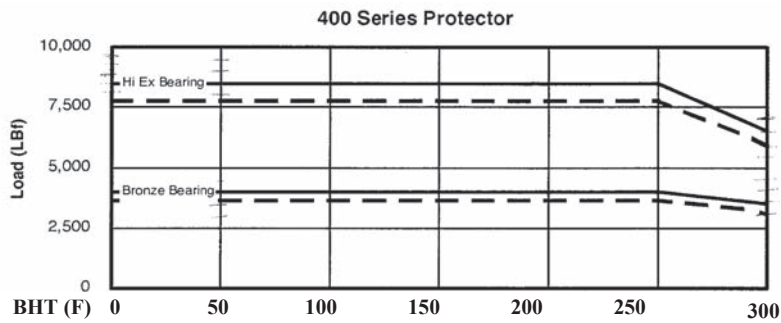
AVAILABLE HOUSINGS FOR GN1600 PUMPS

Housing #	Maximum Stages
10	12
20	28
30	43
40	59
50	75
60	90
70	106
80	122
90	137

ESP CABLE DIMENSIONAL AND PRICING DATA

AWG Size	OD (in)	Weight (lbs/ft)	Price (\$/ft)
1	1.357	1.69	11.30
2	1.280	1.44	9.69
4	1.091	1.00	6.46
6	1.000	0.75	5.11

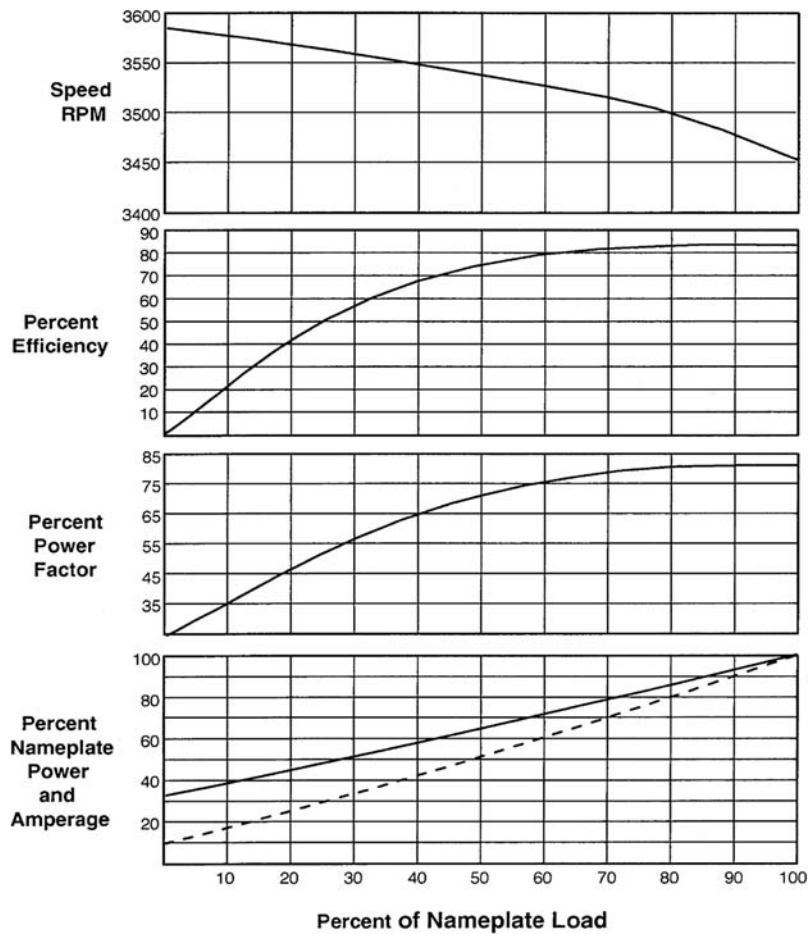
ESP PROTECTOR LOAD RATING DATA



ESP PROTECTOR SHAFT RATING DATA

Series	Shaft Diameter (in)	Rated HP
375	0.875	256
400	0.875	256
540	1.187	637
562	1.187	637

ESP MOTOR PERFORMANCE CURVES



CLASS PROBLEM ASSIGNMENT

Subject: Conventional Installation Design with Motor Slip	Class Problem #9
Name of the Trainee:	Date:

PROBLEM STATEMENT

Using the data given in **Class Problem #8**, select the components of an ESP installation by including the effects of the ESP motor’s slip.

INSTRUCTIONS

Most of the calculations are identical to those performed for solving **Class Problem #8** and must not be repeated here. The only calculation steps to be inserted after the motor selection are described in the following.

For your convenience, some of the calculated parameters to be used in solving this problem are culled here from **Class Problem #8**.

Liquid Rate at PIP	1,728 bpd	Pump BHP at 3,500 RPM	93 HP
Selected motor power	96 HP	Selected motor current	25.5 A
Calculated TDH	4351 ft	Number of pump stages	106

SOLUTION

Finding the Actual Motor Speed

To calculate the common speed of the ESP motor and the submersible pump, fill out the form until convergence is found.

Iteration Step	1	2	3	4	5	6
Pump BHP						
Motor loading, %						
Motor RPM						
Modified pump BHP						
BHP difference						

Converged results:

Actual motor speed, RPM =

Required motor HP =

Motor loading =

Find the following parameters from motor performance curves provided for **Class Problem #8** at the calculated motor loading:

Motor efficiency, %
Power factor
Percent amperage, %

Actual motor current from Eq. (5.33) =

Find the Pump Head Including the Effects of Motor Slip

Liquid rate at 3,500 SPM from Eq. (5.34) =

Head/stage read from performance curve at 3,500 RPM =

Total pump head from Eq. (5.35), H_{slip} =

Compare H_{slip} and TDH:

System can produce required rate YES NO

ESP Design and Analysis**Gabor Takacs, PhD
Instructor****CLASS PROBLEM ASSIGNMENT**

Subject: ESP Installation Design for a Gassy Well	Class Problem #10
Name of the Trainee:	Date:

PROBLEM STATEMENT

Select the main components of an ESP installation running at 60 Hz in a well with considerable gas production.

Pump setting depth = 5,500 ft	WHP = 100 psi	SG of oil = 0.86
Perforations at 6,000 ft	Liquid rate = 3,800 STB/day	SG of water = 1.09
Tubing size 3½" new	Static BHP = 2,000 psi	SG of gas = 0.65
Casing ID 6.366"	Water cut = 50%	PI = 2.5 bpd/psi
Bottom-hole temperature = 180F	Available frequency = 60 Hz	GOR _{prod} = 200 scf/bbl

INSTRUCTIONS

Follow the steps outlined on the following sheets.

SOLUTION**Inflow Calculations**

Find flowing BHP from PI equation (Eq. 5.1):

$$\text{FBHP} = \quad \text{psi}$$

Starting from this pressure, the PIP pressure is found from multiphase flow pressure drop calculations performed for $\text{GLR}_{\text{prod}} = \quad$; the result is:

$$\text{PIP} = \quad \text{psia}$$

Gas and Liquid “In Situ” Volume Calculations at Suction Conditions

Find solution GOR from Eq. (4.24):

$$\text{Oil API} = 141.5/\text{SG Oil} - 131.5 = \text{API degree}$$

$$y =$$

$$R_s = \text{scf/STB}$$

Calculate gas volume factor

Gas critical properties from Eqs. (4.27)–(4.28):

$$p_{pc} = \text{psia}$$

$$T_{pc} = \text{R}$$

Reduced parameters:

$$p_{pr} =$$

$$T_{pr} =$$

Deviation factor from Eq. (4.26):

$$Z =$$

Gas volume factor from Eq. (4.25):

$$B_g =$$

Free gas volumetric rate from Eq. (4.15):

$$q'_g = \text{cuft/day}$$

Find liquid volumetric rate

Calculate oil volume factor at PIP (Eq. 5.5)

$$F =$$

$$B_o =$$

Liquid rate from Eq. (4.22):

$$q'_l = \text{bpd}$$

Natural Gas Separation Calculations

Annulus cross-sectional area

$$A = 0.0055(ID_c^2 - OD_t^2) = \text{ft}^2$$

Liquid superficial velocity, from Eq. (4.17):

$$v_{sl} = \text{ft/s}$$

Density of gas phase, from Eq. (4.19):

$$\rho_g = \text{lb/ft}^3$$

Density of liquid phase, from Eq. (4.20):

$$\rho_l = \text{lb/ft}^3$$

Terminal rising velocity of gas bubbles Eq. (4.18):

$$v_b = \text{ft/s}$$

Natural gas separation efficiency, from Eq. (4.16):

$$\eta_n = \%$$

Amount of gas entering the pump, from Eq. (4.21):

$$q'_{\text{ing}} = \text{bpd}$$

Calculate **Turpin** Φ from Eq. (4.30):

$$\Phi = \text{OK} \quad \text{gas separation needed}$$

Rotary Gas Separator Calculations

Assumed efficiency of gas separator = $\%$

Gas ingested by the pump from Eq. (5.36):

$$q'_{\text{ing}} = \text{bpd}$$

“In situ” fluid volume the ESP pump must handle, from Eq. (5.37)

$$q'_{\text{fluid}} = \text{bpd}$$

The **Turpin** parameter Φ from Eq. (4.30):

$$\Phi = \text{ , operation of the pump is } \mathbf{Stable \ Unstable}$$

Calculate TDH

For vertical multiphase pressure drop calculations in the tubing string the gas-liquid ratio is found from Eq. (5.39):

$$\text{GLR} = \text{scf/STB}$$

The pump’s required discharge pressure with this value was found as:

$$p_d = \text{psi}$$

The required TDH from Eq. (5.38):

$$\text{TDH} = \text{ft}$$

Select Pump Type Using Table C.1

Series = Pump type =
 Performance parameters at the desired rate from attached performance curve:

Head/stage	ft/stage
BHP/stage	HP/stage
Shut-in head/stage	ft/stage
Allowed shaft power	HP
Shaft diameter	in
Housing burst pressure	psi

Calculate number of stages required
 Number of stages = $TDH/(ft/stage)$ = stages
 Pump housing # with stages (from the table attached)

Check Pump for Mechanical Strength

Find liquid-phase SG:

$SG_m = \rho_l/62.4 =$
 Pump BHP= Number of stages $\times$ BHP/stage $\times SG_m =$ HP
 Shaft checks **OK** **not OK**

Maximum, internal housing pressure = Shut-in head/stage $\times$
 Stages $\times 0.433 \times SG_m =$
 = psi
 Housing checks **OK** **not OK**

Select Protector

Downthrust from Pump (Eq. 5.14) = lbs
 Series selected =
 Type selected =
 Load rating at BHT = lbs **OK** not OK
 Shaft power rating = HP **OK** **not OK**

Select Motor

Motor series considered: Series _____

Check flow velocity around motor (Eq. 5.18):

$$v_1 = \quad \text{ft/s} \quad \quad \mathbf{OK} \quad \mathbf{not\ OK}$$

Power required by rotary gas separator = $\quad$ BHP

Required motor HP = Pump BHP + Separator BHP = $\quad$ BHP

Select Motor from Table D.1:

Power = $\quad$ HP

Voltage = $\quad$ V

Current = $\quad$ A

Motor loading = $\quad$ %

Actual motor current (Eq. 5.21) = $\quad$ A

Motor efficiency = $\quad$ %

Motor power factor = $\quad$

Select Cable

Cable current = Actual motor amps = $\quad$ A

Cable length = Pump depth + 100 ft = $\quad$ ft

Cable type selected = $\quad$

Select cable size in **Class Problem #19**

Selected cable size = AWG # $\quad$

Calculate voltage drop in cable, V/1,000 ft

Drop read from graph = $\quad$ V/1,000 ft $\quad$ Appendix F

Adjusted drop:

$$V_{\text{adj}} = V_{\text{graph}} (1 + 0.00214 (\text{BHT} - 77)) =$$

$$= \quad \text{V/1,000 ft}$$

$$\text{Voltage drop in cable: } V_{\text{adj}} \times \text{Cable length/1,000} =$$

$$= \quad \text{Volts}$$

Find power lost in cable

Total resistance of cable $\quad$ Ohms/ft = $\quad$

$$R_T = \text{Length/1,000 Ohms/ft} (1 + 0.00214 (\text{BHT} - 77)) =$$

$$= \quad \text{Ohms}$$

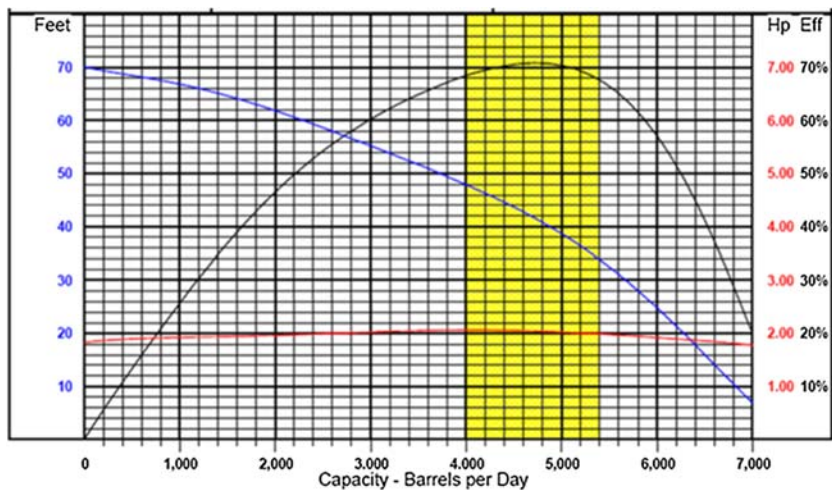
$$\text{Power loss} = 3 \text{ Current}^2 R_T / 1,000 = \quad \text{kW}$$

Calculate Surface Voltage, Power

Surface voltage = Motor voltage + Drop in cable = $\quad$ Volts

$$\text{Surface kVA} = 1.732 \times \text{Surface voltage} \times \text{Cable current/1,000} = \quad \text{kVA}$$

$$\text{Predicted power requirement} = \text{kVA} \times \text{Power factor} = \quad \text{kW}$$



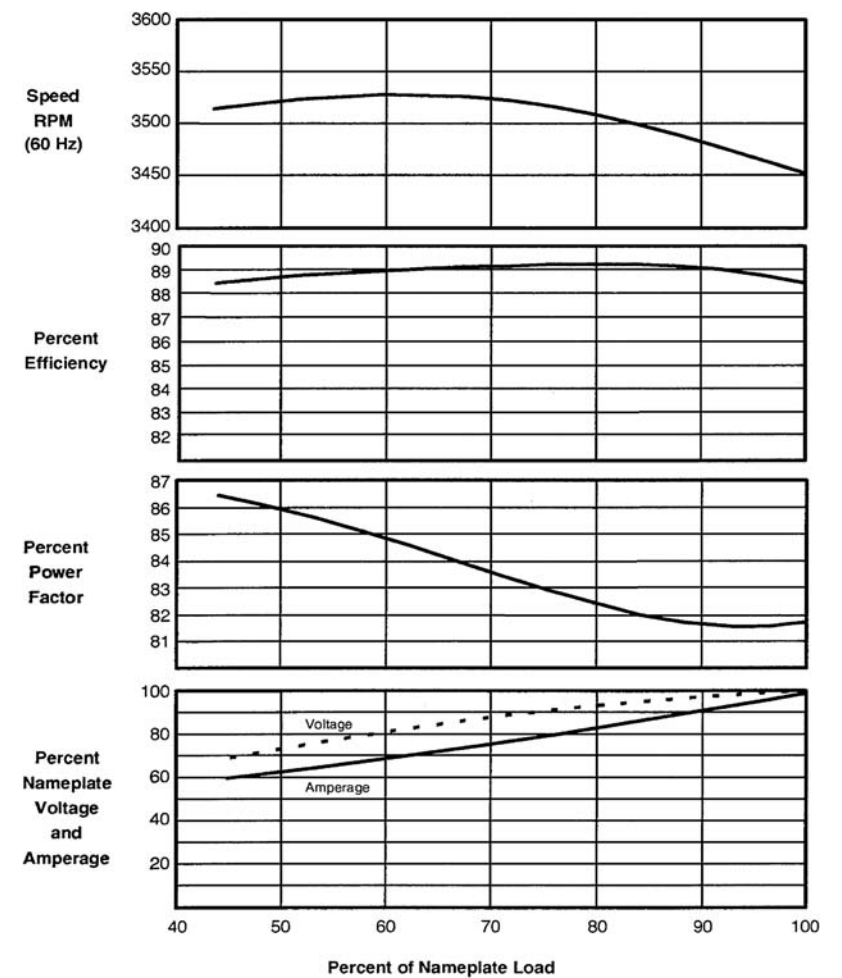
AVAILABLE HOUSINGS FOR S5000 N PUMPS

Housing #	Maximum Stages
10	5
20	11
30	18
40	25
50	31
60	38
70	44
80	51
90	57
100	64
110	71
120	77
130	84
140	90
150	97

ESP CABLE DIMENSIONAL AND PRICING DATA

AWG Size	OD (in)	Weight (lbs/ft)	Price (\$/ft)
1	1.357	1.69	11.30
2	1.280	1.44	9.69
4	1.091	1.00	6.46
6	1.000	0.75	5.11

ESP MOTOR PERFORMANCE CURVES



ESP Design and Analysis

Gabor Takacs, PhD
Instructor

CLASS PROBLEM ASSIGNMENT

Subject: Nodal Analysis of ESP Installations #1	Class Problem #11
Name of the Trainee:	Date:

PROBLEM STATEMENT

Determine the liquid production rate of an ESP system using a 100-stage **GN4000** pump. Well data are given as follows:

Pump setting depth = 5,500 ft	Static BHP = 3,000 psi	PI = 4 bpd/psi
Perforations at 6,000 ft	Water cut = 100%	
Tubing ID = 2.992" old	Available frequency = 60 Hz	

Assume wellhead pressures of 100, 200, and 300 psi.

INSTRUCTIONS

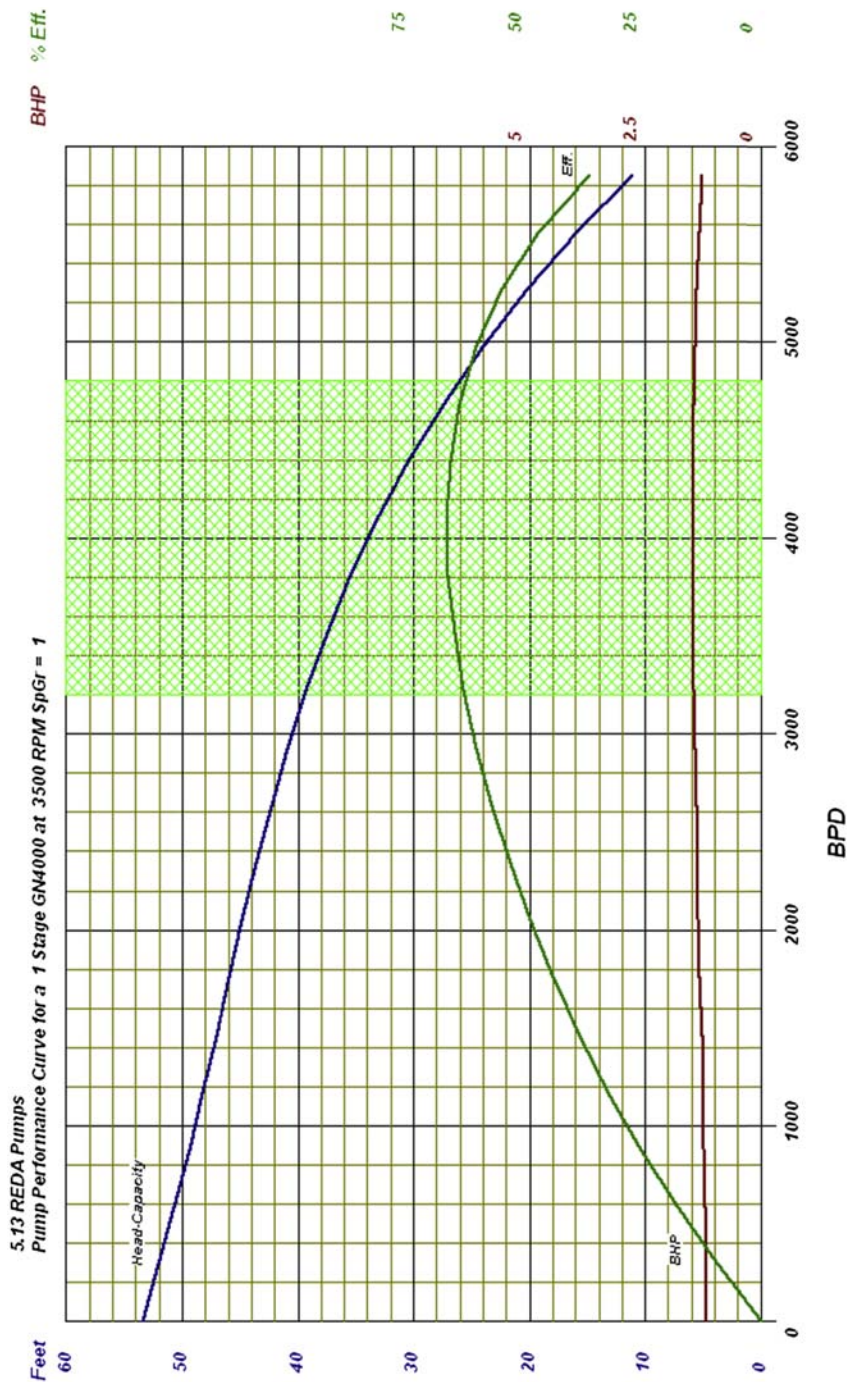
Use the pump performance curve attached and plot the formation plus pump performance curve on the same sheet.

If you use Eq. (2.15) to find the frictional pressure drop in the tubing (not Fig. A.1), then use the following conversion factor: 1 bpd = 0.0292 gpm.

SOLUTION

To plot the performance curve of the formation plus the pump, fill out the following form.

Liquid Rate (bpd)	FBHP (psi)	Frictional Head (ft/100 ft)	Frictional Loss (ft)	Pump Head Required at Wellhead Pressure		
				100	200	300
0						
500						
1,000						
1,500						
2,000						
2,500						
3,000						
3,500						
4,000						
4,500						
5,000						



ESP Design and Analysis**Gabor Takacs, PhD
Instructor****CLASS PROBLEM ASSIGNMENT**

Subject: NODAL Analysis of ESP Installations #2	Class Problem #12
Name of the Trainee:	Date:

PROBLEM STATEMENT

The well with the parameters given in the following is choked on the wellhead to produce 3,200 bpd of water while an **SN3600** pump with 120 stages is installed at 60 Hz operation. Investigate the use of a VSD unit and find the required electrical frequency to produce the same liquid volume without choking the well. Calculate the energy wasted because of choking.

Pump setting depth = 6,500 ft

Static BHP = 2,500 psi

WHP = 100 psi

Perforations at 7,000 ft

Water cut = 100%

Tubing ID = 2.992" old

PI = 3 bpd/psi

INSTRUCTIONS

The performance curve of the given pump is attached. Plot the performance of the formation plus the ESP on the same diagram.

To find the wasted power, use Eq. (6.5).

If you use Eq. (2.15) to find the frictional pressure drop in the tubing (not Fig. A.1), then use the following conversion factor: 1 bpd = 0.0292 gpm.

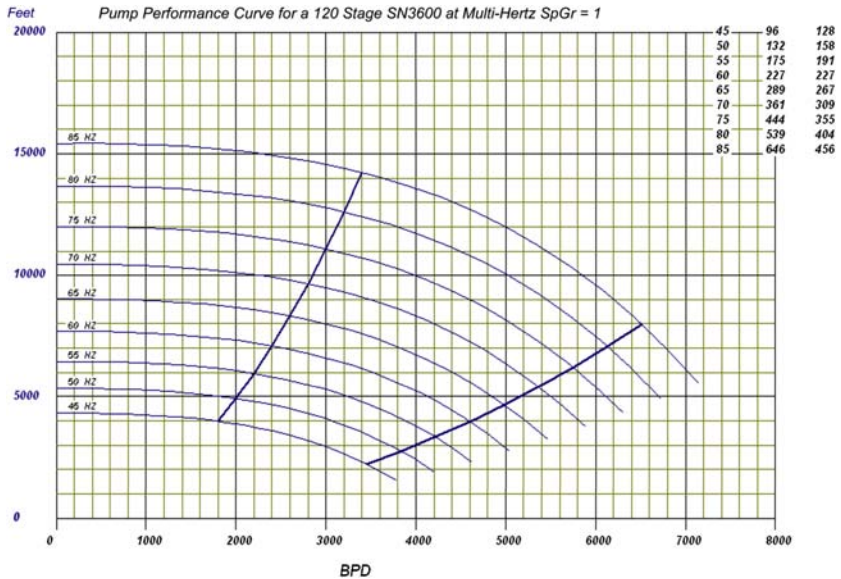
CALCULATIONS

Liquid Rate (bpd)	FBHP (psi)	Frictional Head (ft/100 ft)	Frictional Loss (ft)	Required Head (ft)
1,000				
1,500				
2,000				
2,500				
3,000				
3,500				
4,000				
4,500				
5,000				
5,500				
6,000				
6,500				
7,000				

Required electrical frequency for 3,200 bpd = Hz

Head drop across production choke = ft

Power wasted due to choking: HP



ESP Design and Analysis**Gabor Takacs, PhD
Instructor****CLASS PROBLEM ASSIGNMENT**

Subject: Nodal Analysis of ESP Installations #3	Class Problem #13
Name of the Trainee:	Date:

PROBLEM STATEMENT

Find the maximum liquid rate achievable from the well defined in the following for two cases: (1) at 60 Hz operation and (2) if the motor is run at 85 Hz. Investigate the effect of the flow line size on the rate and use flow line IDs of 2", 3", and 4". An ESP pump J12000N with 41 stages is used.

Pump setting depth = 3,400 ft	Static BHP = 3,000 psi	Separator pressure = 200 psi
Perforations at 3,400 ft	Water cut = 100%	Flow line length = 2,000 ft
Tubing ID = 2.992" new	PI = 10 bpd/psi	

INSTRUCTIONS

Use the pump performance curve sheet supplied and plot the calculated performance curve of the formation plus the pump on the sheet.

If you use Eq. (2.15) to find the frictional pressure drop in the tubing (not Fig. A.1), then use the following conversion factor: 1 bpd = 0.0292 gpm.

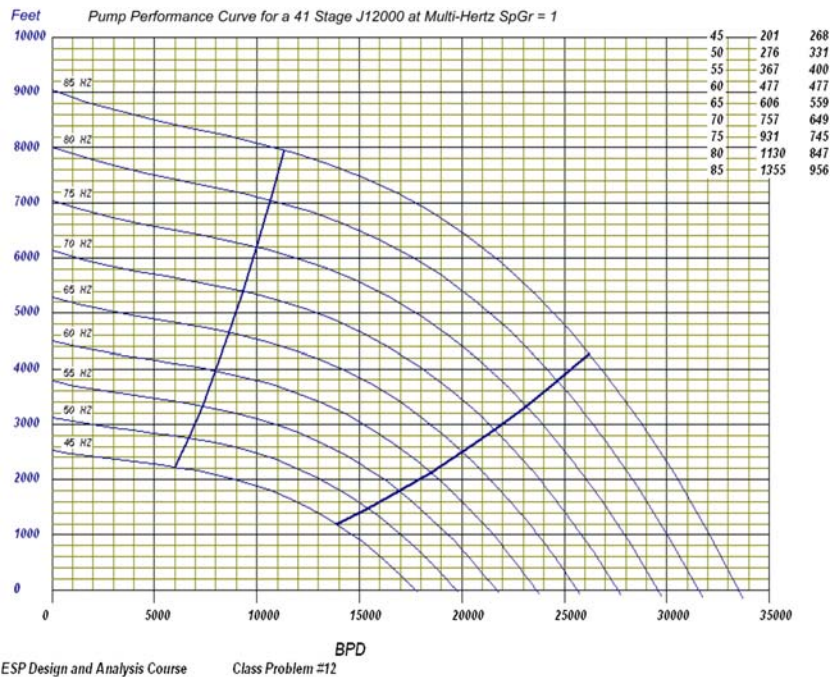
To find the wellhead pressures for the different flow line sizes, add the pressure drops in the flow line to the separator pressure.

SOLUTION

Fill out the following form to define the performance curve of the formation plus the ESP pump.

Rate (bpd)	Flow line Frictional Gradient			WHPs for Flow line Sizes		
	Size (in)			Size (in)		
	2 (ft/100 ft)	3 (ft/100 ft)	4 (ft/100 ft)	2 (psi)	3 (psi)	4 (psi)
0						
5,000						
10,000						
15,000						
20,000						
25,000						
30,000						

Rate (bpd)	FBHP (ft)	Tubing Frictional Gradient (ft/100 ft)	Tubing Friction (ft)	ΔH_{pump} for Flow line Sizes (Eq. 6.4)		
				Size (in)		
				2 (ft)	3 (ft)	4 (ft)
0						
5,000						
10,000						
15,000						
20,000						
25,000						
30,000						



ESP Design and Analysis

Gabor Takacs, PhD
 Instructor

CLASS PROBLEM ASSIGNMENT

Subject: Nodal Analysis of ESP Installations #4	Class Problem #14
Name of the Trainee:	Date:

PROBLEM STATEMENT

Inflow to a well produced by ESP equipment is known to follow the **Vogel IPR** curve with SBHP = 1,600 psi and $q_{\text{max}} = 6,500$ bpd. The ESP pump has 60 stages, and pump performance is given at 60 Hz by the following three points on the single-stage performance curve:

	Minimum Recommendation	BEP	Maximum
Q (bpd)	2,400	3,500	4,600
Head (ft)	59	51	33

Using the coordinate system of the IPR curve, find the liquid rate produced (1) at 60 Hz and (2) at 70 Hz operation.

Other relevant well data are listed as follows.

Pump setting depth = 3,500 ft	Perforations at 4,000 ft	WHP = 100 psi
Tubing ID = 2.441" old	Liquid SG = 1.0	

INSTRUCTIONS

First calculate the performance parameters at 70 Hz, using the affinity laws. Then calculate and plot the performance curves of the ESP–well subsystem for the given number of pump stages. Plot your results on the sheet provided.

SOLUTION

Pump performance curve data for 60 stages

Hz	Minimum		BEP		Maximum	
	Rate	Head	Rate	Head	Rate	Head
60	2,400		3,500		4,600	
70						

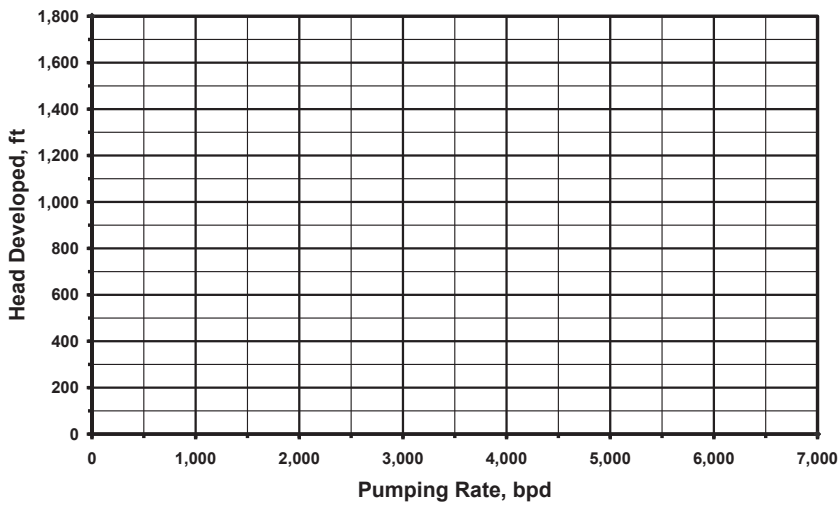
Calculate performance curves for the ESP—well subsystem.

Hz	Rate (bpd)	Δh_{fr} (ft/100 ft)	Δp_{fr} (psi)	FBHP (psi)
60	2,400			
60	3,500			
60	4,600			
70				
70				
70				

Calculate well IPR data.

FBHP	0	160	320	480	640	800	960	1,120	1,280	1,440	1,600
Q											0

Plot your results in the coordinate system provided to find the rates at the two frequencies.



ESP Design and Analysis**Gabor Takacs, PhD**
Instructor**CLASS PROBLEM ASSIGNMENT**

Subject: Nodal Analysis of ESP Installations #5	Class Problem #15
Name of the Trainee:	Date:

PROBLEM STATEMENT

Using the data of **Class Problem #14**, find the liquid rates received from the installation for every 5 Hz increment in the frequency range of 40–75 Hz. Develop an **Excel** sheet for the solution and plot your results.

INSTRUCTIONS

Follow the solution recommended in the previous **Class Problem**.

Subject: Well Testing Using ESP Units	Class Problem #16
Name of the Trainee:	Date:

PROBLEM STATEMENT

Well testing was performed on a well equipped with a 100-stage GN1600 pump set at 5,000 ft and the liquid rate received was 2,500 bpd with negligible gas production. The well is perforated at 6,500 ft, and the well fluid's specific gravity equals $\gamma_l = 0.85$.

The wellhead pressures measured during testing are as follows:

$\text{WHP}_1 = 500$ psi in the shut-in case and

$\text{WHP}_2 = 100$ psi in the stabilized flowing case.

Calculate the well's PI assuming a linear productivity performance.

INSTRUCTIONS

Use the pump performance curve attached.

SOLUTION

First, find the pump's shut-in head: $H_0 =$ _____ ft.

Find the static liquid level in the well from Eq. (6.19):

$L_{\text{st}} =$ _____ ft

Then find the dynamic liquid level in the well from Eq. (6.20):

$L_{\text{dyn}} =$ _____ ft

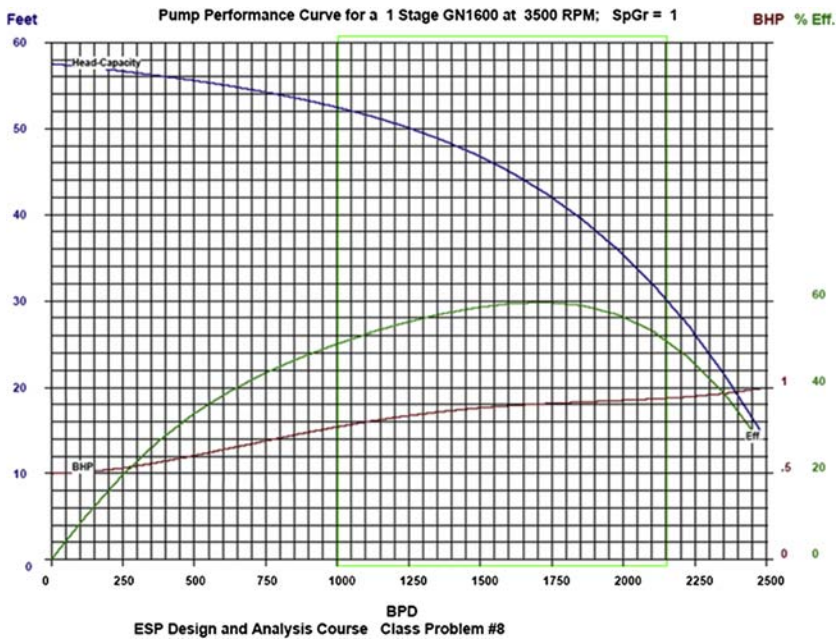
The well's static and flowing bottom-hole pressures are:

SBHP = _____ psi

FBHP = _____ psi

Finally, the well's PI is found:

PI = _____ bpd/psi



ESP Design and Analysis**Gabor Takacs, PhD
Instructor****CLASS PROBLEM ASSIGNMENT**

Subject: Well Performance Analysis	Class Problem #17
Name of the Trainee:	Date:

PROBLEM STATEMENT

Calculate the following well's PI:

Static liquid level = 4,400 ft	Dynamic CHP = 250 psi	Gas SG = 0.75
Static CHP = 300 psi	Depth of perforations = 6,400 ft	
Dynamic liquid level = 5,500 ft	Oil SG = 0.85	

During normal production of a liquid rate of 3,000 bpd, the casinghead was shut in for 10 min and a casinghead pressure increase of 8 psi was experienced.

INSTRUCTIONS

Use the Fig. 6.12 to find the gas pressures in the annulus and Fig. 6.13 to find the gradient correction factor.

SOLUTION

First, calculate the static bottom-hole pressure.

Gas gradient from Fig. 6.12: _____ psi/1,000 ft

Gas column pressure = _____ psi

Liquid column pressure = _____ psi

SBHP = _____ psi

Now calculate the flowing bottom-hole pressure.

Gas gradient from Fig. 6.12: _____ psi/1,000 ft

Gas column pressure = _____ psi

Iterate for liquid gradient correction factor

No. of Iteration	Assumed ε_l	Parameter Group	Calculated ε_l
1	1.0		
2			
3			

Liquid column pressure = _____ psi

The well's flowing bottom-hole pressure, FBHP = _____ psi

Finally the well's PI:

PI = _____ bpd/psi

ESP Design and Analysis**Gabor Takacs, PhD
Instructor****CLASS PROBLEM ASSIGNMENT**

Subject: Well Testing Using ESP Units	Class Problem #18
Name of the Trainee:	Date:

PROBLEM STATEMENT

Find a well's SBHP and PI from the data of a well test performed with an ESP installation using a VSD unit. The liquid rates received at different electrical frequencies are given here:

Frequency (Hz)	Pumping Rate (bpd)
50	2,050
60	2,850
70	3,670

The ESP has 120 stages, and pump performance at 60 Hz is given by the following three points on the single-stage performance curve:

	Minimum Recommendation	BEP	Maximum
Q (bpd)	2,100	3,000	3,200
Head (ft)	11.8	12.5	11.0

Other relevant well data are listed as follows.

Pump setting depth = 4,500 ft	Perforations at 5,500 ft	WHP = 200 psi
Tubing ID = 2.992" old	Liquid SG = 1.0	

INSTRUCTIONS

First calculate the ESP pump’s performance parameters for the given number of pump stages at 50 Hz and 70 Hz, using the affinity laws. Then calculate and plot the performance curves of the ESP–well subsystem.

SOLUTION

Pump performance curve data for 120 stages

Hz	Minimum		BEP		Maximum	
	Rate	Head	Rate	Head	Rate	Head
60	2,100		3,000		3,200	
50						
70						

Calculate performance curves for the ESP–well subsystem.

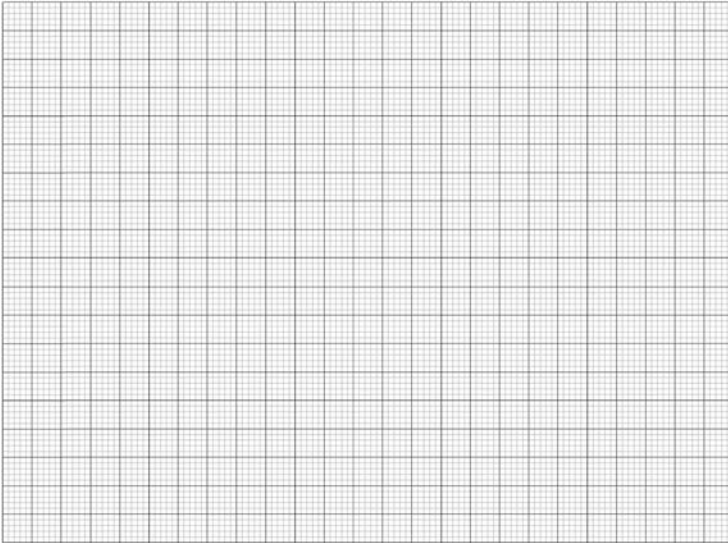
Hz	Rate (bpd)	Δh_{fr} (ft/100 ft)	Δp_{fr} (psi)	FBHP (psi)
60	2,100			
60	3,000			
60	3,200			
50				
50				
50				
70				
70				
70				

Now plot the performance curves for different frequencies on the chart provided.

Plot the measured pumping rates at the proper curves, and after fitting these points with a straight line, read the well's SBHP and calculate its PI.

The well's SBHP = _____ psi

The well's PI = _____ bpd/psi



CLASS PROBLEM ASSIGNMENT

Subject: Cable Size Selection Based on Economic Analysis	Class Problem #19
Name of the Trainee:	Date:

PROBLEM STATEMENT

Find the cable size that ensures the minimum of operational costs for the given installation. Use the following economic data.

Expected Cable Life	60 months
Prime interest rate	12%/year
Cost of electricity	5 c/kWh

The properties of the possible cable sizes are given here:

Size (AWG)	Resistance (Ohm/1,000 ft)	Unit Price (\$/ft)
1	0.139	11.92
2	0.170	10.46
4	0.271	8.71
6	0.431	7.21

SOLUTION

AWG	Capex (\$)	Payback (\$/month)	Cable Resistance (Ohms)	Power Loss (kW)	Cost of Power Loss (\$/month)	Total Cost (\$/month)
#1						
#2						
#4						
#6						

Cable size selected: AWG # ____

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'Note: Page numbers followed by "f" indicate figures, "t" indicate tables and "b" indicate boxes.'

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Electrical Submersible Pumps Manual

Design, Operations, and Maintenance

Gabor Takacs, PhD

Professor at the Petroleum Engineering Department, University of Miskolc, Hungary

Electrical submersible pumps (ESPs) are ideally suited to lift large amounts of liquids from oil wells. *Electrical Submersible Pumps Manual: Design, Operations, and Maintenance* in 2009 was the first book devoted solely to ESP technology, and this second enlarged and completely updated edition presents the latest information necessary to operate and optimize ESP systems.

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Key Features

- Recognize the different components of the ESP system and their operational features
- Understand how special well conditions, such as high viscosity and production of free gas or abrasives, can be overcome
- Apply the best practices to design ESP installations
- Understand the features and operational advantages of permanent magnet motors
- Prevent expensive failures by using proper monitoring, surveillance, and troubleshooting methods

About the Author

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