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Preface

The report is the result of a Master Thesis conducted in cooperation with Statoil ASA which is the world largest offshore oil and gas company.

I would like to thank the following persons for help and guidance with the Thesis:

- Conrad Carstensen UiS, Professor, internal tutor
- Ole Johan Samdal Statoil, Principal Engineer, external tutor
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I would also thank Statoil for giving me the opportunity to take part in positive work environment consisting of highly knowledgeable people.

Bernt Ståle Hollund

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Sammendrag

En form for kunstig løft som har vært benyttet i mange tiår er “Electrical Submerged Pumps” (ESP). ESP ble først benyttet i 1926, siden har teknologien utviklet seg langsomt og dagens design har mange likheter med det originale systemet. ESP blir benyttet i stadige tøffere reservoar forhold, med hensyn på temperatur, viskositet, GOR etc. Dette er en stor utfordring for levetiden til ESP. Derfor er det nødvendig å forbedre eksisterende teknologi for å kunne benytte pumpene på en kostnadseffektiv måte.

Denne rapporten inneholder anbefalinger for ESP design og tilstandsovervåking for Peregrino feltet. Disse anbefalingene er basert på en oversikt av ESP feil-orsaker utført i en database ved navn ESP-RIFTS. Den anbefalte design løsningen består av en ESP med; faste impellere, tandem tetningsseksjon, oppgraderte tekniske spesifikasjoner med vekt på materialer, og benyttelse av avstandsprober for vibrasjonsovervåking. I tillegg er en livssykluskostnads analyse utført som identifiserte utskiftnings operasjoner av ESP som den mest kritiske kostnadsdriveren for Peregrino.

Abstract

One form for artificial lift that has been used for decades is Electric Submersible Pumps (ESP). Since the birth of ESP in 1926 has the technology evolved slowly and ESP used today has a lot of similarities with the original system. Operators are utilizing ESP in more and more challenging reservoirs with respect to temperature, viscosity, GOR, etc. This is a major challenge for ESP run life, so it is necessary to improve the existing technology to gain profit from producing oil with ESP.

This report includes recommendations for ESP design and surveillance for the Peregrino field. These proposals are based on a survey of ESP failures conducted in the database ESP-RIFTS. The proposed ESP design includes; use of compression pumps with tandem seals, upgraded technical specifications with concern to materials, and utilization of proximity probes for vibration measurement. In addition is a LCC analysis performed, that identified work-over costs as the most critical cost drivers for Peregrino.

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Nomenclature

<u>Terms and definitions</u>

Artificial Lift
A method to transport well fluids from a well to surface with use of down-hole equipment.

Best Efficiency Point (BEP)
The flow rate where a centrifugal pump is at its best efficiency.

Bubble Point
The state characterized by the coexistence of a substantial amount of liquid phase and an infinitesimal amount of gas phase in equilibrium.

Bubble Point Pressure
Liquid pressure in a system at its bubble point.

Cable
Component of an ESP system that carries electric power from surface to the down-hole motor.

Cable (MLE)
Segment of the cable that connects the cable to the motor.

Casing
Pipe extending from the surface and intended to line the walls of a wellbore.

Chamber
Segment of a seal section.

Coating
Surface treatment on a material

Corrosive Environment
Operating environment where the combination of temperature and chemicals causes degradation of equipment.

Differential Pressure
Difference between pump intake pressure and discharge pressure.

Diffuser
Stationary stage part which converts the pumped fluid velocity to pressure.

Efficiency
Output work divided by input work.

Failure Rate
Total number of failures observed within a group of production periods, divided by the sum of the known runtime for all ESP systems in the same group.

Flow Rate
Volume of fluid pumped per unit of time

Functional Test
Test performed to confirm proper equipment operation.

Functional Test – Factory Acceptance Test (FAT)
Test performed to confirm proper ESP component operation.

Impeller
Stage part which rotates with the shaft and adds energy to the fluid being pumped.

Maximum Recommended Flow Rate
The highest flow rate for a particular pump stage as specified by the vendor.

Minimum Recommended Flow Rate
The lowest flow rate for a particular pump stage as specified by the vendor.

Motor
Component of the ESP system which converts electric energy to mechanical energy.

Mean Time to Failure (MTTF)
Represents the average runtime one can expect to get until a ESP system failure occurs.

Operator
User of the ESP equipment

Perforations
Section of casing perforated to allow in-flow of well fluids.

Producing gas oil ratio (GOR)
Ratio of produced gas to produced oil.

Production Period
Any individual ESP system installed in an individual well for a period of time.

Productivity Index
The number of barrels of oil produced pr day pr decline in well bottom-hole pressure in pounds pr square inch.

Pump Intake
Component of the pump which provides a flow path to the first impeller.

Seal Section
ESP component with main objective to protect the motor from well fluids.

Shaft
Solid or tubular bar that transmits torque in the ESP system.

Stage
Part of the ESP pump where the impeller and diffuser creates pressure.

Stator
Segment of the motor which contains electrical laminations and coiled wire.

Tubing
Pipe located in a well to serve as production conduit.

Water Cut
Ratio of produced water to produced liquids, given in percentage.

Wellhead
Component which include valves that control the well.

Abbreviations

AC	Alternating Current
API	American Petroleum Institute
BEP	Best Efficiency Point
BHP	Brake Horsepower
BPD	Barrels / day
DC	Direct Current
DGU	Discharge Gauge Unit
ESP	Electric Submersible Pump
ESP-RIFTS	Electric Submersible Pump - Reliability Information Failure Tracking System
ET	Electronics Temperature
ft	Foot
GOR	Gas Oil Ratio
HP	Horsepower
HSP	Hydraulic Submersible Pump
Hz	Hertz
IEC	International Electrotechnical Commission
ISO	International Organization for Standardization
JIP	Joint Industry Project
kW	Kilowatt
LCC	Life Cycle Costs
MGU	Motor Gauge Unit
MLE	Motor Lead Extension
MNOK	Million Norwegian Kroner
MT	Motor Temperature
MTTF	Mean Time To Failure
PCP	Progressive Cavity Pump
PDP	Pump Discharge Pressure
PI	Productivity Index
PIP	Pump Intake Pressure
PIT	Pump Intake Temperature
PPD	Pump Pressure Differential
RPM	Revolutions per minute
SAGD	Steam Assisted Gravity Drainage
TDH	Total Dynamic Head
TVD	True Vertical Depth
VM	Vibration Measurement
VSD	Variable Speed Drive
WO	Work-over

1 Introduction

1.1 Background

As an oil field is produced, the reservoir pressure declines. After a while the pressure becomes insufficient to lift the produced fluids to the surface. When this natural lift becomes insufficient, artificial lift methods are necessary to lift hydrocarbons to the surface.
One form of artificial lift that has been used for decades is Electrical Submerged Pumps (ESP).

Since the birth of ESP in 1926 the technology has not evolved significantly and the technology of today has many similarities with the original ESP system designed in 1926. Today approximately 10% of the world oil production is produced with electrical submerged pumping. ESP are also known for lifting much greater liquid rates than most of the other types of artificial lift, both on-and offshore [1]. In the future, ESP will be used in more challenging reservoirs with high temperature, viscous oil, long step outs and large gas/oil ratios. This is a major challenge for existing ESP systems. It is necessary to improve present technology to make sure that oil production with use of ESP is done in an economical beneficial way. Statoil will start producing from several new challenging fields in the close future by use of ESP, and are investigating how they can improve existing technology to be able to increase the profitability of utilizing ESP. One of these fields is Peregrino, which is located offshore Brazil and include 30 oil producing wells about 2.500m below the sea level.

1.2 Main Objective

The main objective of the thesis is to establish best practice proposals to enhance ESP run life for the Peregrino field with concern to the following aspects:

- ESP Design
  - Recommend technical specifications for pump and seal section
- ESP Surveillance
  - Recommend surveillance system for the selected pump

In addition will it be performed a Life Cycle Cost (LCC) analysis to identify cost drivers.

1.3 Secondary Objectives

The thesis has the following secondary objectives:

- Describe design and operation of ESP
- Perform a survey of ESP failures

To be able to select technical specifications is it necessary to understand how ESP equipment is designed and operates. In addition, should design limits for the system be identified for different applications. To understand ESP limitations and weak areas, a survey of ESP failures are created. This survey will also help identify critical parameters that should be monitored by a surveillance system to enhance run life.

1.4 Work Method

Vendor catalogues, public information on the internet and internal Statoil documents are used to describe ESP systems and area of application. To collect operational data to generate the survey of ESP failures a designated database called ESP-RIFTS are utilized. This database contains failure information from several major oil companies which utilize ESP in a wide range of wells. Engineering software is used for ESP sizing. How to manually calculate ESP size for Peregrino is outlined in this report.

1.5 Limitations

There are several forms for artificial lift. However ESP is seen by Statoil to be the right concept to optimize.

Electric Submersible Pumps are used in offshore and onshore applications, where they are producing either water, oil or a combination. In this Thesis, offshore oil field applications will be considered.

When performing a survey of ESP failures the main focus is on mechanical down-hole equipment. ESP equipment that will be considered when determining best practice for Peregrino is the pump and seal section.

The Peregrino field has three different production ranges; short, medium and long well. This report will consider medium well production range.

The ESP industry is utilizing the American Petroleum Institute (API) standards and no ISO standard exists at this time, however an ISO standard is under development. The report will therefore have some traces of American standards when it comes to numerical units.

2 Artificial Lift

2.1 Oil production

In a reservoir, oil and gas are contained in the pore spaces of the rock. Lifting the hydrocarbons from the reservoir to surface require a certain energy. All reservoirs contain energy in the form of pressure, in the compressed fluid itself and in the rock [3]. In some reservoirs the composition of the rock allows the hydrocarbons to move freely, making it easy to recover. However in other reservoirs the rocks do not part as easily with the oil and gas and require special techniques to move the fluids from the pore spaces in the reservoir rock to the surface [4]. The driving force in a reservoir is either by water or gas. A water drive reservoir occurs when there is a big underlying aquifer where the water is able to flow into the oil layer. Once production from the oil layer begins it will create a pressure drop, the aquifer then responds by expanding up into the oil layer to replace the voidage. A gas drive reservoir derives its energy from gas expansion of a gas cap or from breaking out of solution [5]. Early in a wells production life the reservoir pressure will be sufficient to push the hydrocarbons up to the surface. When the pressure differential becomes insufficient for the oil to flow naturally, some method of lifting the oil to surface must be implemented. One can either use something called pressure maintenance or artificial lift [4]. Pressure maintenance is about injecting water or gas into the reservoir to maintain the pressure on an acceptable level. Artificial lift systems distinguish themselves from pressure maintenance according to Cook by adding energy to the produced fluids in the well; the energy is not transferred to the reservoir [3]. The purpose of any artificial lift method is to add energy to the produced fluids, either to improve or to enable production. Some wells need artificial lift to increase production rate, others need artificial lift to be able to start producing [4]. There are several different forms of artificial lift that can be used for different operating conditions [6].

2.2 Artificial Lift Methods

The most common forms for artificial lift are:

- Rod Pumps
- Gas Lift
- PCP
- HSP
- ESP

2.2.1 Rod Pumping

Rod pumps are the most widely used form of artificial lift. A rod pump typically consists of a prime mover, gearbox, walking beam, sucker rod strings and a pump see Figure 1. The rod pump works by reciprocating a rod string that activates a displacement pump. The pump has a plunger and valve assembly that converts the reciprocating motion to vertical fluid movement. The rod pump also contains a counterweight that reduces the power requirement and increases efficiency [6].

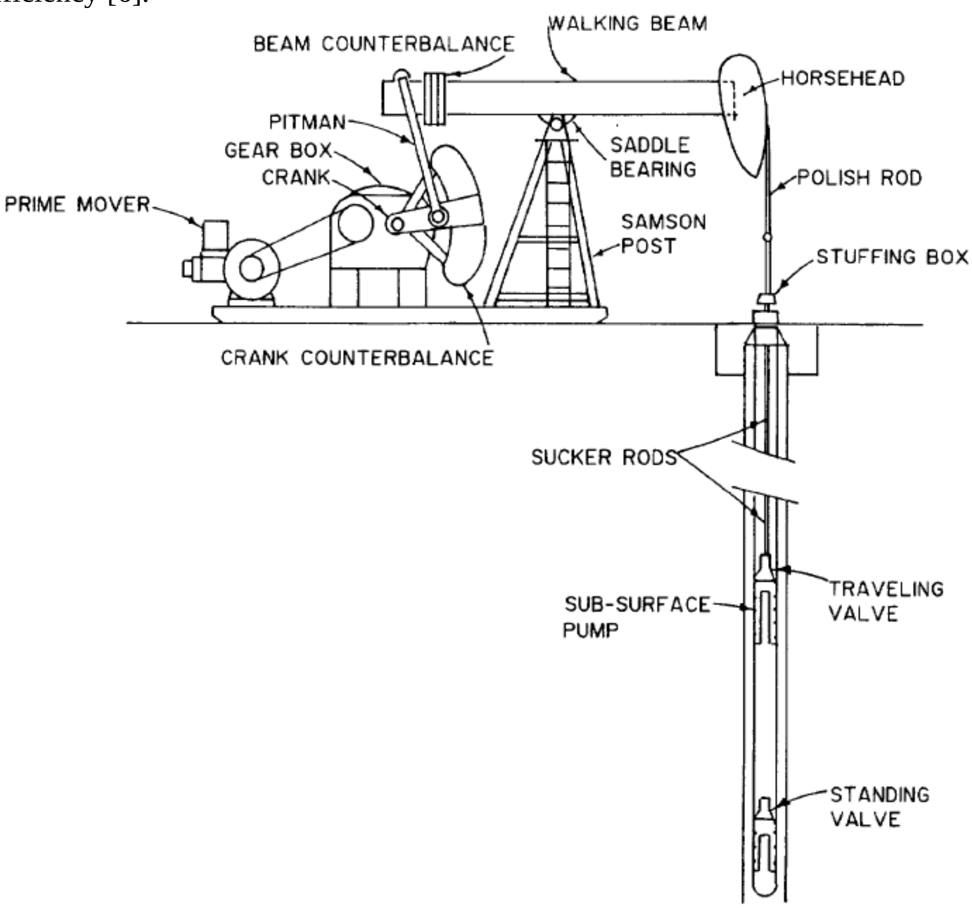

Figure 1. Rod Pump. [7]

Advantages

- High system efficiency
- Economical to do maintenance and repairs
- Flexibility – can adjust production through stroke length and speed

Disadvantages

- Limited to low production volumes, <1.000 BPD.
- Takes a lot a surface space
- Mainly onshore application
- Many moving parts (friction, material fatigue) [4].

2.2.2 Gas Lift

Gas lift is a form of artificial lift where compressed gas is injected through gas lift mandrels and valves into the production string. The injected gas moves the reservoir fluid to the surface by reducing the hydrostatic pressure of the fluid column in the tubing below the reservoir pressure [4].

Gas lift systems can be installed to operate continuously or intermittently, depending on the producing characteristics of the well and the arrangements of the gas lift equipment. Intermittent gas lift is often used in low producing low pressure wells. Intermittent operation allows for the build up of pressure in the reservoir. Continuous gas lift is often used in high pressure, high flow wells (100-75.000 BPD) [6].

Advantages

- High flow rate
- Excellent gas handling
- Good solids handling
- Low maintenance

Disadvantages

- Can not be applied if no source of gas is present
- High initial capital purchase cost
- Difficult to operate.
- It is the least energy efficient method of artificial lift [4].

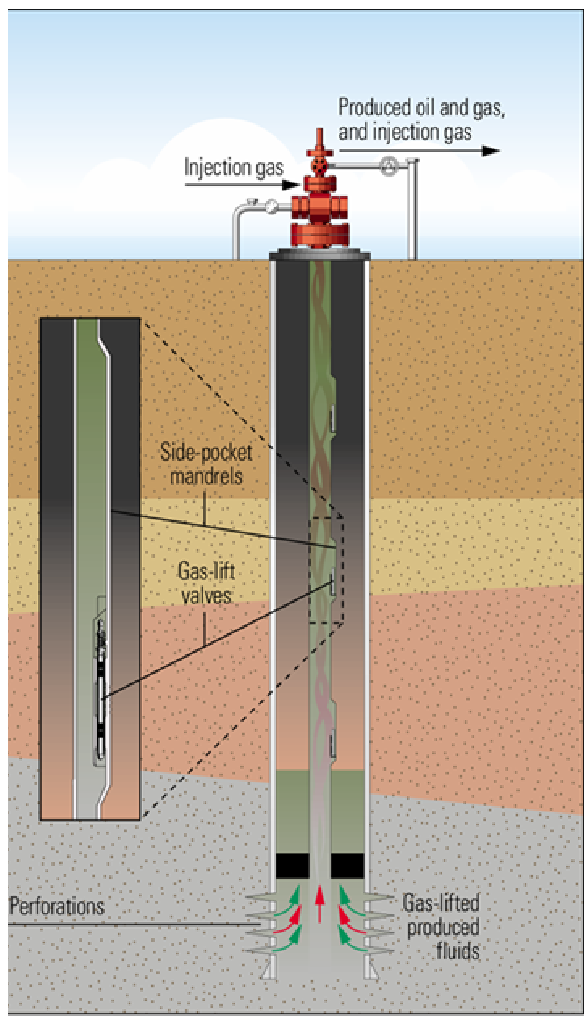

Figure 2. Gas Lift. [8]

2.2.3 Progressive cavity pump (PCP)

PCP systems normally consist of a surface drive, drive string and down-hole progressive cavity pump see Figure 3. The PC pump consists of a single-shaped rotor which turns inside a double helical elastomer stator. The stator is connected to the production tubing string and remains stationary during pumping. In most cases the rotor is attached to a sucker rod string which is suspended and rotated by the surface drive. The rotor turns inside the stator creating a series of sealed cavities. The fluid travels up the pump as one cavity closes and the next opens. The result is a non-pulsating positive displacement flow with a discharge rate proportional to the size of the cavity, rotational speed of the rotor and the differential pressure across the pump.

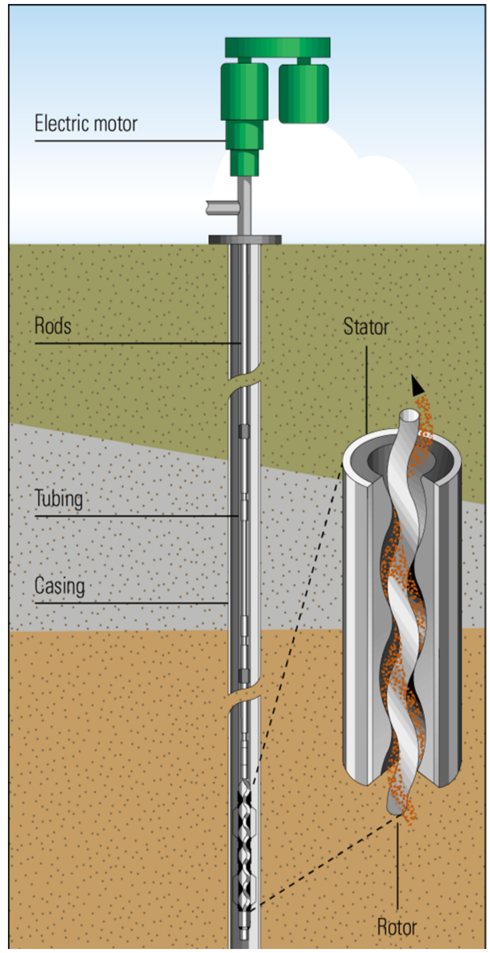

Figure 3. Progressive Cavity Pump. [8]

In some cases, PCP systems are connected to Electric Submersible Pumps rather than using a sucker rod string.

Advantages

- Low capital investment
- High system efficiency
- Low power consumption
- Good gas handling
- Excellent solids handling
- Simple installation
- Portable, lightweight surface equipment

Disadvantages

- Limited lift capabilities
- Restricted flow rates [4].

2.2.4 Hydraulic Submersible Pumps (HSP)

A HSP system normally consists of a surface power fluid system, a prime mover; a surface pump and a submerged pump (see Figure 4). In a HSP system power fluid (crude oil or water) is taken from a storage tank and fed to a surface pump. The surface pump will boost the power fluid and send it to a wellhead. Then the power fluid passes through the wellhead valve and is directed to the down-hole pump [4]. There are two different types of hydraulic pumps; piston or jet. In a piston pump assembly, power fluid actuates the engine, which in turn drives the pump, and power fluids return to surface with the produced hydrocarbons. In a jet pump assembly the venturi principle is being employed to bring the hydrocarbons to surface [7].

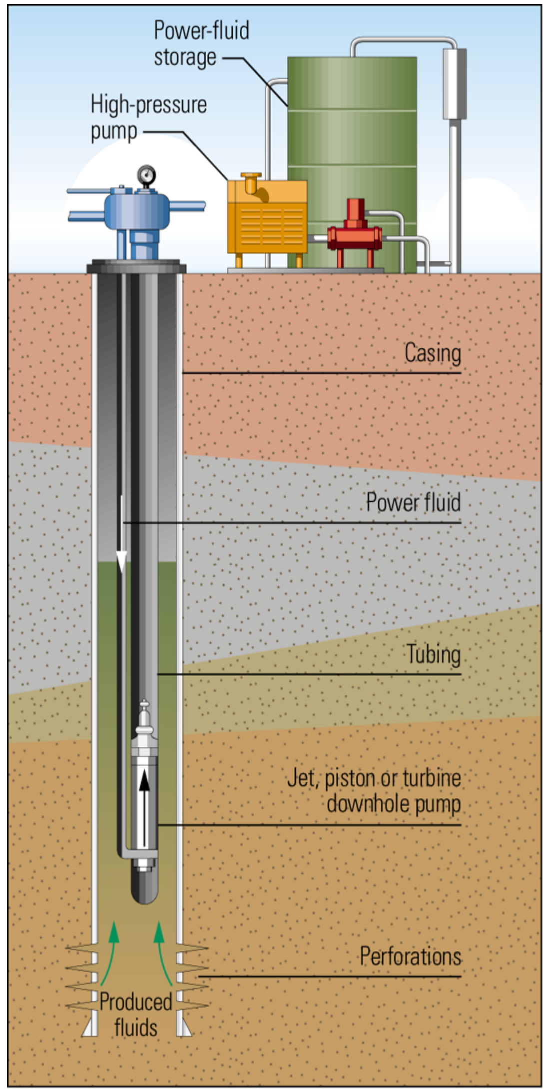

Figure 4. Hydraulic Submersible Pump. [8]

Advantages

Jet pump

- No moving parts
- High volume capability
- Multiwell production from a single system
- Low pump maintenance

Piston pump

- Positive displacement-strong drawdown
- Double-acting high-volumetric efficiency

Disadvantages

- High initial capital cost
- Difficult to operate
- Only economical in cluster wells
- If there is a problem with the surface system or prime mover, all wells are shut-down [4].

2.2.5 Electric Submersible Pumps (ESP)

A typical ESP system consist of an electric motor, seal section, gas separator, multi stage centrifugal pump, power cable, surface control mechanism and transformers see Figure 5. The centrifugal pump is driven by an electric motor that gets power supply from surface. [6]

The ESP system is installed above the well perforations. When fluids enter the wellbore they flow past the motor which is connected to the bottom of the string and provides cooling. Then the fluids flow through the seal and into a gas separator that removes a great part of the gas (separators are optionally). Further the fluids enter the pump intake and get lifted by several pump stages to the surface [4]. The ESP system deliver an effective and economical means of lifting large volumes of fluids from deep wells under a variety of well conditions. ESP is a versatile form of artificial lift and is in operation all over the world [6].

ESP is normally used in high volume (over 1.000 BPD) applications.

Advantages:

- Minimal surface equipment
- High resistance to corrosive down-hole environments
- Can be used in deviated wells
- Can handle high temperatures
- Can handle a wide range of flow rates

Disadvantages:

- Poor ability to pump sand
- Sensitive to gas
- Major work-over [4].

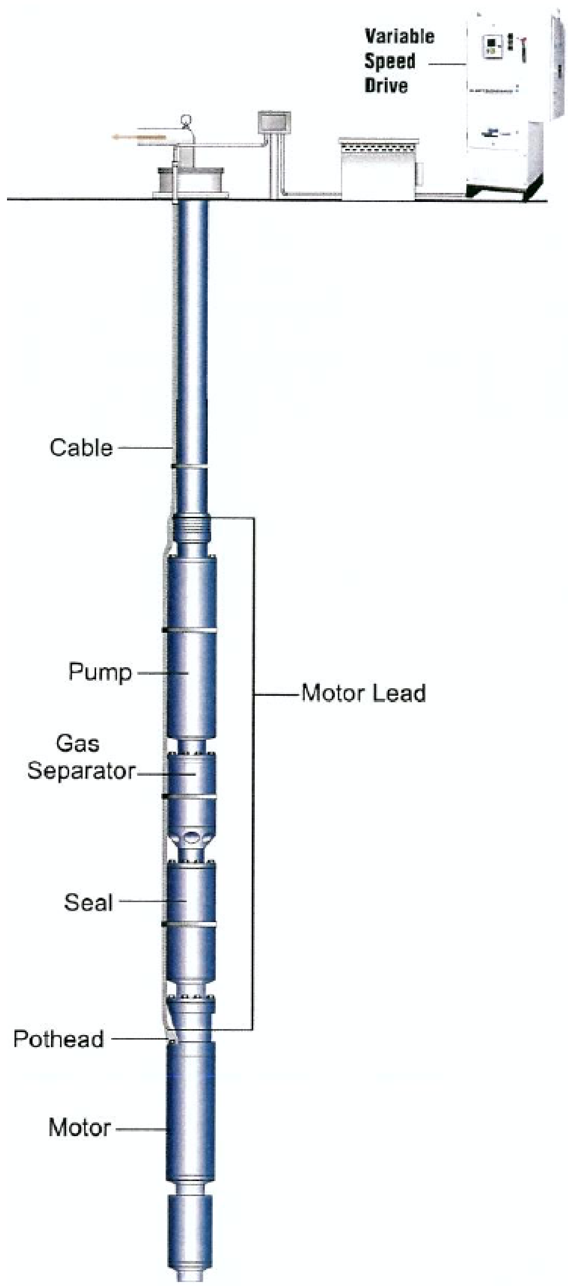

Figure 5. Electric Submersible Pump. [6]

A more detailed explanation of each component and its mission will be given in chapter three.

3 Electric Submersible Pumps

A Russian named Armais Arutunoff invented electrical submersible pumping in 1910. He also became the founder of the company Russian Electrical Dynamo of Arutunoff (REDA). Arutunoff received his US patent for the electrical submersible pump in 1926 and the same year the first ESP system was successfully operated.

The first ESP had a 6 meter long electrical motor and a multistage centrifugal pump. Between the motor and the centrifugal pump there were attached a seal. Electric power was supplied form a three phase cable from surface. The whole ESP unit was run into a well on the bottom of a tubing string. Today, these are still the main components in an ESP system, and from an external viewpoint there have been small changes since Arutunoff's days. What has changed significantly over the years is the component materials and functionality. ESP performance has increased significantly; motor effect from 105 HP to 1600 HP, and liquid rates from 1.000 BPD to around 30.000 BPD [1]. See Figure 6.

ESP equipment has during its long history had a continuous improvement. In the early 1950s seal sections with mechanical seals on their shafts were introduced, this was the first breakthrough in improving ESP run life. The new seals provided a much better protection against leakage of well fluids into the motor. In the early days of ESP history, production of gassy wells was a significant problem and their simple gravitational separators did not solve the problem. The first rotary gas separator was introduced in the early 1970s and was a major improvement on gas separation. Other ESP components have also evolved during this period, but the next breakthrough came when the first VSD unit were introduced in 1977. This made it possibly to adjust ESP performance to handle changing well conditions. [1]

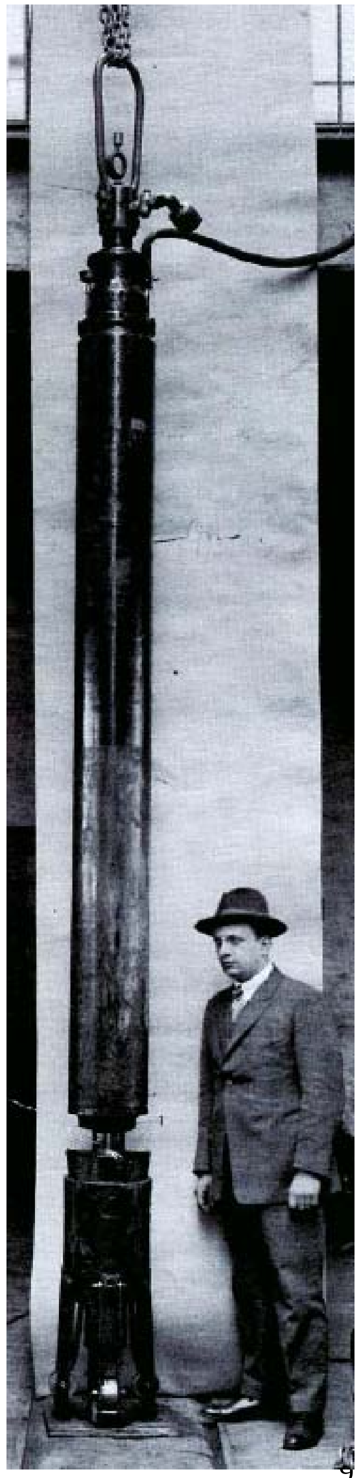

Figure 6. Arutunoff and his ESP motor. [2]

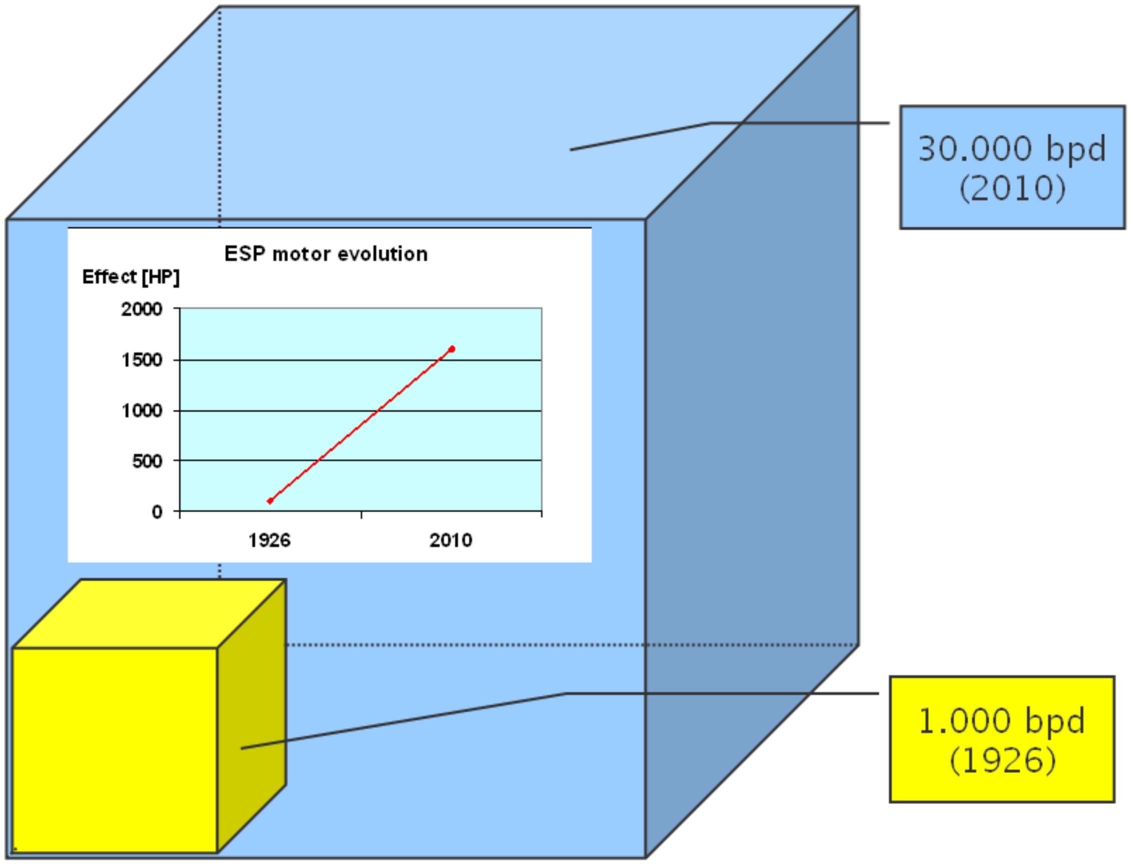

Figure 7. ESP Evolution

3.1 ESP Equipment

3.1.1 Pump

The heart of the ESP system is the submersible pump; to get an understanding of how the whole ESP unit functions it is important to understand the operation of the pump. This is why the description of the system components has to be started with a thorough analysis of the construction and operation of the pump. Pumps in the petroleum industry can be classified in two groups; displacement pumps or dynamic pumps. Rod and PCP pumps are of the displacement type while ESP's work on the dynamic principle. ESP utilize submersible centrifugal pumps driven by electric motors, that convert the energy from the rotating shaft into centrifugal forces that lift well fluids to surface [6]. Main features of centrifugal pumps in ESP systems:

- multistage pumps
- they have radial or mixed flow configurations
- operates in a vertical position

The submersible pumps used in ESP systems have had a continuous evolution over the years but their basic operational principle remains the same. Well fluids, 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. [1]

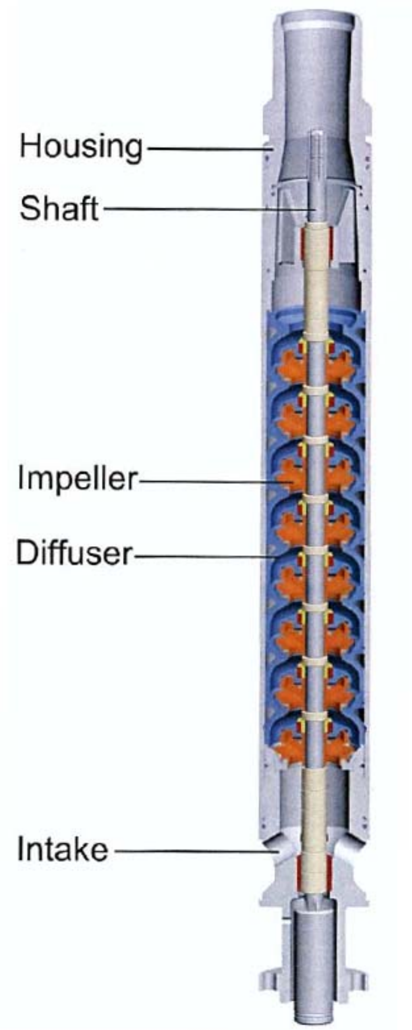

Figure 8. Main components of ESP. [6]

Main components in the submerged pump:

Impeller

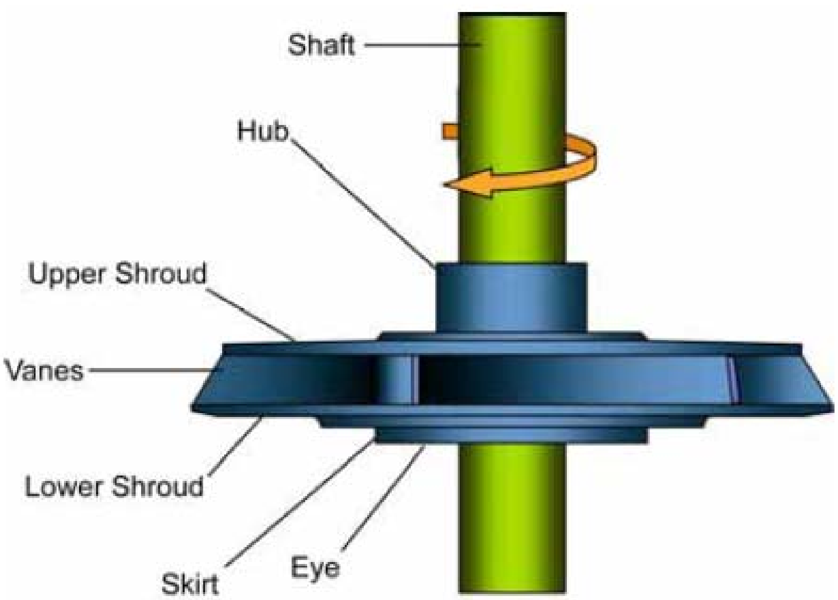

Figure 9. Impeller and Sub-Components. [6]

The impeller is locked to the shaft and rotates with the RPM of the motor. When the impeller rotates it transfer centrifugal force on the production fluid. Figure 9 is an illustration of an impeller keyed to a shaft, and sub-components of the impeller [6]. Two types of impeller designs are available; fixed and floating impellers.

Fixed impellers

Pumps with fixed impellers (also called compression pumps) have impellers locked in position in the shaft. The impeller hubs (see Figure 9) are in contact with each other so that they have no clearance to move axially [6].

Floating impellers

In pumps with floating impellers (also called floater pumps) are the impellers allowed to move axially between the diffusers, since the impeller hubs is not stacked on top of each other [2].

Diffuser and Pump Stage

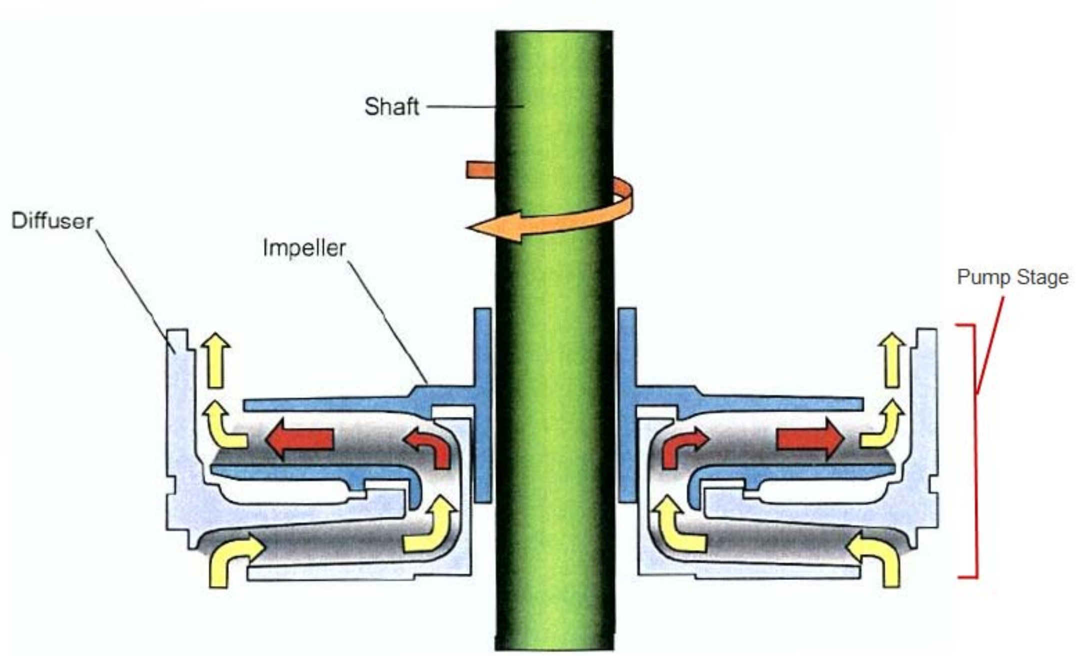

Figure 10. Pump Stage. [6]

The diffuser turns the fluid into the next impeller and is stationary. 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. A pump stage is formed by combining an impeller and a diffuser. [1]

Shaft

The pump shaft is connected to the motor (through the gas separator and seal section), and spins with the motor speed. The pump shaft turns the impeller with the help of keys fitted into the key-way of the impeller. [1]

Intake

The pump intake is attached to the lower end of the pump and provides a passageway for fluids to enter [6].

Other components in the submerged pump include the radial bearings along the shaft, which provide radial support to the pump shaft. An optional thrust bearing takes up part of the axial forces arising in the pump, but most of those forces are absorbed by the seal thrust bearing. The flow capacity of the submerged pump depends on the following factors:

- the rotational speed provided by the motor
- diameter of the impeller
- design of the impeller
- the actual head against which the pump is operating
- fluid properties (density, viscosity, etc.)

For constant speed ESP applications the most important factor is impeller size, which is limited by the ID of the well casing. Pumps with big impellers can produce larger liquid rates, although impeller design also has a significant impact on pump capacity. ESP pumps which are available today come in different capacities from a few hundred to around 80.000 BPD of production rate, and with OD diameters from around 3 -11 inches. Smaller pumps are used up to the rates of 1.500 – 3.500 BPD. For ensuring proper assembling and ease of handling, pump length are limited to about 6 – 8m. Up to three pump-sections can be connected together in series, to achieve higher operational heads usually required in deeper wells. Such an assembly can have several hundred pump-stages; the maximum number of stages is limited by one or more of these factors:

- the strength of the pump shaft
- the maximum burst-pressure rating of the pump housing
- the maximum allowed axial load on the pump's main thrust bearing

Individual pump stages handle the same fluid volume and develop the same amount of head. Head is a measure of the pressure exerted by the fluid, often in meter of bar. Each pump stage creates a certain amount of head in order to lift the fluid to surface. Head is created by utilizing the power generated by the motor and transferred through the shaft. The impeller rotates at the same speed of the shaft and transfer centrifugal energy to the fluid. The impeller forces the fluid to the outside of the stage where it exits the impeller and enters the diffuser of the next pump stage. The diffuser then redirects the fluid up into the next impeller and the process repeats. The head one stage produces is the net of the energy imparted by the impeller and the energy lost while passing through the diffuser. The head that one stage develops can than be multiplied by the number of stages to determine the total head the pump will deliver. [6]

3.1.2 Electrical Motor

The main purpose of a motor is to convert electrical energy into mechanical energy that turns the shaft. The shaft is connected through the seal and gas separator and rotates the impellers inside the pump.

An ESP motor consists of the following major components:

- Rotors
- Stator
- Shaft
- Bearings
- Insulated Magnet Wire
- Winding Encapsulation
- Rotor and Stator Laminations
- Housing
- Bearing. [6]

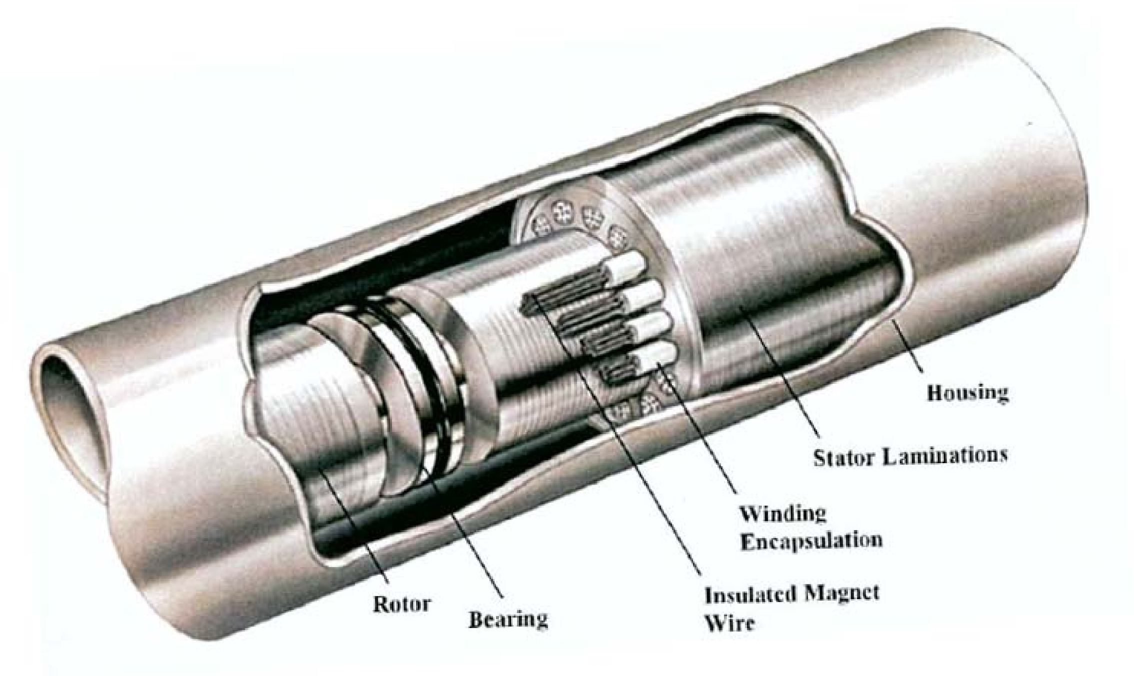

Figure 11. Motor Cut-Away Illustration. [6]

An ESP motor is a three-phase, two pole, squirrel cage induction-type electric motor. It work on the principle of the electromagnetic induction that states an electric current induced in any conductor moving in relation to a magnetic field generated in the stator. The field rotates with the changes of direction of the AC current since the electromagnets change their magnetic poles twice for every cycle of the AC current. The motor synchronous speed is equal to the speed of the magnetic field, which depends on the frequency and the number of poles the motor has [1]. ESP motors normally run at approximately 3600 RPM on 60 Hertz power systems. The operating voltage of ESP motors can vary from 230 – 7000 volt. By increasing the length or diameter of the motor the effect can be increased to achieve the required horsepower. But since we have a determined diameter in an oil well ESP motors are often made very long, maybe 10m to get enough power. [6]

Figure 11 shows the basic construction of an ESP motor. The stator which is connected to the housing is a hollow cylinder made up of a great number of tightly packed steel discs called stator laminations. This solution prevents that eddy-currents are occurring in the metal of the stator. Inside the laminations there are several slots which accommodate the insulated copper stator windings called "magnet wire" connected to the AC power. Along the perimeter of the motor there are three pairs of coils. To make sure that no electrical failures are occurring in the windings, the motor must have a insulating system which include:

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

The rotor consists of rotor laminations and is located inside the stator, separated from it by an annular air gap. The slots of rotor laminations contain a set of copper bars making up the squirrel cage. The centre bore of the rotor laminations has an axial keyway that accepts the key that connects the laminations to the motor shaft and allows transmission of torque to the shaft. Because of high rotational speeds, the rotors are made up of short segments with radial bearings between them. The rotating magnetic field developed in the stator windings induces a current in the rotor which creates a magnetic field. The interaction of the two magnetic fields turns the rotor and drives the motor shaft, which again are connected to the pump impellers. A motor shaft can be up to 10 m long, it is therefore crucial to eliminate radial vibrations. This is why there are radial bearings located at several places along the shaft's length. The motor is filled with refined oil that provides dielectric strength, lubrication, and cooling. The motor shaft is hollow to allow the oil to circulate, and a filter is provided to remove solid particles from the oil. Electric motors used in ESP are very different from "normal" motors which is common on the surface, the most important differences are:

- their length to diameter ratio is much greater than surface motors
- they are cooled by the well fluid and not surrounding air
- they are connected to the surface power source by long cables, where a substantial voltage drop can occur

As mentioned earlier the only way to increase motor power is to increase the length of the motor. But it is possible to connect two or three motors in tandem to achieve higher power ratings, see Figure 12. The two motors are mechanically coupled but work independently in an electrical sense. Motor power can reach 2.000 HP, allowing the production of 30.000 BPD. [1]

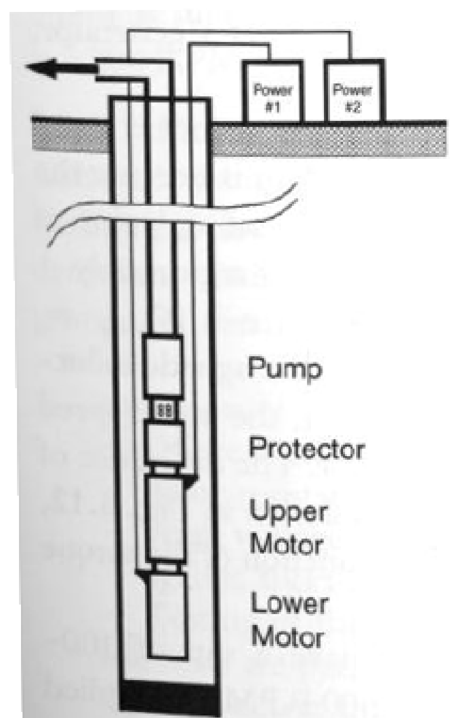

Figure 12. Tandem motor. [1]

3.1.3 Gas Separator

If free gas is allowed to enter a centrifugal pump it will deteriorate its performance. This is because it is a great difference between the specific gravity of liquid and gas. The amount of kinetic energy passed on to the fluid in a centrifugal pump, greatly depends on the fluid density. Since liquid is denser than gas, it receives a great amount of kinetic energy that after conversion in the pump stage, increases the pressure. Gas however, although being subjected to the same rotational speed, cannot generate the same amount of pressure increase. This is why ESP pumps always should be fed by single phase well fluid to ensure reliable operation. Pumping of well fluids with free gas can have the following effect on the ESP pump:

- The head developed by the pump decreases.
- The output flow fluctuates; cavitation can occur at higher flow rates causing damage to the pump stages.
- In wells with extremely high gas/oil ratio, gas locking may occur.

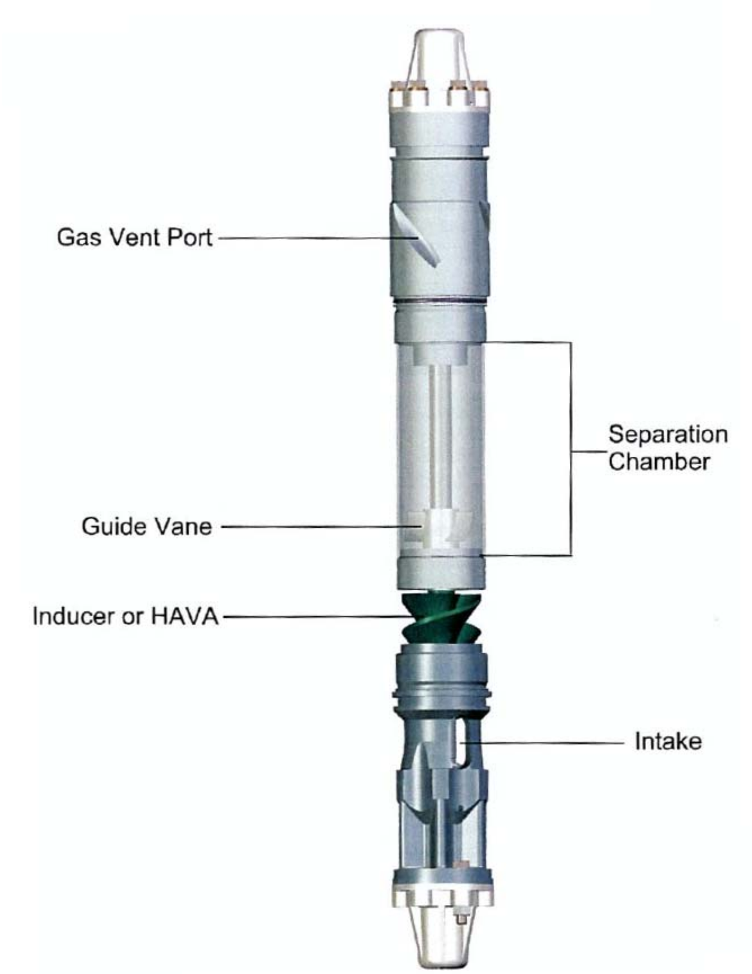

Figure 13. Rotary gas separator. [6]

In wells with high GOR, gas separators replace standard pump intakes and helps improve pump performance by separating a portion of the free gas before it enters the first stage. This helps gas locking from occurring and improve the reliability of ESP systems [6]. The most common separator type used in ESP is rotary gas separators, see Figure 13. They work on the principle that a multiphase fluid, if spun at high speed is separated to liquid and gas phases because of the different levels of centrifugal force acting on the liquid and gas particles. The rotational spin is provided by the separator shaft which is driven by the motor. Separation takes place in the separator chamber. Where the heavier fluid is being forced to the outer wall in the chamber and the lighter gas gathers along the shaft. Then the gas is being directed into the casing annulus and the liquid is being directed to the pump intake. [1]

Typical separator efficiencies is 80% or higher, this efficiency is affected by flow rates, viscosity, and percentage of free gas vs. total volume produced. In extremely high gas conditions, tandem gas separator assemblies can be used to further improve the separator efficiency [6]. Statoil have to comply with Norwegian regulations that requires that no gas should be vented through the casing annulus, this is because of safety barrier issues. Thus Statoil have to use a form for separator that does not direct free gas into annulus but still manage to separate the gas from the well fluid. One such method is Schlumberger advanced gas handler called Poseidon, shown in Figure 14. This makes use of a gas handler utilizing special pump stages originally devised for transferring multiphase mixtures. Poseidon contains impellers with helio-axial vanes and diffusers providing a smooth axial flow. This method ensures an almost homogeneous distribution of gas particles in the fluid.

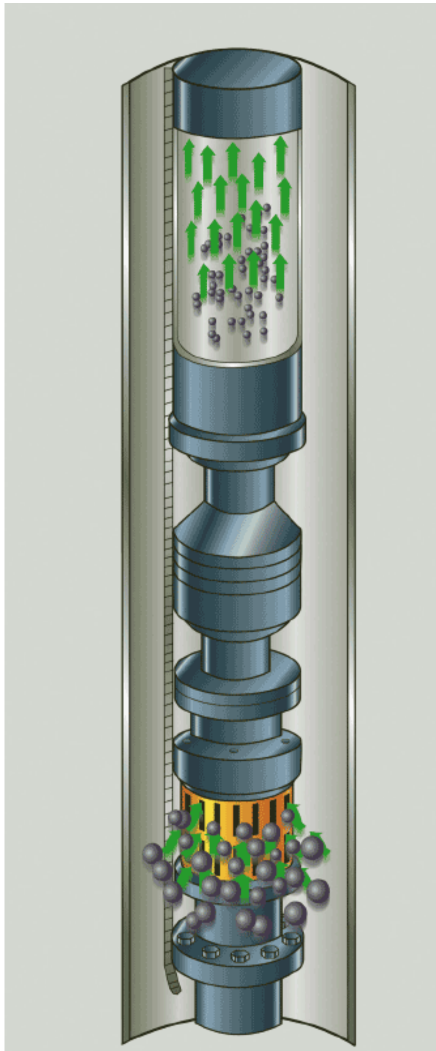

Figure 14. Poseidon. [10]

Poseidon can either be connected above a gas separator when gas is allowed to travel up annulus, or it can be connected above a standard intake if the gas has to go through the pump [9]. The unit can handle well-streams with up to 75% of free gas content. It can handle flow ranges between 5.000 BPD and 9.000 BPD and need a substantial power of 50 HP to operate. [1]

3.1.4 Seal Section

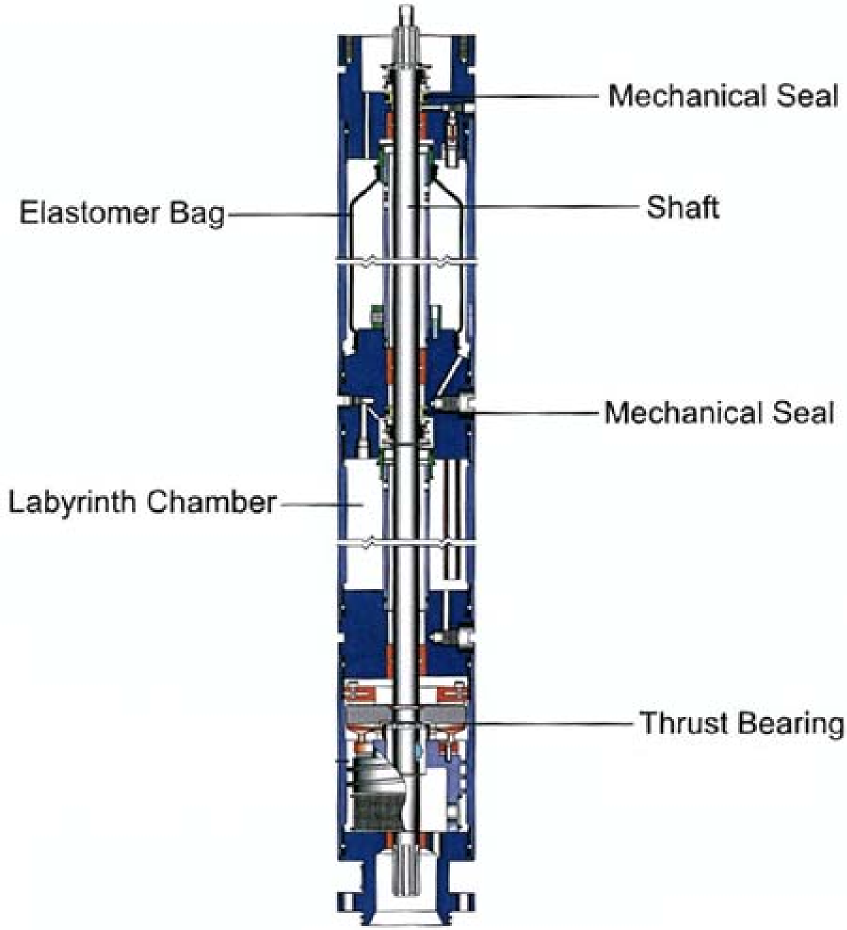

Figure 15. Seal Components. [6]

Main components in a Seal Section:

- Mechanical Seals
- Elastomer Bags
- Labyrinth Chamber
- Thrust Bearing

The seal section connects the motor shaft to the gas separator shaft. Seal sections also perform four crucial functions: [6].

1. It allows for expansion and contraction of the motor oil. High well temperature and heat generated in the motor itself causes the motor oil to expand. Since the seal is connected directly to the motor, the expanding oil is allowed to enter the seal during normal operation. During shutdowns, the oil in the motor shrinks because of the decreased motor temperature and part of it previously stored in the seal is sucked back to the motor. The bag and labyrinth help accomplish this function. [1]

2. The seal equalizes the inside pressure with the surrounding annulus pressure. This equalization keeps well fluid from leaking into the motor. Well fluids which get into the motor can cause dielectric failure and loss of lubrication. Well fluid is allowed to migrate into the top chamber of the seal section equalizing the pressure within the unit. The well fluid is contained in the upper chamber and cannot migrate into lower chambers. [6]

3. It isolates the clean motor oil from well fluids. The seal contains several shaft seals that prevent well fluid from leaking down the shaft. A rubber bladder acts as a positive barrier to the well fluid. The labyrinth chambers separate motor oil and well fluids based on the difference in densities between the two liquids. [6]

4. It provides the mechanical connection between the motor and the pump, and absorbs the thrust load produced by the pump. This is accomplished by the thrust bearing, which must be capable of overcoming the net axial force acting on the pump shaft. [1]

3.1.5 Power Cable

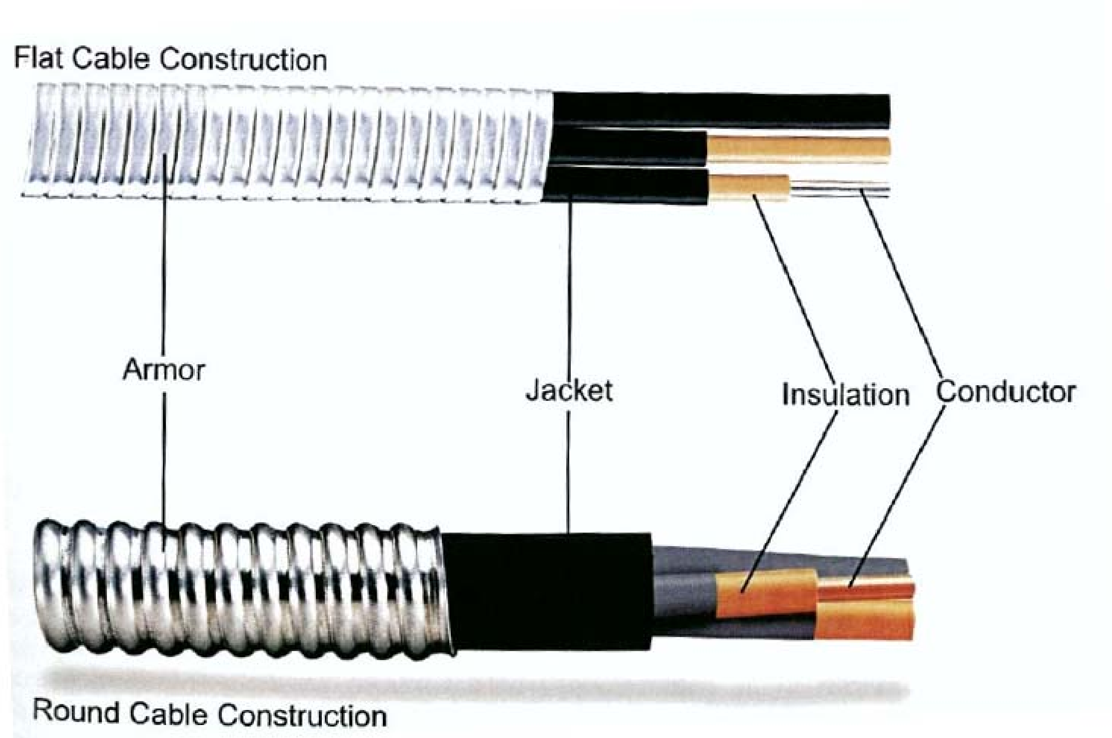

Figure 16. Cable Cutaway. [6]

The ESP cable transfer electric power from the surface power source to the motor and act as the critical link between surface and the down-hole equipment. The cable is a three phase electric cable that runs down the production tubing. ESP cables operate in harsh conditions and must meet the following requirements:

- they must have a small diameter so they fit in the casing annulus
- they must retain their dielectric properties when subjected to hot liquids and gasses
- they must be well protected against mechanical damage. [1]

ESP cables can be made in both round and flat configurations. Most cables are composed of the following components: (See Figure 16.)

- Three copper conductors carrying the AC current
- Individual insulation of each conductor preventing short circuits and current leakage
- A jacket which provides the structural strength and protection, and prevents contact of the insulations with the downhole equipment.
- A metal armor providing improved mechanical protection. [1]

Because of the very unforgiving conditions in oil wells, cables must be durable in a wide range of conditions. Long cable life is best achieved by preventing decompression, and mechanical damage resulting in durable long lasting ESP cables. [6]

3.1.6 Surface Equipment

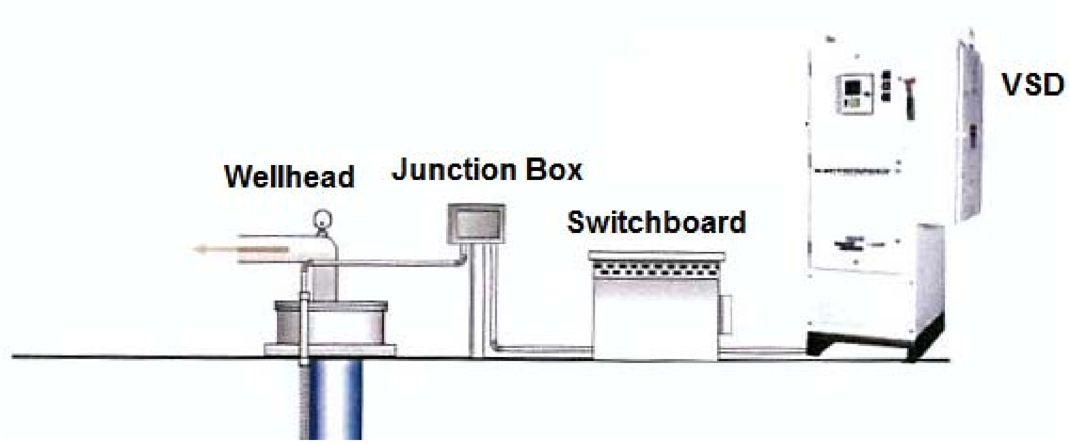

Figure 17. ESP surface equipment. [6]

Wellhead
Wellheads used in ESP installations are designed to carry the weight of the down-hole equipment and maintain annular control. They need to seal off the tubing and the electric cable. [1]

Junction Box
As seen in Figure 17 the electric cable from the well is joined with the cable from the switchboard inside the junction box. The junction box performs the following functions:

- It provides a connection point between the down-hole and the surface electric cables
- It provides a vent to the atmosphere for gas that migrates up the power cable. This prevents fire or explosions.
- It provides an accessible test point for electrical checks of the submerged equipment. [1]

Switchboard
The control centre in an ESP system is called the switchboard, which controls the operation of the entire system. The main functions is to provide a controlled on/off switching of the ESP equipment and monitor and record operating parameters.

In addition the switchboard can protect the ESP equipment from downhole or surface problems. Downhole problems a switchboard can prevent are:

- Overload of the motor
- Under-load of the motor
- Unbalanced currents

Surface problems a switchboard can prevent are:

- Too high or too low input voltages
- Voltage unbalance
- Lightning strikes. [1]

Transformers
Available electrical power on a oil platform is usually at 6000 volts or higher. Since ESP motors operate at voltages between 250 and 4000 volts, transformers must be used to provide the right voltage level [1]. ESP transformers are oil-filled, self-cooling units and are available in either three single phase units or a single three phase configurations [6].

Variable Speed Drive (VSD)

Figure 18. Variable Speed Drive. [10]

VSD makes it possible to vary ESP performance by controlling the speed of the motor. If this is achieved it can have the following main benefits:
- better control of motor temperature
- improve gas handling
- adjust to changing well conditions . [6]

Normally in oil fields power supply voltage is quite high and the required surface voltages should be individually adjusted on each well. If a VSD unit is used, the general arrangement looks like the schematic showed in Figure 19. Here, the VSD provides the required frequency. Step-down and step-up transformers do the necessary adjustments to ensure that required voltage is available to the ESP.

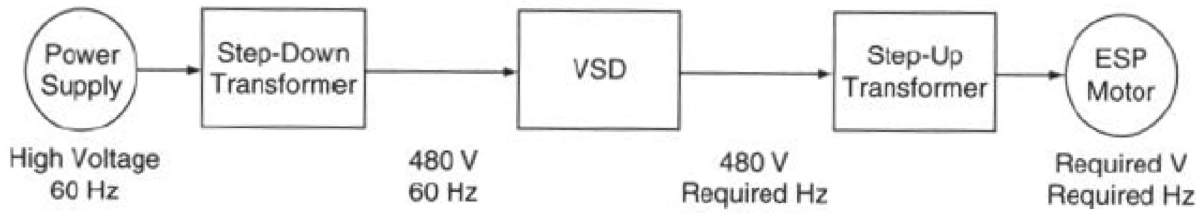

Figure 19. Electric power arrangement of a typical ESP well. [1]

The VSD converts the input frequency (normally 60 Hz) into the required operating frequency. VSD contain the following main components:

- Rectifier section. Converts AC voltage and current into a DC voltage and current.
- DC control section. Provides a smooth DC waveform to the next section.
- Inverter section. Converts the DC voltage back to an AC voltage at a determined frequency. [1]

VSD is widely accepted as an important tool to ensure operational flexibility of ESP systems. VSD are commonplace in oil wells where down-hole conditions are subject to changes (applies to most oil wells) [2].

3.1.7 Miscellaneous Down-hole Equipment

In addition to the equipment and their components described so far, proper operation of an ESP requires several other down-hole equipments that will briefly be described in this section.

Motor lead extension (MLE) is the part of the power cable that runs outside the submerged ESP components down to the motor terminals. This section of the cable is normally flat since it is restricted space between annulus and the equipment. The upper end of the MLE is spliced to the main cable. The MLE operates in a very harsh environment because of restricted space, high mechanical stresses and temperatures involved. The heat load is at a maximum at the head of the motor, thus this is where cable temperature is greatest. Because of this the MLE usually are replaced every time a cable is reused.

A check valve is placed a couple of joints above the pump to maintain a full liquid column in the tubing string during shutdowns. The check valve prevents that fluids are leaking from the tubing down through the pump when the system is shut-off. If fluids flow backwards through the pump it can cause severe damage when the pump is started again.

A drain valve is installed right above the check valve, and prevents that a wet tubing string gets pulled. The drain valve contains a break-off plug that after being sheared, opens a hole in the tubing which liquid can flow through to the well bottom.

Centralizers are used to ensure proper cooling and to prevent rubbing of the power cable against the casing by centre the ESP in the wellbore. Centralizers are very useful in deviated wells where the ESP tends to stick to one side of the casing. They also prevent damage of the coating applied to the outside of the ESP equipment.

A down-hole sensor is normally installed below the motor, and has the required measuring devices to continuously monitor important parameters. Transducers send signals to the surface through the power cable. Modern down-hole sensors use very accurate transducers: strain gauges for pressure and resistive thermal devices for temperature [1]. More information regarding ESP surveillance are given in chapter 5.5

3.1.8 Pump Hydraulics

Head

The pressure delivered by the pump is called head, and can be measured in meters or bars. Each pump stage in the ESP pump creates a certain "head", this can be multiplied with the number of stages to determine the total head the pump delivers. The flow rate a certain pump can deliver depends on the rotation of the impellers, stage design, the dynamic head the pump is operating against, and fluid properties. Figure 20 shows a pump curve of one pump stage for a 60 Hz ESP pump. One can see the recommended operating range to the pump among other pump characteristics.

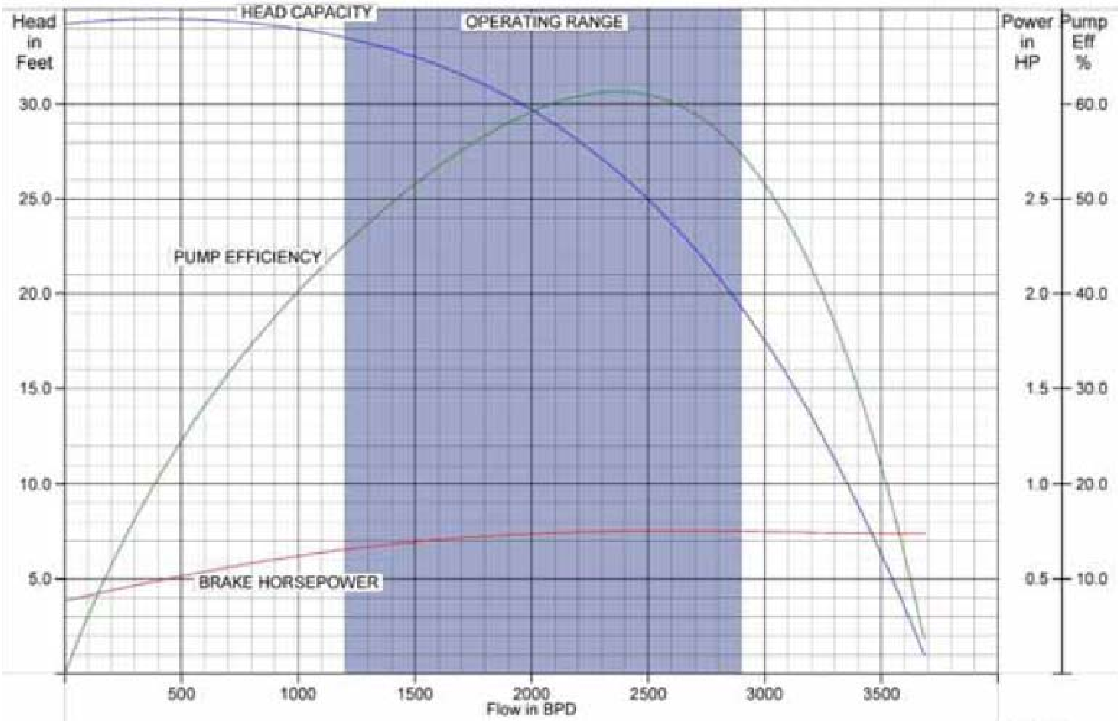

Figure 20. Pump Curve. [6]

Pump Curve

The pump curve, also called performance curve, indicates the relationship between the head developed by the pump and flow capacity through the pump. Performance characteristics showed in a pump curve are (see Figure 20):

- Operating range
- Head capacity [feet]
- Pump efficiency [%]
- Pump effect [BHP]

Generally we can say that when the capacity increases, the head decreases. The pump can develop its highest head when there is no flow through the pump; which is, when the discharge valve is closed. The pump effect curve is plotted based on the actual performance test data. This is the actual effect in BHP required by the pump, to deliver the hydraulic requirement.

Pump Thrust

Pump thrust is a description of the forces acting on the pump components when fluid is flowing through it. Pump thrust consists of two components; shaft thrust and hydraulic thrust. Total pump thrust is the net of these forces.

Hydraulic Thrust

The total hydraulic thrust consists of two components; an up and down-thrust component. Up-thrust is created by the velocity of the fluid as it passes through the impeller. Down-thrust is created by the pressure generated by the pump stage. The net of these two components make up the total hydraulic thrust. Fluid characteristics, such as viscosity have an impact on hydraulic thrust. When the ESP pump is in operation, the fluid pumped circulates on top and below the impeller shrouds. Figure 21 illustrates how the pressure from the fluid acts on the upper and lower shrouds. The cross sectional area on the upper shroud is largest, which results in a net force acting downwards, this force is called down-thrust.

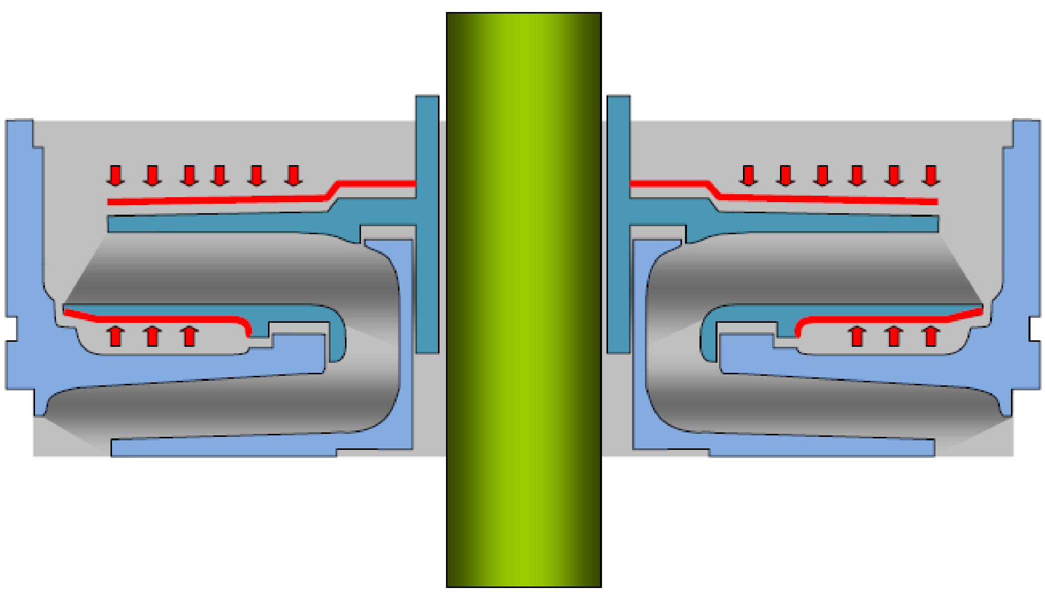

Figure 21. Forces acting on impeller. [6]

Up-thrust is the force occurring from the velocity of the fluid as it flows through the stage. When operating the pump within the recommended range, the down-thrust force is greater than the up-thrust force.

Shaft Thrust

There are two places where thrust can be produced in a pump.

1. The first is produced by fluid pressures (P_T & P_B) on the impeller surfaces (see Figure 22). The fluid pressure on top of the impeller area (A_T) generates a down-ward force on the impeller. The fluid pressure on the bottom area (A_B) and the momentum force (F_M) of the fluid produces an upward force. The sum of these is called the impeller thrust force (F_I).

   $F_I = P_T\ A_T - P_B\ A_B - F_M$

   P_T & P_B are largest at shut-in conditions (zero flow) and decline as flow rate is increased. F_M is zero at shut-in and increases to its maximum value at the wide open flow.

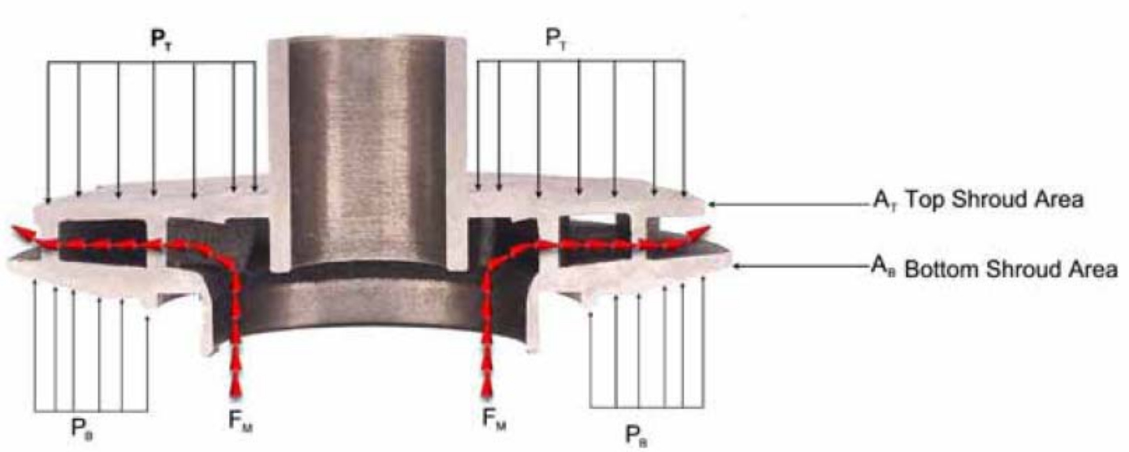

Figure 22. Cut-away picture of impeller. [6]

2. The second is produced by fluid pressures acting on the end of the shaft (see Figure 23) and is called shaft thrust (F_S). In this case, the pressure (P_D) produced by the pump minus pump inlet pressure (P_I) acting on the shaft area (A_S) produces a downward force (F_S).

   $F_S = (P_D - P_I)A_S$ [6].

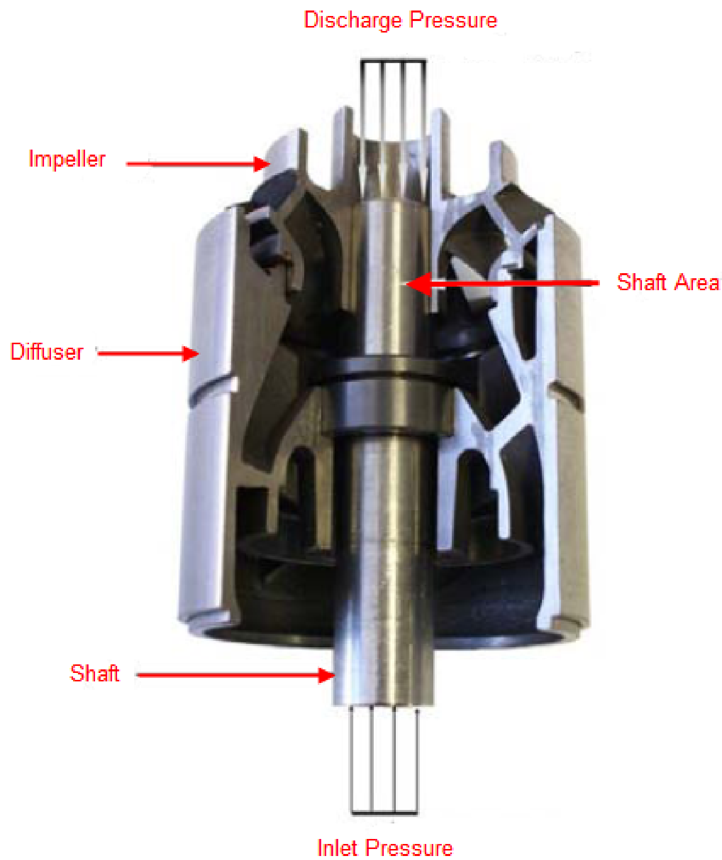

Figure 23- Cut-away picture of pump stage. [6]

3.2 Special ESP designs

Main applications include onshore and offshore operations in wells that require high head or high flow rate, pumping water, oil or a mixture. Produced liquid rates are normally between 2.000 and 20.000 BPD, heavily decreasing with well depth which range from 300 to 3.000 m [1].

3.2.1 Shrouded ESP

The submerged motor is normally placed above the perforations so that well fluids flow past and cools the motor. As the well pressure is reduced, the ESP can be set below the perforations. A motor shroud is then used to direct the well fluids past the motor.

The shroud has to cover the pump intake, seal section and motor. As seen in Figure 24 well fluids are directed from the perforations downwards along the outside diameter of the shroud. Further the fluids is routed to the pump intake through the annular space between motor outside diameter and the inside diameter.

A motor shroud can also be applied for gas separation purposes when placed below the perforations. The separations process uses the natural buoyancy of the fluids for separation. According to Baker Hughes the production of many gas wells has been significantly increased by implementing shrouded ESP configurations to pump down the water level. [6]

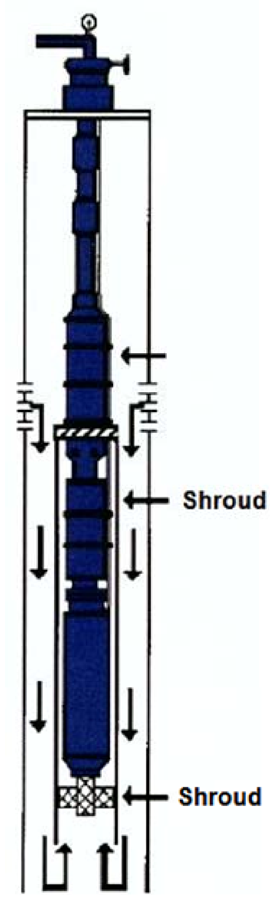

Figure 24. Shroud configuration. [2]

3.2.2 Steam Assisted Gravity Drainage (SAGD)

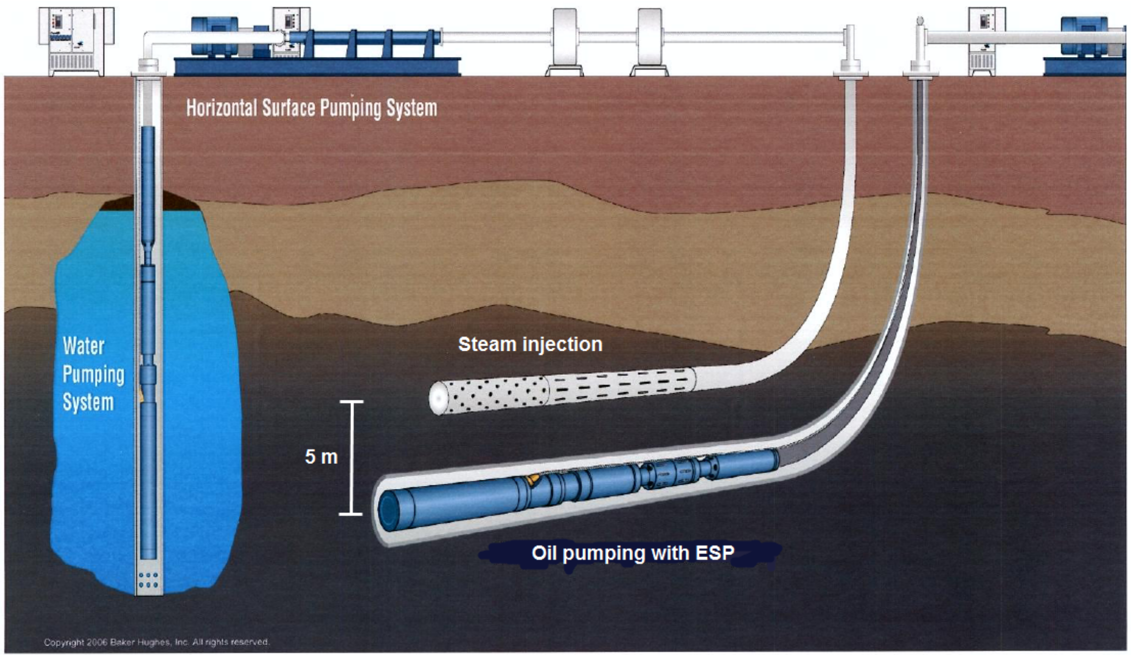

Figure 25. SAGD production. [6]

SAGD is a thermal oil recovery process which is used to produce bitumen. Bitumen is very heavy oil which is extremely viscous and does not flow naturally [11]. Effective production of bitumen requires specialized thermal recovery techniques. ESP systems have proven effective for this task with the SAGD method, which include drilling of two horizontal wells a few meters from each other, see Figure 25. The upper well injects steam, which heats up the bitumen and reduces its viscosity. The well fluids then and starts to flow down into the production well and are being pumped to surface with the ESP. The well fluid can be more than 200˚C and causes therefore a challenge of cooling the ESP motor [12]. Statoil are involved with such SAGD projects in Canada who probably held one of the world richest bitumen deposits. An ESP pump is also used for the water pumping in a SAGD process [6].

3.2.3 ESP with Deep Set Packer

Some countries, Norway included require that packers are installed on ESP installations because of regulatory policies. The packer acts as a barrier between the producing well and the surface. See Figure 26. A packer also isolates the casing above the packer from damaging wellbore fluids, and protect against cable damage due to gas saturation in a high pressure well. A packer can be equipped with an electrical feed through penetrator to provide fast and reliable cable hook-up. [6]

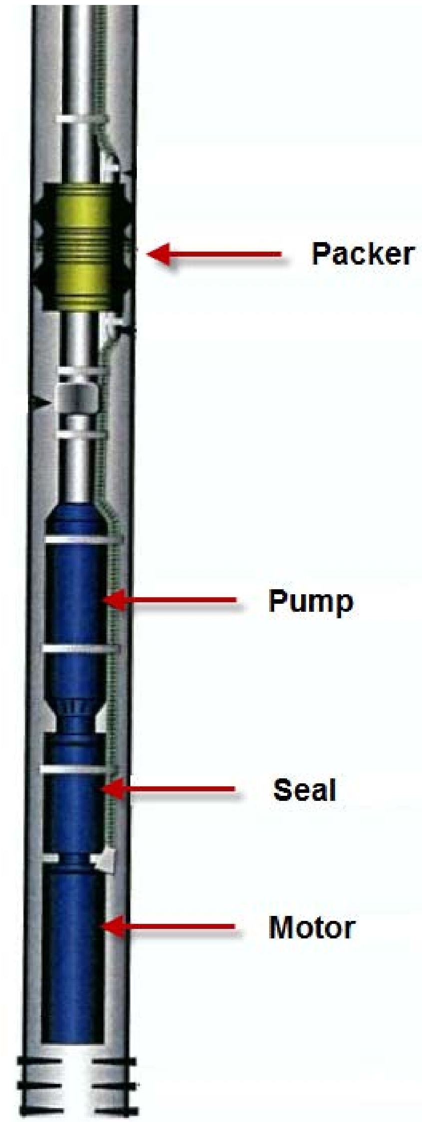

Figure 26. ESP with Packer. [6]

3.2.4 ESP with "Y" Tool

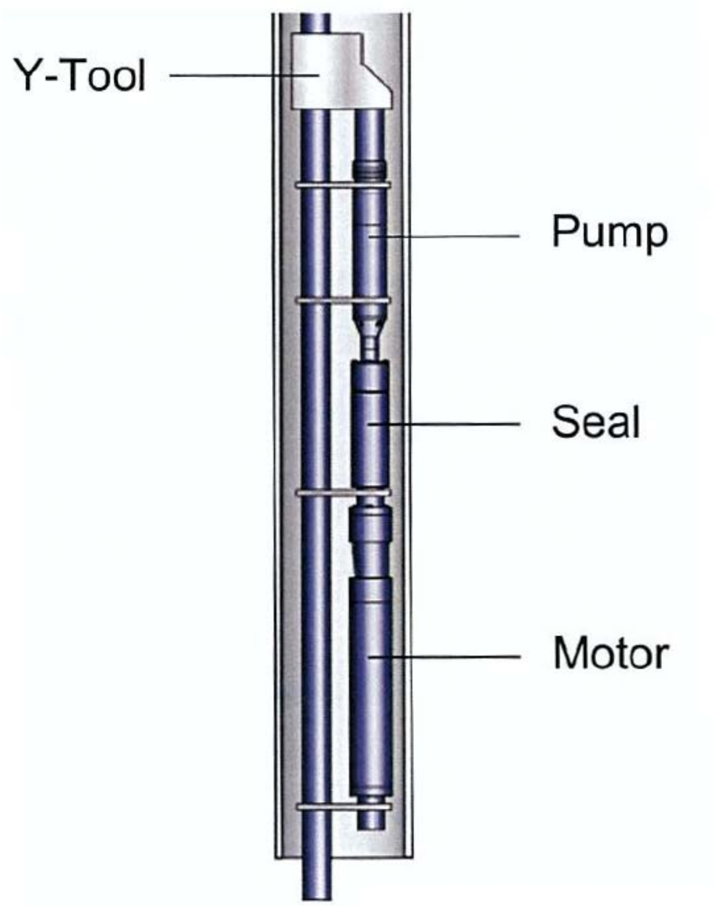

Figure 27. Y-Tool configuration. [6]

The Y-tool is a production tool with an inverted Y shape (see Figure 27) mounted at the bottom of the production tubing. One side of the Y-tool is in line with the tubing and one side is offset and contains the ESP. The straight section provides access to the wellbore below the ESP, and the following operations can be performed:

- downhole surveys with wireline or coiled tubing
- formation treatment
- well completion
- well logging. [1]

Y-tool installations have played a major role in finding and excluding excessive water or gas entry by undesirable subzone contributors. Usually a Packer and a Y-tool are used together in an ESP installation. [6].

3.2.5 Dual ESP

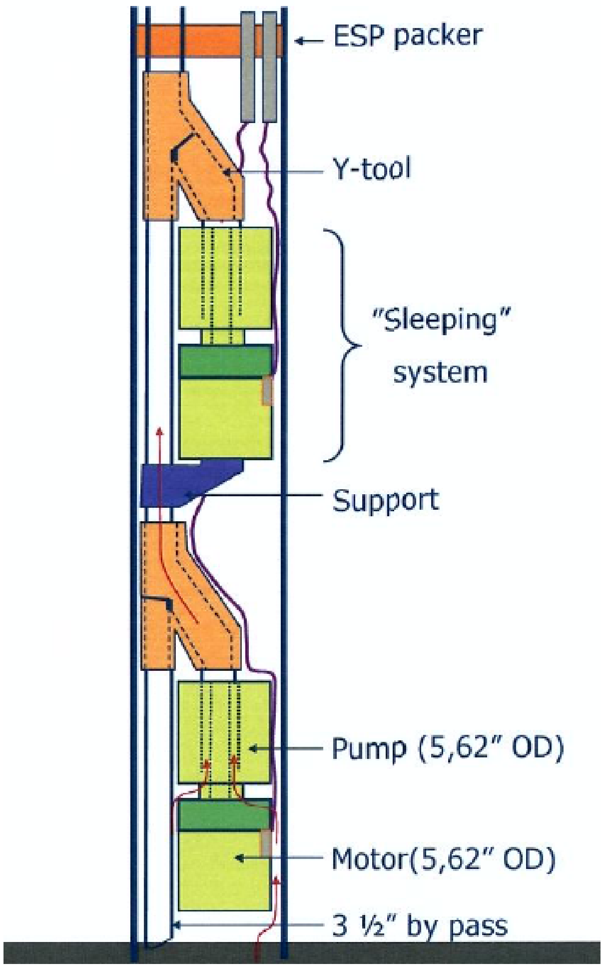

Figure 28. Dual Y-tool system. [13]

A dual ESP configuration consists of two identical pumps. Figure 28 illustrates a dual system with two Y-tools, one pump is in operation while one is in standby mode. If one pump fails the next can be started to continue production, resulting in minimal downtime. The two ESP are completely independent with duplication of all components [14]. A dual configuration provides redundancy, and hence increases system availability. Dual systems can be beneficial if pump lives is short, since work-over costs are usually very high. Work-over rigs can also be scheduled in advance while the well is still producing [15].

3.2.6 Booster Pump

ESP can be used as booster pumps, in case of long step-outs to push the well fluids from the seabed to surface. Figure 29 illustrates an ESP pump installed just below the seabed, where well fluids enter at the top and are being guided down by a shroud to enter at the bottom of the ESP. This is to provide cooling for the ESP motor before the fluids enter the ESP pump.

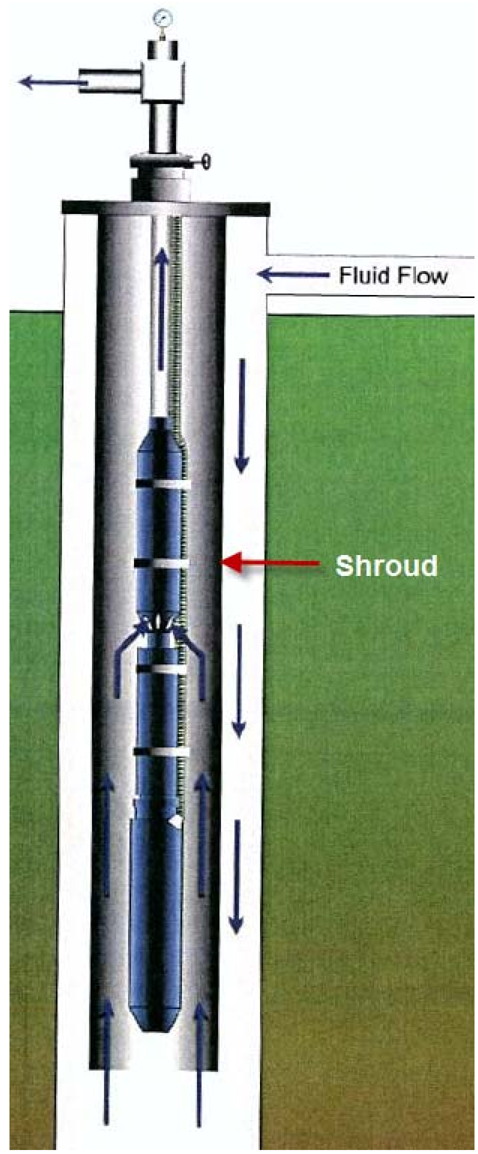

Figure 29. Booster Pump [6]

If required several booster pumps can be connected together in series or in parallel. If series connected the discharge from one pump is connected to the inlet of the next one. The pressure increase from pump to pump but the flow rate stays the same. If parallel connected, the pumps delivers the same discharge pressure and are connected to a manifold. But it's possible to achieve higher flow rates.

Booster pump are often used to add pressure to long pipelines for pumping fluids to storage or processing facilities. They can also be used in water flood project for increasing the pressure of water injection systems. [6]

4 ESP failures – degradation and influencing factors

4.1 Common ESP Failures

Electrical Failures
As will be described in section 4.2, in the majority of cases the root cause of an electrical failure is unrelated to the electrical system. However, purely electrical problems may also lead to failure. Since the electrical system is series-connected, the weakest component may be the critical factor:

- Problems with the power supply 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.
- Motor controllers are usually reliable in operation, but in extreme conditions (high/low temperatures, high level of moisture, etc.) may fail [1].

Section 4.2 shows that many electrical problems are related to short circuit. The motor can have a short circuit because of the following reasons; the oil is not clean (water break-through, dirt), large amount of noise from the transformer which result in wear of insulation and that the oil can become contaminated. Cables can have a short circuit because of electric discharge in the insulation. This is a phenomenon which is specially known at high voltages. ESP cables do not fulfil the requirements given in IEC regarding electric discharges. The reason for short circuits could be; Poor manufacturing quality to cable insulation, water break-through, too much noise from the transformer and damage on cable [16]. If the down-hole electric cable fails, it will immediately lead to system shutdown. The possible causes of cable failures are:

- Mechanical damage (crushing, cutting, etc.) during running or pulling operations
- Corrosion
- Insulation deterioration due to high temperature or wellbore gases
- Currents above the design limit can rapidly increase cable temperature leading to insulation breakdown [1].

Pump Failures
ESP pumps operate in harsh environments and are subjected to the detrimental effects of the well-stream. Pump bearings are lubricated by the fluid pumped and thus may fail much earlier than the motor bearings which are lubricated by high quality oil. Typical root causes to pump failures are listed in the following:

- Up-thrust wear occurs when the pump is operated at flow rates greater than the maximum recommended flow rate. In floating-type pumps the impellers are forced up against the diffusers, the up-thrust washers may be overloaded and it can result in a destroyed pump stage. Solids production accelerates the process.
- Down-thrust wear typically occurs at flow rates lower than the minimum recommended rate and is also accelerated by solids production. Down-thrust washers are destroyed after the pump stage is destroyed.
- Scale formation can plug or even lock pump stages [1].

More information on how solids production affects the run-life of the ESP pump is described in section 4.3.

Motor Failures
Most of the motor failures are electrical in nature. However, several other conditions may become the primary cause of motor failure:

- Overloading of the motor leads to heating of the motor's wiring; can lead to damage or burnout. Overload is usually caused by:
  - Undersized motor
  - High specific gravity of the well fluid can cause the actual total dynamic head (TDH) to be greater than assumed.
  - Pump failures that result in increased power requirement
  - Irregular (high, low, or unbalanced) voltage at the motor terminal
- Leaking protectors allow well fluids to enter the motor causing a gradual contamination of the motor oil which can lead to short circuit or motor burnout.
- Insufficient cooling. Cooling is insufficient when; flow velocity past the motor is lower than required for an efficient transfer of the generated heat from the motor. [1]

Seal Failures
Seal section failures may be attributed to the following factors:

- Broken or damaged mechanical seals in the seal section can cause well fluids to leak into the motor. This can be a result of:
  - Vibrations transmitted from a worn pump
  - Wrong equipment selection or improper installation
- The ESP main thrust bearing in the seal may fail when the pump operates in excessive up- or down-thrust mode.
- Labyrinth-type seals may fail in deviated (more than 30 degrees) well sections.

Shaft Failures
Typical shaft failures in the ESP system can be classified as follows:

- Torsional yield failure occurs when the torque limit of the shaft is exceeded. Can result in permanent deformation or a break of the shaft.
- Torsional twist in the shaft absorbs the energy during system start-up. If the shaft twists more than allowed by a connected part then that part can be damaged.
- Bearing wear mostly occurs when solids are produced when clearances in radial bearings increase. This will weaken the shaft stability and vibrations will increase.
- Lost lubrication in radial bearings of the pump can because of the metal-to-metal rubbing create increased local temperatures, and bearing materials weld together. This can lead to a damaged or broken shaft. [1]

4.2 Survey of ESP failures

A database called ESP-RIFTS has been used to create a survey of ESP failures; this database is a JIP between 13 oil companies. These participants share run life and failure information from all over the world, both onshore and offshore [17]. Before the results from the analysis are presented, an explanation of the uncertainties to the results and a specification of the query will be given. The data is collected from 13 operators, which may have different opinions to what for example installation failures imply. Maybe some think that completion failures is installation failures and vice versa. That means that we cannot be to sure that the data is entirely accurate. There are challenges of sharing ESP failure information in a common database. As stated by Alhanati, one of the main challenges in sharing ESP failure data in a database like ESP-RIFTS is to ensure data consistency. A common data set used by participants is important for establishing meaningful relationships between the types of failures observed, the utilized equipment, produced fluids, operating practices, etc. It is also important to have a common terminology and format for classifying failures. Because that will ensure that all participants have similar interpretations of a failure event, and that data collection and analyses are performed in a consistent manner. Establishing a common terminology is a challenge however, since failures are generally described in qualitative terms, strongly influenced by the experience and knowledge of the observer [18]. In order to meet these challenges the “minimum” and “general” data sets were developed to assist participants of ESP-RIFTS to collect a common set of parameters.

The Minimum Data Set is a minimum list of parameters participants must provide to be allowed to share failure information in the database [19].

The General Data Set is a desired list of parameters which gives a more detailed explanation of the failures, but is optional for participants to provide [19].

Since not all ESP-RIFTS participants provide the “general” data set, failure information is varying in quality and accuracy. The reader of this report do also have to be aware that ESP RIFTS cover only down-hole ESP equipment, surface power supply equipment, are therefore excluded [20]. There are 5 qualification levels that can be chosen when performing a query in the database.

1. **Qualified**: Records that are complete relative to the Minimum Data Set and have been qualified by C-FER and the Participant.
2. **Incomplete – No Dates or Failure Information**: Records that do not have run time (start/stop dates) information, and/or failure information (i.e. it is unknown if the ESP system failed or how long it has run).
3. **Incomplete**: Records that is largely incomplete relative to the Minimum Data Set, but that the Participant may attempt to make complete.
4. **Historical**: Incomplete historical records that will not be completed because the missing data is not readily available.
5. **Inconsistent Records**: Records that appear to be inconsistent or inaccurate and require further investigation by C-FER or the Participant in order to correct [19].

Confidence in data analysis is strongly dependent on the quality of the data collected. High quality data is defined by ISO 14224 as:

1. Complete in relation to a specification
2. Consistent with a standard set of definitions and formats
3. Accurate with respect to the actual installation that it describes

These three characteristics are checked by ESP-RIFTS as follows:

1. Completeness is checked by comparing a production period to the minimum data set. A production period with 100% of the minimum data set parameters is considered complete.

2. The consistency of the data is checked by comparing a production period with:
    - A terminology described in a common nomenclature standard, i.e. the failed component exists in the standard.
    - Other information regarding the production period, i.e. runtime.
    - Records from other production periods, i.e. that overlapping do not occur.
    - Basic laws of engineering/science, i.e. the fluid rate pumped are in the range of the specific ESP.
    - Given data values are within credible ranges, i.e. water cut is between 0 and 100%.

3. Accuracy of the data is checked by comparing a production period with the original data provided by the participant.

Feedback from participants are used to spot possible errors, participants can use the following tools to contribute to improving the quality of their data:

- **Data processing memos.** It is created a data processing memo after the data has been qualified which lists any inconsistencies and assumptions regarding the data.
- **ESP-RITS Report Card.** A report Card is generated after each data upload which indicated the level of completeness of all data in ESP-RIFTS.
- **Data Status Table.** A Data Status Table is updated after each upload which indicates at what stage the participant data are in the data processing procedure. [19]

The analysis performed in the database considered qualified information which is the most accurate one since it neglects all records that are inconsistent and incomplete. The pie charts in this chapter are based on estimated failure rates from the database. It was chosen to look at ESP with 300 – 1600 HP motors used in oil wells since this is systems that Statoil are most interested in.

The query in the database returned the following result:

Company Records	6
Division Records	8
Field Records	17
Well Records	135
Production Period Records	211

Table 1. Query Result

Table 1 shows that the analysis results is based on information from 6 companies, 8 company divisions, 17 oil fields, 135 oil wells and 211 ESP. Now we will look at the results from the analysis to get an understanding of how ESP's fail under certain circumstances. The results are illustrated with pie charts and tables which are based on the failure rate. Failure rate is by ESP-RIFTS defined as; "the total number of failures observed within a group of production periods, divided by the sum of the known runtime for all ESP systems in the same group." Production period is any individual ESP system installed in an individual well for a period of time [19].

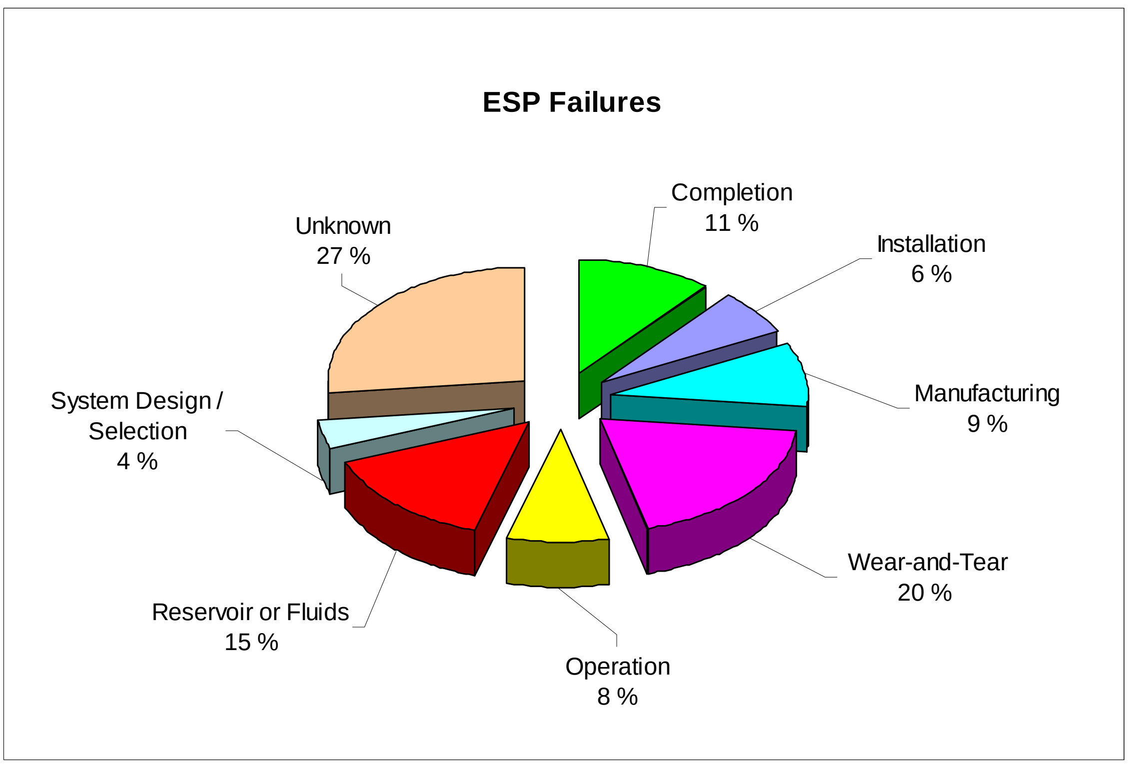

Figure 30. ESP failures.

According to Takacs the weakest link in an ESP is the electrical system. Electrical failures however often originate from a mechanical problem which is the real cause of the failure. Hence every failure should be analyzed properly and its actual cause identified [1]. Failure analysis will therefore be a central part of this chapter. As Figure 30 illustrates there are many different factors that can lead to equipment failures, this section will describe how each of these factors can lead to failures. The analysis result found that the ESP has an average MTTF of 946 days, which is approximately 2.5 year.

4.2.2 Completion (11%)

We start with completion failures which are usually related to failure of other completion components that may lead to the ESP system failure. According to ESP-RIFTS is the most common specific completion failure-cause registered in the database, failure of the sand control completion. However the failure cause can also be, for instance a casing collapse or a tubing failure resulting in damage to the power cable [20].

Examples of failure causes can be:

- Failure of perforations or liner
- Failure in the sand control system
- Wellbore completion failure

Failure of the wellbore completion can in example be: casing, tubing, packer, safety valve or liner failures [20].

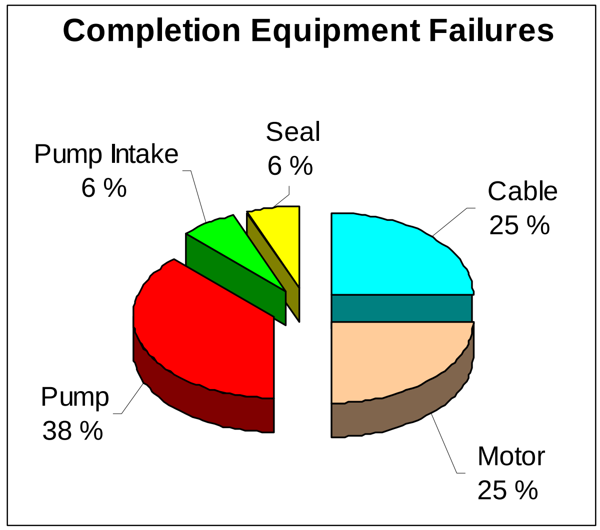

Figure 31. Completion Failures.

Primary Failed Item Major Component	Primary Failure Descriptor	Number of Records
Pump (38%)	Damaged	2
Motor (25%)	Short Circuit	2
	Phase Unbalance	1
Cable (25%)	Damaged	2
	Short Circuit	2
Seal (6%)	No data	
Pump Intake (6%)	Disconnected	1

Table 2. Completion Equipment Failures

Pump
As we see from Figure 31 the analysis results showed that the pump has the highest failure rate in completion failures. According to Table 2 is there two records on damaged pumps, both were damaged because of high sand production as a result of improper sand control completion [19]. Further explanation of the damage sand production can lead to will be given later in the reservoir and fluid section, and chapter 4.1.

Motor
From Table 2 we see that two ESP motors have got a short circuit because of a completion failure. This is according to the analysis result because the ESP went on over-current and that the motor grounded [19]. There was also recorded one motor with phase unbalance. In this case the casing collapsed and the ESP was left in the well [19].

Cable
It is recorded two cases of a damaged cable and two cases of a cable that had a short circuit. See Table 2, the damage on the cables was according to the analysis caused by a cannon protector and severe stress [19].

Pump Intake
It was found one record of a failed pump intake, see Table 2. This failure was because of failure or improper sand control system that caused problems to the shaft intake [19].

Solution
As mentioned above the most common reason for completion failures are because of faulty sand control completion. Hence, it is important to make sure that the sand control which often consists of a gravel pack is of the adequate size, is installed correctly, and are free of manufacturing problems. Since the objective of the gravel pack is to prevent intrusion of sand in the wellbore, a failure of this protection system will cause solids to enter the pump. This will as explained in section 4.3 have detrimental effect for the pump if it is not designed to handle solids.

4.2.3 Installation (6%)

Installation failures are usually related to damage to the ESP system that occurred during the system installation in the well. The most common specific installation failure cause registered in the database is improper splicing of the cable, often the splice to the wellhead penetrator [20]. As we see from Figure 32 the analysis also showed that the cable has the highest failure rate when installation failures have occurred.

Examples of failure causes can be:

- System assembly
  - Mounting of the equipment
- Well cleanout
- ESP field service
  - Testing
  - Well optimization
- Reran damaged equipment

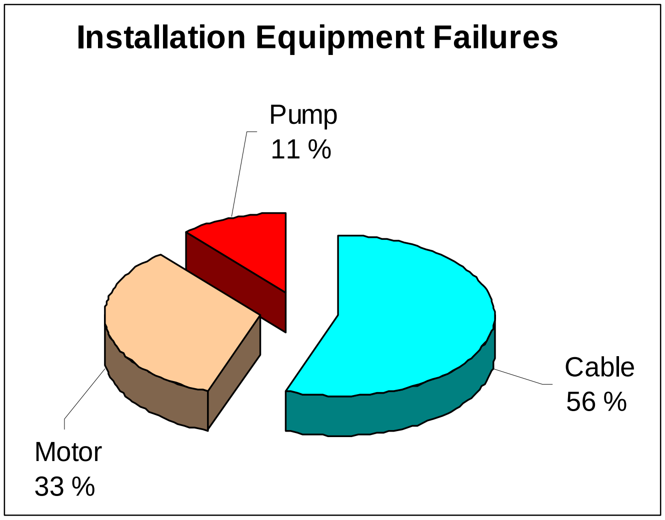

Figure 32. Installation Failures.

System assembly failures can in example include cable splices and flange connections [20].

Primary Failed Item Major Component	Primary Failure Descriptor	Number of Records
Pump (11%)	Worn	1
Motor (33%)	Short Circuit	3
Cable (56%)	Short Circuit	4
	Damaged	1

Table 3. Installation Equipment Failures

Pump
The worn pump showed in Table 3 was actually a worn pumped that was re-installed [19].

Motor
It is recorded three motors that had a short circuit, see Table 3. The analysis did not include any specific answers to why this happened, other than it happened during ESP field service [19]. As described in section 4.1 is there several factors that can cause a short circuit in the motor.

Cable
From Table 3 we see that it was recorded four cables that had a short circuit. These failures happened during ESP field service. One had problems with the electrical connector at the wellhead, one with the splice at the wellhead, one with a splice at the tubing hanger [19]. It was found no explanation of the fourth cable that had a short circuit. There was also not found any data on the damaged cable in ESP-RIFTS. See section 4.1 for description of how cable short circuits occur.

Solution
Proper installation of all components is crucial to avoid failures. It is important that everyone whose working with the installation is following a thoroughly installation procedure. According to the API standard for ESP installations, rig personnel must be aware that the equipment can easily be damaged. They should have good understanding of handling procedures and adequate supervision must be provided to ensure that the installation operation goes according to plan [21]. The standard also includes guidelines for the well service rigs. They must be equipped with the necessary installation equipment and during installation be placed over the centre of the well [21]. Since cable splicing is a major failure cause, care should be made when mounting each cable section together. In addition is proper inspection of the mounted cables important to prevent faulty installation.

4.2.4 Manufacturing (9%)

In Figure 33 we see that the cable is the component that has the highest failure rate in manufacturing failures. The most common reason for cable failure in the database is short circuit [19].

Examples of failure causes can be:

- Wrong material selection
- Improper fabrication or assembly of components
- Improper equipment testing or quality control [20].

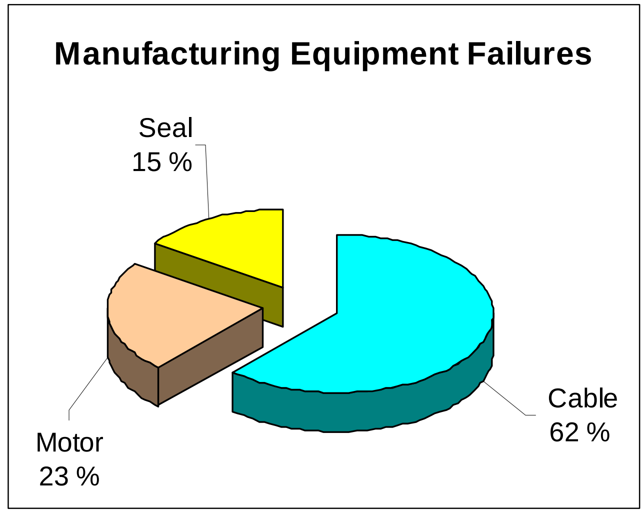

Figure 33. Manufacturing Failures.

Primary Failed Item Major Component	Primary Failure Descriptor	Number of Records
Motor (23%)	Short Circuit	4
Cable (62%)	Short Circuit	14
Seal (15%)	Contaminated	2

Table 4. Manufacturing Equipment Failures

Motor

The four short circuited motors in Table 4 failed because of fabrication problems at the vendor. In all four cases the motor grounded [19].

Cable

As seen in Table 4, there was found fourteen cases of a short circuited cable. Section 4.1 describes possible failure causes of a failed cable.

Seal

Two seals were contaminated, see Table 4. In both cases the seal was invaded and the motor grounded [19].

Solution

The analysis results showed that there were registered problems with corrosion of motor housing. This gives reason to think that it was not chosen materials that could withstand the corrosive environment and this is the reason for the failures. If a power cable has a short circuit, all the down-hole equipment has to be pulled [22]. So it is very important that ESP components have been tested for proper operation before it is installed in the well. Operation of equipment not properly checked before installation can cause premature failure due to manufacturing defects [1]. Full string test, performed both at the manufacturer and on site would also result in better quality control. Today is only full string test performed onshore, as the author is aware of.

4.2.5 Wear-and-Tear (20%)

This category is normal or expected wear-and-tear, here the failures are more equally distributed among the different equipments, see Figure 34. As will be discussed below, these failures are also electrical failures in general.

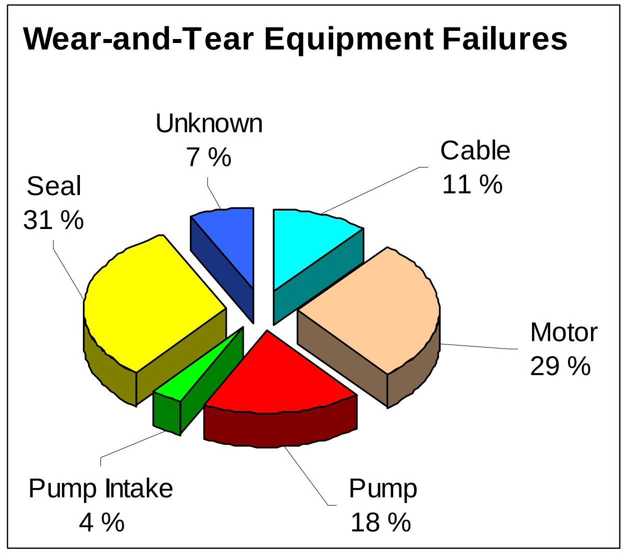

Figure 34. Wear and Tear Failures.

Primary Failed Item Major Component	Primary Failure Descriptor	Number of Records
Pump (18%)	Low Efficiency	3
	Worn	1
	Broken / Fractured	1
Motor (29%)	Short Circuit	5
	Phase Unbalance	1
	Contaminated	1
	Corroded	1
Cable (11%)	Short Circuit	2
	Low Impedance / Resistance	2
Seal (31%)	Contaminated	7
	Corroded	1
Pump Intake (4%)	Broken / Fractured	1

Table 5. Wear-and-Tear Equipment Failures

Pump
The three records of low efficiency failures showed in Table 5, was actually no failure. The pumps were pulled out of the well because they did not produce the required flow rate. The record of the worn pump showed that the pump had severe wear because of sand production [19]. See section 4.3 for further explanation of sand production.

Motor
As Table 5 shows, there are five records of motors with short circuit. Three of those short circuited because the motor grounded, the other two was unknown. Further the table shows that there was found one record on phase unbalance, contaminated and corroded motor. The motor that failed because of phase unbalance had no other data than it was contaminated with produced fluid, so it is likely to think that the ESP has been producing fluids with abrasive particles. The effect of abrasives will be discussed in the reservoir or fluids section and in section 4.3. The record that shows a contaminated motor had the following comment in the database; ESP got ground fault and failed to restart. Top & lower pump, and AGH shaft have excessive end and side play. The lower pump sleeve was missing on the stage bearing of the pump head. The intake was collapsed. The oil in the tandem protector and motor was found to be dirty. The record that shows a corroded pump had hole in the motor housing due to corrosion. [19]

Cable
It was found two records of cables that had short circuited, see Table 5. There was no explanation to why it had occurred, other than in one of the records the motor lead extension was grounded. Table 5 also shows that there are two records on cables with low impedance / resistance, but there was not given any explanation in the database to why it happened.

Seal
It was found seven records of contaminated seals and one with corrosion see Table 5. Comments to the records included:

- The cable armor shows corrosion.
- ESP motor evidence phase to ground due to water contamination.
- The ESP fails due to over-current alarm.
- Severe corrosion.
- [19]

Pump Intake
There was found one record of a broken/fractured pump intake, see Table 5. It was not found any explanation in the database to why this failure occurred.

Solution
As seen from above almost every failure has to do with sand production and corrosion. This give reason think that the materials used have not the right properties to handle the harsh environments. Further explanation on sand production is given in section 4.3.

4.2.6 Operation (8%)

The motor has absolute highest failure rate in operation failures, see Figure 35.

Examples of failure causes can be:

- Inadequate condition monitoring
- Poor operating procedure
    - For example if it is performed unnecessary many start/stop.
- Well treatment
    - Injection of inhibitors [20].

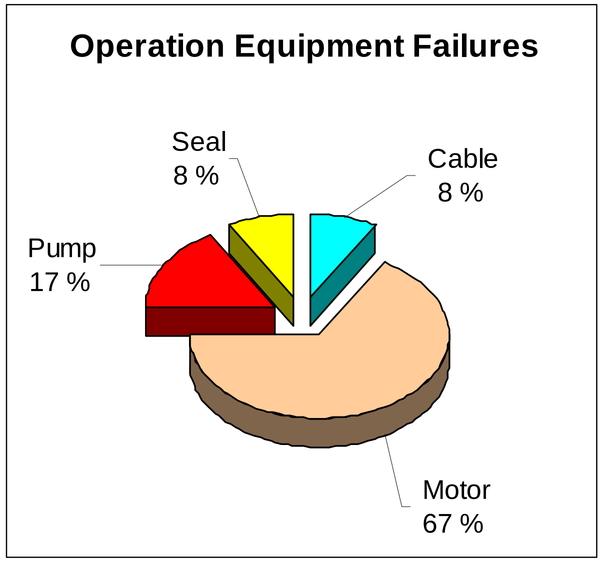

Figure 35. Operation Failures.

Primary Failed Item Major Component	Primary Failure Descriptor	Number of Records
Pump (17%)	Stuck	1
Motor (67%)	Short Circuit	3
	Phase Unbalance	1
	Corroded	4
	Unknown	1
Cable (8%)	Short Circuit	1
Seal (8%)	Corroded	1

Table 6. Operation Equipment Failures

Pump

It was recorded one incident of a stuck pump, see Table 6. The failure cause was according to ESP-RIFTS sand production because of inadequate monitoring [19].

Motor

As seen in the table there was recorded three cases of short circuited motors. Two of these failures happened because of corrosion in motor housing, the last one failed because of poor operating procedure. One motor failed because of phase unbalance as a consequence of inadequate monitoring, and four motors corroded [19].

Cable
It was recorded one cable that had short circuited because the motor lead extension grounded [19].

Seal
The table also shows one seal that corroded; the corrosion was so severe that it was a hole in the seal section [19].

Solution
As mentioned above is the motor the most affected part of the ESP, and it failed mostly because of corrosion or short circuit. However, if the corrosion of the motor housing is so severe that it corroded hole in surface, then well fluids can enter and the motor gets a short circuit. Proper material selection with respect to corrosion resistance is important to avoid such failures. Sufficient training of operators is also something one should prioritize to make sure the motor is run inside its recommended frequency range. In addition should it be paid attention to the following aspects as defined by API RP 11S.

Checks before start-up
Make certain that the flow line hook-up is completed, that all valves are of proper pressure ratings and are properly installed. All valves should be in their proper operating position. Perform electrical checks, phase-to-ground, and phase-to-phase. Phase-to-phase readings must be balanced.

System start-up
It is recommended that the tubing is filled before start-up. This means these installations must be equipped with tubing check valves and drain valves. With all checks completed, start the ESP. For control of the pump discharge rate, the pump can be started against a restricted choke, but should not be started against a closed choke or valve.

Gathering of operating data
Accurate operating data:

- Is required to monitor the system under normal operating conditions
- Will provide information that will be useful in troubleshooting the well under abnormal operating conditions.
- Will be useful in accurate resizing of the equipment, if required

Analyzing operating data
Analysis of operating data must consider both permanent well installation data (i.e., tubing size and length, casing size, perforation depth, fluid characteristics, etc.) as well as production test data. Once the pump is in the well and operating, it should be analyzed to determine if it is functioning properly.

Fore more information see API RP 11S, 2008.

4.2.7 Reservoir or Fluids (15%)

Well conditions failures will result in most damage to the pump, as we see in Figure 36.

Examples of failure causes can be:

- Presence of asphaltene in the well fluid
- Bottom-hole temperature above approximately 80˚C [23]
- Free gas in well fluid
- Sand content in well fluid
- Scale formation
- Paraffin in well fluid
- High water cut in the well fluid
- Low or no inflow of fluids in the well

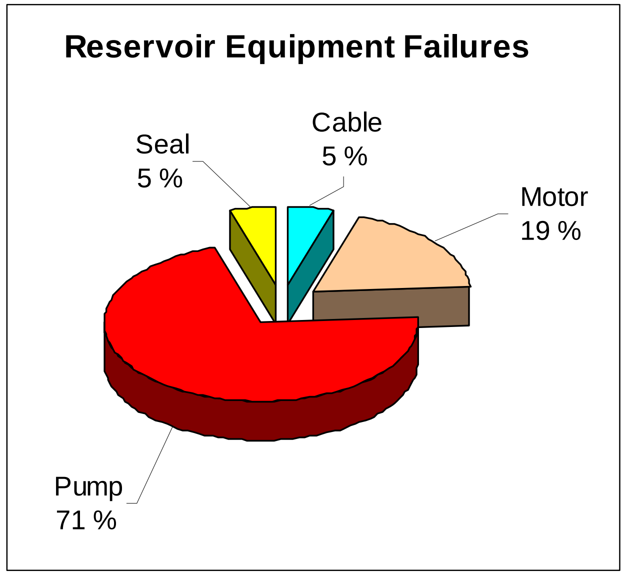

Figure 36. Well Condition Failures.

Primary Failed Item Major Component	Primary Failure Descriptor	Number of Records
Pump (71%)	Stuck	11
	Plugged	2
	Eroded / Pressure Washed	2
	Broken Fractured	1
Motor (19%)	Leaking	1
	Phase Unbalance	1
Cable (5%)	Broken / Fractured	1
Seal (5%)	Corroded	2

Table 7. Reservoir or Fluids Equipment Failures

Pump

All the prime failure descriptors showed in Table 7 have according to ESP RIFTS failed because of sand in the fluid. As mentioned in chapter three, the ESP pump can have either “fixed” or “floating” impellers. According to Wilson examples of sand wear seen in pumps with “floating” impellers can be radial bearing wear, thrust bearing wear and erosional wear [24]. See section 4.3 for more information about solids production.

Motor
As Table 7 there is one record on phase unbalance and one record of a leaking motor. There is no description in the database to why the phase unbalance. The leaking motor had severe corrosion [19].

Cable
The table shows one record of a broken/fractured cable. The failure cause is according to the database sand production. It was also found corrosion and scale in the lower seal and motors [19].

Seal
It is recorded two seals that corroded, but no further explanation was given in the database. See Table 7.

Solution
A brief explanation on how to protect the down-hole equipment against harsh well conditions is given in the following:

- **Corrosion** on pump and motor housings can have detrimental effects. To protect against this one must choose proper housing materials and utilize surface coatings.

- **Scale** deposits on motor housings will obstruct the motor in getting enough cooling effect from the well fluid, which can result in motor burn out. Special coatings applied to the motor housing are the usual solution. Scale formation in pump stages reduces the liquid rate and can result in a plugged pump.

- **Sand or abrasive production** problems can be avoided or reduced by selecting a proper pump type and materials.

- **High well temperatures** can have detrimental effect on the motor, which can overheat and burn out. Here it is important to select a pump that can handle the well temperature.

- **Formation gas** can lead to gas lock in the pump. By utilize proper gas handlers or separators one will be able to a certain degree withstands this threat. Free gas can also migrate into the power cable; to avoid this one can select a cable with proper insulation.

- **High viscosity** fluids will require higher motor effect because of higher specific gravities. In addition will the efficiency of the pump decrease while the friction in the tubing increases, which also leads to increased motor requirement. To make sure the ESP system is capable of deliver the desired flow rate one must select motor and pumps with the proper specifications. [1]

4.2.8 System Design / Selection (4%)

Examples of failure causes can be:

- Wrong selection of ESP equipment for the given application.
    - Inadequate pump flow or head capacity.
    - Motor power capacity
    - Use of low-grade materials
- Improper system design / selection, including use of improper data or errors in calculations.[20]

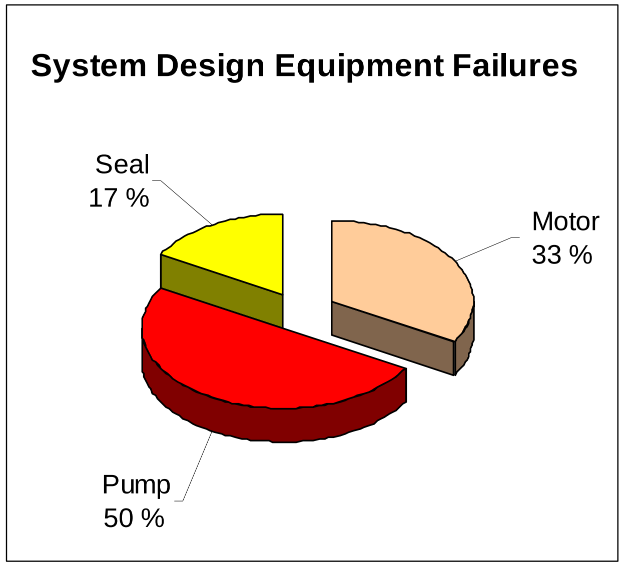

Figure 37. System Design Failures.

Primary Failed Item Major Component	Primary Failure Descriptor	Number of Records
Pump (50%)	Leaking	2
	Stuck	1
Motor (33%)	Short Circuit	2
Seal (17%)	Broken / Fractured	1

Table 8. System Design / Selection Equipment Failures

Pump

The pump has the absolute highest failure rate in system design failures, see Figure 37. It was found two records on pumps that had start leaking; this was because the housings had corroded. See Table 8. One record on a stuck pump was also identified; this was because of scale formation [19].

Motor

The table shows that there was found two records of motors with short circuit. In the comment for the first motor it was mentioned that sand production was suspected and that erosion damage was found. This can mean that the erosional wear have caused well fluid to enter the motor through holes in the motor housing which caused a short circuit. It was found dirty oil in the second motor that failed. This gives reason to think that the short circuit happened because of contaminated oil as a result of a seal section that was operating insufficient. [19]

Seal
The table shows that one seal was found broken/fractured; according to the database was it the seal shaft that was broken [19].

Solution

It is important to conduct a proper installation design to prevent against system design / selection failures. Initial sizing of ESP equipment is the most important factor to succeed in optimizing the run life of ESP. This implies that the unit must be operating within the recommended operation window with regards to flow range, temperature, GOR etc. If the ESP is run outside its operation window, pump wear is accelerated leading to premature pump and motor failures. Reliable data on the well productivity and accurate information on fluid properties like viscosity, GOR, and temperature is important to be able to select proper equipment. [1]

Component	Subcomponent		
Pump	- Base/Intake - Coupling - Diffusers - Head/Discharge - Housing	- Impellers - O-Rings - Screen - Shaft - Shaft Support Bearings	- Snap-Rings - Thrust Washers
Motor	- Base - Coupling - Filter - Head - Housing	- Oil - O-Rings - Rotor Bearing - Rotors - Shaft	- Stators - Thrust Bearing - Varnish
Cable	- Main Power Cable - Motor Lead Extension - Packer Penetrator	- Pigtail - Pothead Connector	- Wellhead Penetrator - Splices
Seal	- Bag Chamber - Base - Coupling - Head - Housing	- Labyrinth Chamber - Mechanical Seals - Oil - O-Rings - Relief Valves	- Shaft - Thrust Bearing
Pump Intake	- Base - Coupling - Diffusers - Discharge Ports/Screen - Head - Housing	- Impellers - Inducer Section - Intake Ports/Screen - O-Rings - Radial Bearings - Separation Section/Rotor	- Shaft - Coupling - Diffusers - Discharge Ports/Screen - Head - Housing
Other	- Down-hole Sensors - Shroud		

Table 9. Possible Failed Items. [20]

Note that the Pump Intake also includes gas separator and gas handler [20].

4.3 Solids production

Since both the analysis results from wear and tear and well conditions showed that sand production is causing a lot of failures it was decided to look further into the consequences of sand production. Since an analysis with qualification level "qualified" returned very few results, it was decided to do a bit more open research to get sufficient with data. However the analysis neglected all records that were inconsistent, historical, and incomplete with regards to runtime or failure information. So this analysis can still be found trustworthy, the query returned the following results:

Severe solids	
Company Records	5
Division Records	12
Field Records	15
Well Records	166
Production Period Records	283

Table 10. Query result, severe solids

None solids	
Company Records	8
Division Records	12
Field Records	25
Well Records	284
Production Period Records	663

Table 11. Query result, none solids

As we see from the results in Table 10 and Table 11, the query resulted in more than twice as many results for ESP with none solids production than pumps with severe, so there is reason to think that the results for none solids is the most accurate. As we see from figure Figure 38, when an ESP is subjected to severe sand production the pump has a significant higher failure rate than the other ESP equipment. If we compare this result with ESP that have not produced solids at all (see Figure 39) then we see that it is a more "normal" distribution in the failure rate among the different equipments. From this we can conclude with that solids have a bad influence on pump life time. Figure 40 show also that pumps utilized in wells with severe solids have a considerable lower MTTF than ESP that has not been exposed to solids. The difference is 489 days is favor to ESP with no solids production.

The definition of severe solids is by C-FER Technolgies defined as; "substantial amount of substance or problems observed, strongly believed to be or known to be a problem [20].

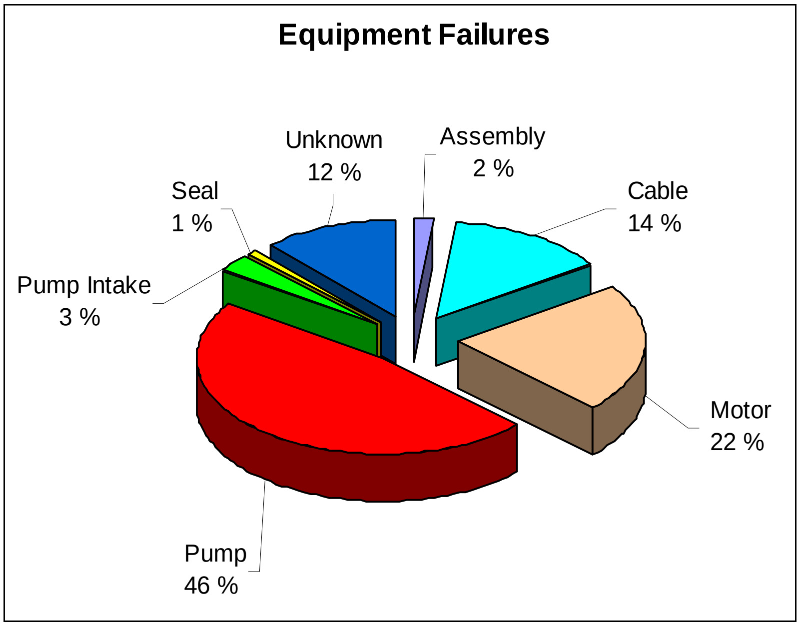

Figure 38. Severe sand

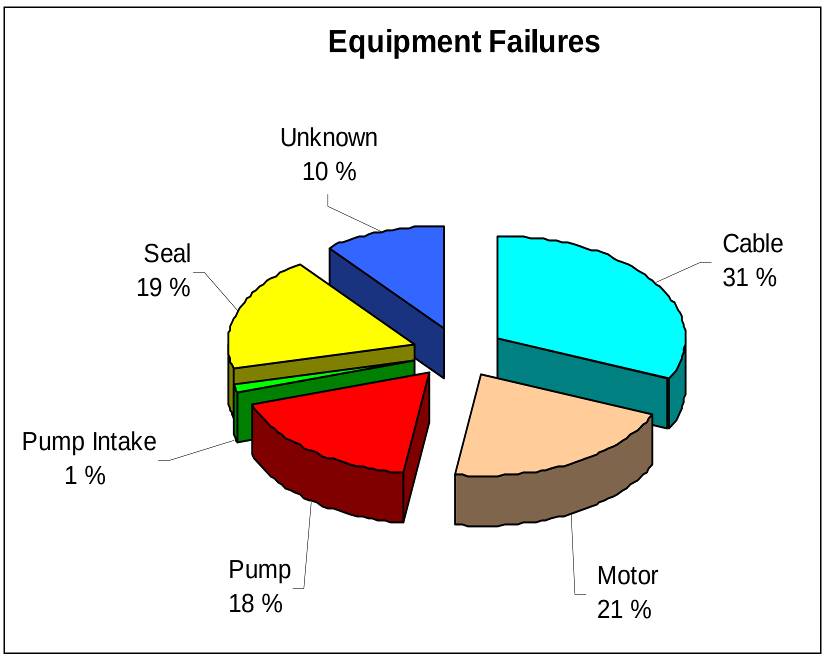

Figure 39. None sand

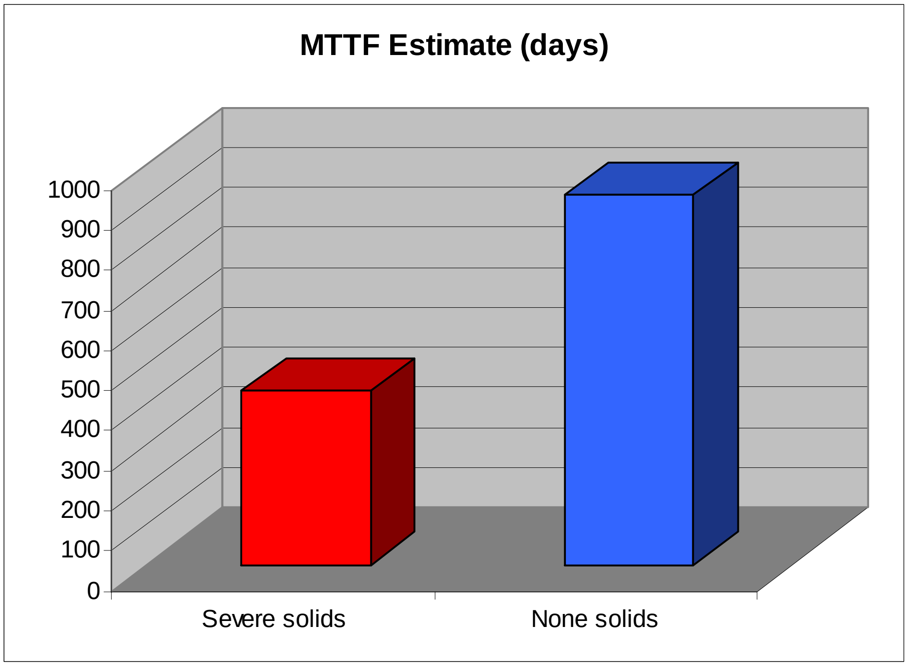

Figure 40. MTTF comparison

The ESP pump is as explained in chapter 2 as a high speed rotating centrifugal/axial pump with stationary and rotating parts that are lubricated by the produced fluid. Since well fluids often contain solid particles, the abrasive action of the particles can easily damage moving parts in the pump. Metal loss caused by abrasion and/or erosion in the pump stage or in bearings, may lead to critical damage to the pump. In addition to the ESP pump can also gas separators be damaged by sand, therefore is it necessary to install special separators in sandy wells which are designed to deal with solid particles. Most of the abrasion problems occur because of production of sand (quartz, SiO_2) along with well fluids. However solids other than sand, like iron sulphide, calcium carbonate, etc may cause abrasion damage in pumps. Characteristic features of oil wells with regards to sand production can be summed up as follows:

- Sand production normally starts at high well rates
- Sand production increases after water breakthrough to the well occurs
- Change in flow rate results in increased sand production

ESP is often utilized in wells with high fluid rates and low bottom-hole pressures, often used in water-flood operations, and frequently involve cyclic operations. Since these application areas match the main causes listed above, are ESP according to Takacs particular prone to sand problems [1]. Sand particles flowing through the pump causes removal of metal particles from different parts of the pump, we can classify the damage into:

- Erosion
- Abrasion

Erosion occurs on a metal surface when it is hit by abrasive material particles contained in the fluid. Abrasion occurs when sliding between two surfaces includes particles between the surfaces [25]. The magnitude of damage caused by these types of wear depends on several factors: size and shape of the solid particles, and the difference in hardness between the affected material and that of the solid particles. The hardness of the attacked material as compared to the abrasive particles has a direct effect on the magnitude of the damage made. Since abrasive particles will not cut anything harder than itself, materials in pumps used for abrasive service must be harder than the abrasive contained in the well fluid. Thus, it is recommended to choose materials that can withstand abrasive wear, and/or use protective coating on exposed areas.

Figure 41 shows the MOHS scale used for comparing the hardness of materials, a material with a higher number will damage any other material with a lower number. We can see that sand is harder than steel and nickel but it cannot damage the harder materials like zirconia and tungsten carbide. Materials of other solids normally present in well fluids (iron sulphate, calcium carbonate) are much softer than sand; this is why sand is referred to as the key abrasive substance in oil wells [1]. The shape and size of the particles are also of importance and affect the damage caused by abrasion or erosion. Abrasion wear is highest when the size of the solid particles is comparable to the clearances in the ESP pump. Wear due to erosion on the other hand, is proportional to particle size and the square of particle velocity. When it comes to particle shape, are rough irregularly shaped particles causing more damage than rounded, smooth ones of similar size.

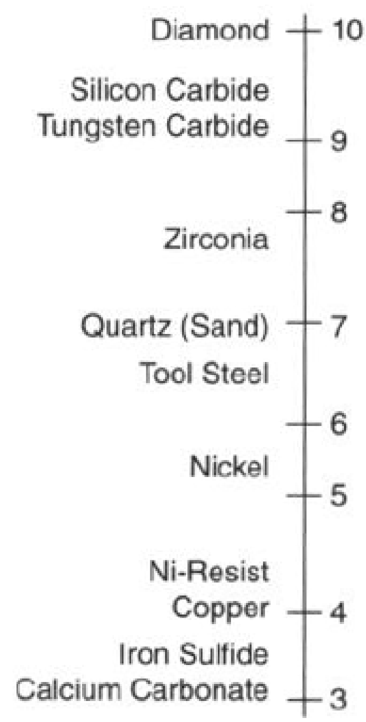

Figure 41.MOHS Scale. [1]

Sand damage in ESP pumps can be classified by severity in the following categories:

1. Erosion in pump stage
2. Abrasion in radial bearings
3. Abrasion in thrust washers and thrust bearings

Erosion in pump stages occur when solid particles is striking the metal surfaces (similar to sandblasting). The magnitude of wear is proportional to the square of flow velocity since the damaging potential of solids is related to its kinetic energy. As mentioned earlier large and rough particles is causing more wear than small and round ones. Solids do most of the damage at points where a change of flow direction takes place, which means at the entrances to the diffuser and impeller. Erosion can also occur around the balance ring in the diffuser, where solids are moved by viscous drag in the stationary fluid. Pump stages can be considerably eroded but it seldom leads to failure because the pump will fail for other reasons long before its completely eroded.

Abrasion in radial bearings (radial wear) is caused by abrasion in the pump radial bearings. Normally, are simple journal bearings providing support of the pump shaft. These bearings have fixed radial clearances; the clearance gap is depending on pump design and machining tolerances. The clearances are usually large enough for sand particles to enter the space between the bearing and the journal. The largest particles, after entering the clearance gap, are crushed and scratch against the bearing surfaces, while the smaller ones may be taken by the flowing fluid without even touching the bearings. Radial wear will cause the clearances in bearings and sleeves to grow; this will result in radial un-stability of the shaft. The shaft starts to rotate eccentrically, causing increased side loads in the bearings which lead to accelerate wear. Further, because of the high axial loads acting on the shaft, it starts to buckle which induces vibrations along the shaft. Such vibrations are critical to the pump and can destroy it in a very short time. Vibrations will be transmitted down to the seal section, failure of the seal results in well fluids entering the motor causing short circuit. Radial wear heavily increase with increasing flow rates because of larger amount of sand particles carried by the fluid.

Abrasion in thrust bearings (axial wear) occurs on the thrust washers and the mating surface in the pump stage. If solids are caught between the washers it can result in worn washers or in metal-to-metal rubbing of the impeller and diffuser. Pumps with fixed impellers completely eliminate contact between impellers and diffusers and are therefore well protected against abrasive damage. In case of pumps with floating impellers, the axial force is absorbed by the thrust washers of the impellers that “float” freely. The clearance gap is therefore not fixed but varies with the magnitude of thrust and fluid viscosity. A pump is in the down-thrust condition when it is in the recommended capacity range, and the clearances are normally too small for large quantities of sand to enter the gap. If a pump is in the up-thrust condition however, then the clearance between the washers are big enough for large grains of sand to enter, this may lead to worn-out washers and/or abraded stages.

Solutions

The most logical countermove against solids is to make sure that materials in critical points in pump stages are harder than sand, since sand is the most aggressive abrasive in well fluids, see Figure 41**.** However, soft materials like rubber can also withstand solids damage in journal bearings. Because of the resilient nature of rubber, solids entering the clearance between the bearing and the journal will not remove any material. In addition cannot solid particles imbed in the bearing because of the rubber deflection properties.

Erosion in pump stages
Manufacturers can as minimum use materials like an alloy containing 18% nickel (Ni-Resist) in impellers and diffusers, instead of the cheap and softer gray iron to minimize erosion wear. But Tungsten Carbide is preferred since it has hardness close to diamond, and has therefore high wear resistance. Tungsten carbide has also high fracture strength, high thermal conductivity and high resistance to corrosion [39]. Erosion resistant coatings can also be applied on exposed surfaces.

Abrasion in radial bearings (radial wear)
Also to combat this category of wear is material selection an important part. As mentioned earlier are axial abrasion to be found in thrust bearings and the thrust washers of floater pumps. The main thrust bearing is usually situated in the protector and have thrust runners and shoes consisting of very hard materials (i.e. Tungsten Carbide). The wear of the washers (in floater pumps) can in addition to proper material selection be reduced by increasing their surface area. As stated earlier compression pumps are not affected with radial wear.

Abrasion in thrust washers and thrust bearings (axial wear)
Radial abrasion is causing the most severe category of sand damage and will therefore be given a more detailed explanation in how to reduce it. Radial bearings with a special resilient (i.e. rubber) bushing pressed into the diffuser bore, placed at regular intervals in the pump can be applied to decrease radial wear. The resilient bearing has longitudinal grooves on its inside surface where solids are washed into and removed by the pump flow. The special bearings should be placed with as short interval as possible to make sure that the pump shaft maintains its radial stability.

To reduce axial wear hardening of wearing surfaces can be applied. Special inserts made of materials of great hardness so that radial and axial wear are minimized, are applied to ensure radial and axial stabilization. However abrasion resistant material are very expansive, so instead of having all pump stages in abrasion resistant materials, a solution can be to have only a few resistant stages at the top and bottom of the pump.

One will normally think that use of extremely hard metals like tungsten carbide and ceramics will increase the abrasion resistance of pump parts. However the application of such materials in pump bearings has proved to be unsuccessful because they are very brittle and are easily damaged if loaded at one point or on a line. Journal bearings are installed by press fitting into their housing, this method causes line loadings which can result in failure if brittle materials are used. Journal bearings for ESP pumps have therefore an improvement potential in order to be able to facilitate the utilization of hard materials.

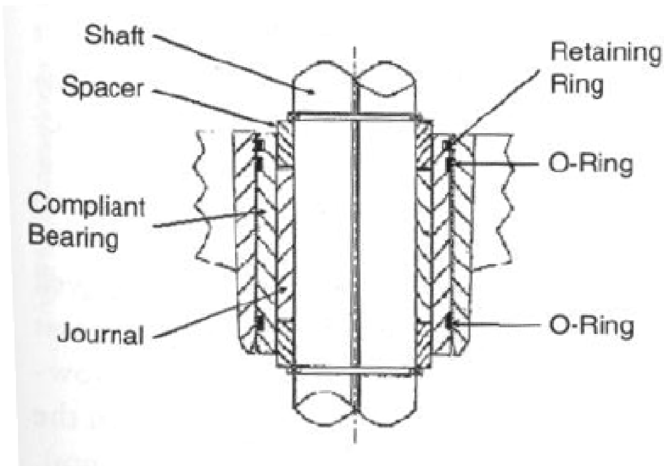

Figure 42. Compliant Bearing. [1]

But there is one solution that may compensate for this problem. Figure 42 illustrates a compliant mounted bearing which can be used together with hard metals. The bearing is mounted into the housing so a fluid chamber is formed by the O-rings. In conjunction with elastic O-rings is the chamber acting as a vibration and shock damper. Point or line loading of the bearing is avoided since this method allows the bearing to find its best running position. This is why a compliant mounted bearing can be used together with extremely hard materials. A preferred material selection of compliant bearings is Tumgsten Carbide as described above. Takacs states that radial instability is the main mechanical failure cause of ESP pumps, and run life can be significant improved if compliant bearings are used. A pump specifically designed to be abrasive resistant is illustrated in Figure 43. Here several compliant bearings are mounted along the shaft, including the head and base bearings. As mentioned earlier; it is a advantage to mount the bearings as close as possible along the shaft to maintain the radial stability. That means it is recommended to mount one bearing in each pump stage, the total number of bearings then depends on the amount of stages.

Other ESP equipment than pumps are also affected by solids in the well fluid. Gas separators are also very vulnerable to abrasive wear because of the great centrifugal forces occurring in the separation chamber. The solid particles is hitting the separator housing with significant speed and is causing the corrosion coating to disappear which result in corrosion. A solution to avoid this is to manufacture the vulnerable separator parts with abrasive resistant materials. [1]

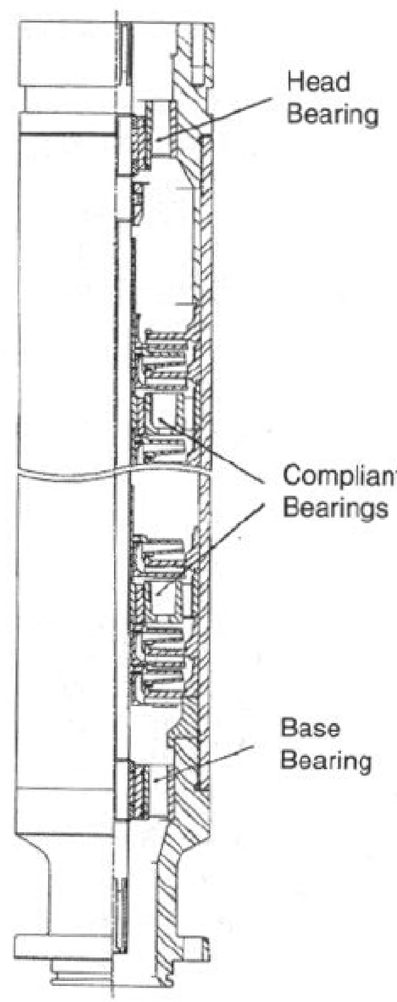

Figure 43. Abrasive resistant pump. [1]

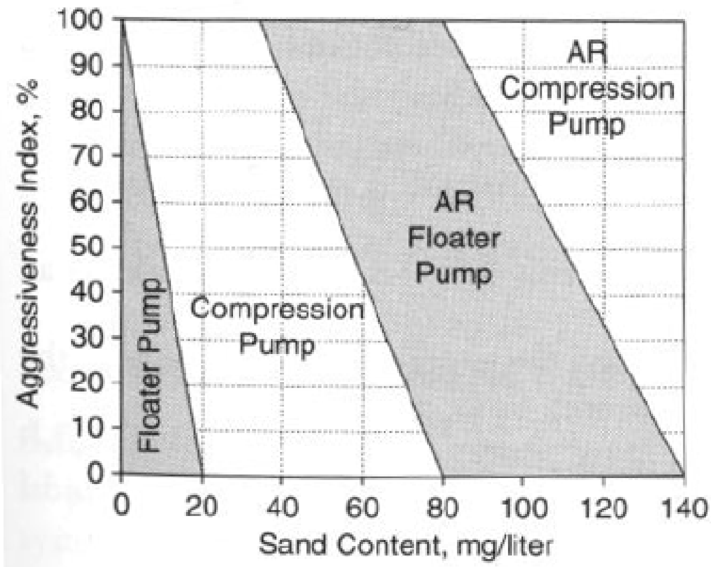

Figure 44. Pump Selection for Abrasive Application. [1]

Figure 44 shows for what sand concentration applications normal compression pumps (fixed impellers), abrasive resistant floater pumps, and abrasive resistant compression pumps is recommended. Dependent on the abrasive aggressiveness, the following general rules can be used to select the proper pump:

- Floating impeller pumps are not recommended if the well fluid contain sand
- Under mildly abrasive conditions, can pumps with fixed impellers be applied without any special features.
- Under aggressive well conditions, can abrasion resistant floating pumps be applied
- Extremely aggressive conditions require the use of abrasion resistant pumps with fixed impellers.

Since an ESP system is placed at the bottom of the production string, it can be sucked large quantities of san into the pump intake at startup. Therefore special care should be conducted when performing a startup in an abrasive well. Further should pumps with floating impellers always be operated in the down-thrust region, and filters or screens must be utilized. [1]

Vendor solutions to sand problems
Common for all major vendors is that they offer solutions that can help combat solids problems by using special bearings, coating, and special materials for vulnerable parts. Pumps with fixed impellers and filters to prevent solids from entering the pump. However Lea & Mokhatab states that operations are easier if solids can be transported through the pump so it is not necessary to stop production to clean the filters [26].

Wood Group ESP can offer optional resistant/radially stabilized pumps for abrasive applications, and different types of coatings. The coating can protect against abrasion as well as scale formation.

Baker Hughes Centrilift offer different options related to sand handling:

- SND Stabilized Normal Duty. Hard bearing in the head and base, in addition to one in the middle.
- SHD Stabilized Heavy Duty. Hard bearing in the head and base, and hard bearings spaced at standard l/d ratio. This is radial support only.
- SSD Stabilized Severe Duty. Hard bearing in the head and base, in addition to a T-flanged sleeve bearing. The T-bearing is in a module for radial and in the diffuser for mixed flow.
- SXD Stabilized Extreme Duty. Derived from the ODIS SP (Super Sand Pump) for radial stages or for mixed flow.

These options have been continued with a new pump design that is more abrasive resistant. The new Centurion pump line incorporates a larger stage opening that help preventing solids and scale from bridging the pump discharge.

Schlumberger REDA offers pumps with several modifications, such as harder material for the pump materials that contain more nickel and chrome. They also offers harder hub diffusers and end shaft journal bearings.

Weatherford states that they can provide special pumps for harsh well conditions. [26]

4.4 Vibration in ESP systems

Since the ESP down-hole equipment have an extremely shaft length - to – diameter ratio; including a motor, seal, gas separator, and pump all connected by a small diameter shaft. In addition, does every one of these components include parts rotating at high speeds, is there a large potential for vibrations to occur [27]. Vibrations originating from one ESP component are transmitted across the system via the shafts connected to each other [1].

Modes of vibration

Vibration modes in an ESP system can according to API 11S8 be:

- Axial
- Lateral
- Torsional

Or a combination of all three [27].

Axial vibrations are affecting the pump, motor and seal section thrust bearings [27].

Lateral vibrations are vibrations occurring sideways with respect to the length of the ESP and affect mostly the radial pump bearings. This kind of vibration can lead to huge consequences, since the long slender shafts in the ESP are very sensitive to radial stability. If stability is lost, it can lead to an eventual failure. [27]

Torsional vibrations can be a problem during start-up and when changing pump speed. This kind of vibration can lead to twisting of the shaft [27].

Sources of vibration

Mass Unbalance
Vibrations caused by mass unbalance may be due to:

- Rotating parts wit a non-homogeneous material. Can include blowholes in castings, inclusions in rolled or forged materials, slag inclusions or variations in material density.
- Rotating parts with unsymmetrical configurations. Can include dissymmetry due to core shifts in casting or rough surfaces on forging.
- Eccentricity in shafts or bearings. Sources of eccentricity can be:
    - Journals not circular or concentric to shaft
    - Bent or bowed shafts
    - Clearances of rotating parts can allow eccentricities
    - Thermal expansion. [27]

Misalignment
Misalignment can result in large axial vibrations. Two types of misalignment can occur; angular and offset misalignment. Angular misalignment is when the centre lines of two shafts meet at an angle. Offset misalignment is when the shafts centrelines are parallel but displaced

from each other. If couplings or shaft bearings are misaligned, it can result in vibrations perpendicular to the shaft (transverse vibration). An axial mode of vibration can occur if flexible couplings have angular misalignment. The longer the shafts are the larger possibility for such vibrations. According to API 11S8, will misalignment occur in both radial and axial directions. If the amplitude of axial vibrations is larger than 50% of the highest radial vibration, then it is reason to think that we have misalignment or a bent shaft. [27]

Flow Induced
The flow through an ESP system can cause vibrations in the pump. The cause of this type of vibration is usually turbulence; however the turbulence will be minimal if the pump is run inside its operation window. Multiphase flow and non-symmetrical fluid passages in the pump can induce hydraulic imbalance. Also, when pumping gassy or highly viscous fluids this problem is known to occur. [27]

Journal Bearing Oil Whirl
Oil Whirl can in lightly loaded journal bearings cause vibrations [27].

Bearing Rotation
If the journal bearings are not properly secured, it may lead to bearing rotation which causes vibration [27].

Mechanical Rub
Contact between the rotating and stationary parts can induce vibrations [27].

ESP are also sensitive to electrical effects like power instability and changing pump speeds which may lead to vibrations [1].

<u>Controlling vibration</u>

According to API 11S8 is there two main categories that vibration control can be grouped into: reduction at the source and reduction of the response.

Reduction at the source

- Balance rotating masses. Vibration occurring from unbalanced rotating components can often be reduced by balancing.
- Balance magnetic forces. Proper design and fabrication of the stator and rotor will minimize the vibrations occurring in the electric motor.
- Control of component clearances. Vibrations occurring when components scratch each other can be reduced by avoiding excessive bearing clearances and by ensuring that dimensions of parts are within acceptable tolerances.
- Control straightness of shafts. Straight rotating shafts will prevent these kinds of vibrations form occurring.

Reduction of the response

- Alteration of Natural Frequency. Resonance which can have a bad influence on the vibration can be avoided by changing the natural frequency of the system. This can be done by using a variable speed drive to operate at a frequency that is not corresponding to a critical speed. Baker Hughes states that the lower the natural frequency the lower the vibration levels [6].
- Additional Damping. The vibrations occurring form a resonance is strongly related to the amount of damping employed. Different techniques can be utilized to increase the amount of damping. However the additional damping decreases the ESP efficiency [27]. According to Baker Hughes has ESP normally high damping due to the fluid being pumped and the motor fluid in the motor and seal [6].

ESP parts	Typical frequencies (relative to rotating speed)	Probable causes
Shafts and rotors	1 or 2 x rpm	Bent shaft
All rotating parts	1 x rpm	Mass / hydraulic unbalance or off-center rotor.
Couplings, shafts, bearings	1 to 2 x rpm, sometimes 3 x rpm	Misaligned coupling and/or shaft bearing.
Sleeve bearing	<1/2 x rpm	Oil whirl in lightly loaded bearings. More prominent in seal chamber.
Anti-friction bearing	>5 x rpm	Excessive friction, poor lubrication, too tight fit.
Mechanical rubbing	1/3 or 1/2 x rpm	Contact between stationary and rotating surfaces
Journal bearing rotation	1/2 x rpm	Journal rotating with shaft
Motor	1 x rpm	Eccentric rotor

Table 12. Vibration analysis of ESP. [27]

Different failure cause can be identified by observing the frequency of the vibrations that occur. Table 12 shows typical frequencies relative to the rotating speed of the ESP and the likely causes of the problem [27]. The higher speed an ESP is operating at the higher the vibration will be because of unbalance. The forces due to unbalance are proportional to the operating frequency squared. This is something one has to be aware of when operating ESP with variable speed drive. [6]

Critical speeds exists in ESP systems, these are either torsional or lateral. If possible, an ESP should not be operated near the critical speed for an extended period of time. This problem can often occur during start-up or when the ESP is operated over a wide speed range. [27]

Measurement of vibration

Accelerometers
Accelerometers are very linear in an amplitude sense, meaning they have a very large dynamic range. They are relatively insensitive to temperature and magnetic influences, however excessive heat can cause damage. Accelerometers are typically used for measuring vibrations with high frequency (10-12000 Hz). Accelerometers are usually usable up to about 1/3 of its natural frequency. Records above this frequency will be accentuated by the resonant response, but can be used if the effect is taken into consideration. It is important that the vibration path from the source to the accelerometer is as short as possible when installing an accelerometer. [28]

Proximity probes
The proximity probe, also called; displacement transducer or eddy current probe, requires a signal conditioning amplifier to generate an output voltage proportional to the distance between the probe end and the shaft. Proximity probes operate on a magnetic principle, and are therefore sensitive to magnetic anomalies in the shaft. Thus it's important to assure that the shaft is not magnetized to assure that the output signal is not contaminated. It is also important to be aware of that it measures relative displacement between the bearing and the journal, and not total vibration level of the shaft or the housing. The frequency response extends from approximately (0 – 1000 Hz) [28]. Proximity probes which can be mounted within the motor housing to measure the relative displacement between the shaft and the housing, are normally not utilized in ESP's because it is difficult to install them [27]

Velocity probes
Velocity probes are made up of a coil of wire and a magnet so arranged that if the housing is moved, the magnet tends to remain stationary due to its inertia. Because of a relatively motion between the magnetic field and the coil a current is being induces which is proportional to the velocity of motion. The probe therefore produces a signal directly proportional to vibration velocity. The unit is self generating and needs no conditioning electronics in order to operate, and it also has low electrical output impedance making it relatively insensitive to noise induction. However in spite of these advantages the velocity probe has also several disadvantages; it is heavy and complex and have poor frequency response (10 – 1.000 Hz). A low frequency resonant system is made up by the spring and the magnet with a natural frequency of approximately 10 Hz. To avoid a large peak in the response, this resonance needs to be highly damped. An issue is that the damping is temperature sensitive, which causes the frequency response and phase response to be temperature dependent. [28]

The surveillance method practiced for ESP vibrations today are accelerometers placed in a motor gauge unit (MGU) below the motor housing (see Figure 45). The accelerometers measure vibration in the radial and tangential direction showed in Figure 46. Radial is the direction from the accelerometer to the centre of the shaft, and tangential is 90 degrees from radial, tangent to the shaft [28]. Further information on vibration surveillance are given in chapter 5.5.

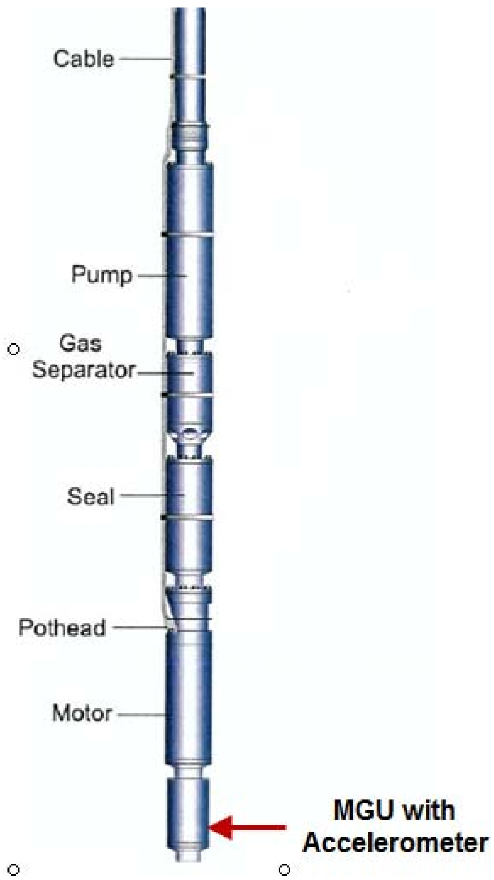

Figure 45. ESP with MGU. [6]

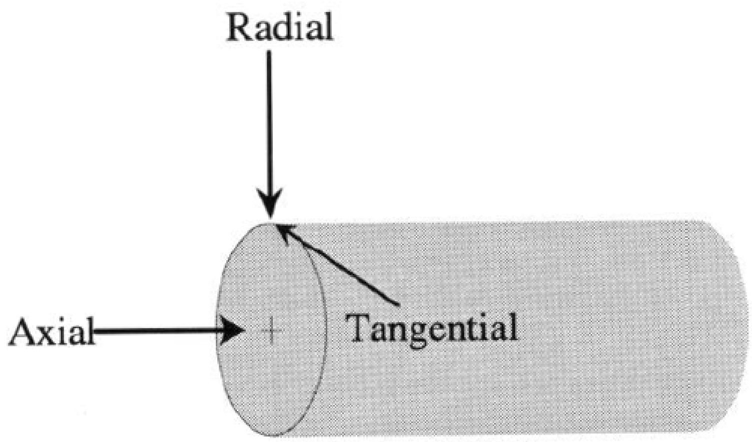

Figure 46. Alignment of vibration axes. [28]

Pump
It can be an idea to conduct vibration tests along with the pump acceptance test recommended in API RP-11S2. A minimum requirement is that measurements should be taken at the mid-point on the housing, top radial bearing, and bottom radial bearing location. It's important to note that pump rate should be held constant while measurements are being taken.

Gas Separator
A minimum requirement is that measurements should be taken at the mid-point on the housing, top radial bearing, and bottom radial bearing location.

Seal Section
A minimum requirement is that measurements should be taken at the mid-point on the housing, top radial bearing, and bottom radial bearing location.

Motor
A minimum requirement is that measurements should be taken at the mid-point on the housing, top radial bearing, and bottom radial bearing location. [27]

5 Best Practice for the Peregrino Field

This chapter will describe how to perform an ESP sizing with focus on the mechanical equipment. Best practices for pump selection, surveillance and life cycle costs will be covered.

5.1 Introduction to Peregrino

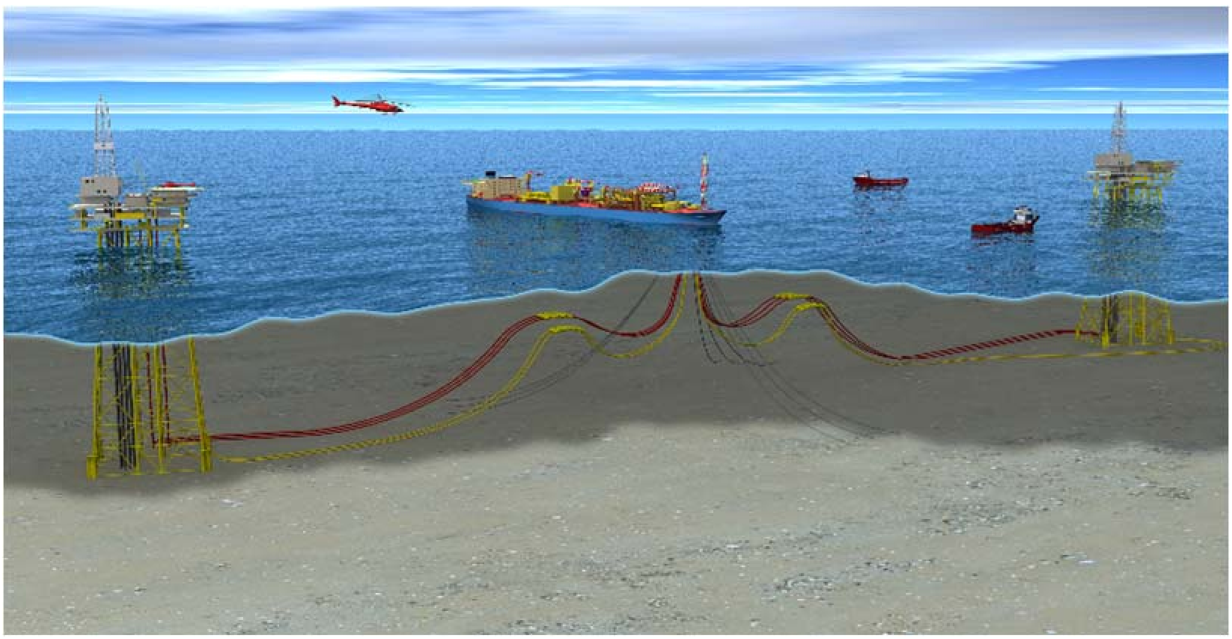

Figure 47. Peregrino Field. [29]

The Peregrino field development consists of two wellhead platforms tied back to an FPSO via dual flow lines, see Figure 47. The field is located 8 km offshore Cabo Frio in Brazil and will have first oil in the end of 2010. The oil in the reservoir is heavy and viscous with very low GOR and bubble point pressure. It will be utilized down-hole ESP for artificial lift, and surface booster pumps at the platforms. All produced water will be re-injected for pressure support [29].

Field information

- 300-600 million barrels of oil reserves, API 14
- Production capacity: 100.000 BPD
- Water depth: 100-120 m
- Production wells: 30 horizontal wells about 2.500 m beneath the seal level
- 7 wells for water injection [30].

A study was performed by Hydro (before the merger of Hydro and Statoil) to evaluate different methods of artificial lift for the Peregrino field. The study evaluated:

- Electric Submersible Pumps
- Hydraulic Submersible Pumps
- Hydraulic jet pumps
- Gas lift

The study conclusion was that ESP was the recommended method since it has high lift capacity and is energy efficient. HSP and jet pumps were neglected since they have a very high power demand because of high amount of power fluid circulation. Gas lift was neglected since the crude in Peregrino is under-saturated and has low GOR. [29]

5.1.1 Operational Challenges

Statoil requested the contractors to consider three different production ranges to propose the best solution for:

Short Well: 2.000 to 5.000 BFPD
Medium Well: 5.000 to 14.000 BFPD
Long Well: 14.000 to 20.000 BFPD

In this report, medium well production range is considered.

Medium Well ESP Design Parameters		
Oil Gravity	14	˚API
Specific Gravity of Water	1,05	(100,000 ppm)
Gas Oil Ratio	70	scf/STB
Oil Bubble point	48 (700)	bar (psi)
Productivity Index (PI)	6	BPD/Psi
Reservoir Temperature	78,6 (173,5)	˚C (deg. F)
Wellhead Pressure	6,9 (100)	bar (psi)
Water Cut	0%, 30%, 50% and 95%	
Expected Production rate	5000 - 14000	BPD
Initial Reservoir Pressure	231 (3352)	bar (psi)
Datum	2340 (7678)	m (ft)
Pump Setting Depth	1960 (6431)	m (ft)
Production casing	9 5/8" #47 lb/ft	
Production Tubing	5 1/2" #17 lb/ft	

Table 13. Process data. [31]

Heavy oil and high flow-rates is the major challenges for the design of the ESP for Peregrino. As seen in Table 13 has the oil a gravity of 14˚API. There is an inverse relationship between API gravity and density, the higher the density the lower API gravity. Oil with API below 22˚

is termed heavy; therefore can we say that the peregrine oil is heavy [32]. Table 13 also shows that the desired production range is very broad (5000 BPD to 14000 BPD), the maximum desired range of 14000 BPD will require a large capacity ESP. It is also expected initial sand production, challenges with sand production is explained in chapter 4.3.

5.1.2 Technical Challenges

This section outlines the technical requirements Statoil claimed from contractors in the tender process for Peregrino.

ESP Pump

The pump had to have the following properties:

- Abrasion resistant
- Corrosion resistant
- Withstand some sand production
- Withstand some free gas
- Able to handle scale inhibitor injection
- Able to handle emulsion inhibitor injection
- Able to handle HCI acid, Calacid, EDTA type scale dissolvers

Testing of equipment should be in accordance with contractor's highest standards of internal developed test procedures and relevant API standards, also ensuring that the following is covered:

- API 11 S2 table 2: Test acceptance criteria should be within +-2% for head and flow rate within recommended operating range.
- Vibration severity should not exceed the values of "New equipment smooth vibration severity" of table C2 in API RP 11 S8 during acceptance test.
- Rotating elements should be balanced to ISO 1940, minimum grade 2,5.

All pump stages had to be of metallic construction. Impellers and diffusers had to be free of manufacturing defects that could cause premature failures and vibrations that could cause stage and or shaft wear. Pumps should have abrasion resistant shaft stabilization features to maximize anticipated operating life. The well completion had to include sand control equipment. The pump had to be capable of operating via a Variable Speed Controller (VSC) from 30Hz to 70Hz operating frequency without any detrimental effects or significant reduction in pump life.

Pump performance criteria for the recommended operating range of the pumps had to be as follows:

- Head capacity shall not exceed +/-5% of published curves
- Horsepower shall not exceed +/-5% of published curves
- Efficiency shall not be less than -10% of published curves at the pump recommended Best Efficiency Point

Advanced Gas Handler
The pump had to have necessary the necessary facility to install an advanced gas handler in the ESP assembly. This must prevent deterioration of head capacity of the pump. The advanced gas handler should also avoid motor load fluctuations due to cycling caused by severe gas interference. A gas handler of the rotary chamber type is preferred, since it will produce the highest possible separating force, good separating efficiency and handle highly viscous fluids.

Pump Intake
A standard X Metallurgy pump intake section shall be supplied as a separate bolt-on module. Pump intake section shall incorporate Abrasion Resistant Technology (ART) (Tungsten Carbide stabilizing journal bearings) in line with the pump design.

Seal / Protector
The ESP seal should be qualified for operating without failure at normal operating conditions:

- Well kill operation: Continuous injecting water into well at 15 deg C at 1000 liter/min.
- Running ESP at 60Hz against closed valve for 10 min.

Tandem seal systems should contain not less than three chambers per unit, both units configured from top to bottom in a bag-bag-lab arrangement. Parallel connections should be used between bag-bag chambers. Series connections should be used between bag-labyrinth chambers. Thrust bearing should be of high load/high temperature. Seal housing diameter should provide adequate clearance for MLE and discharge pressure capillary tube inside can/pod system or in conjunction with a Y-tool and by-pass system. All positive seal sections should include bags constructed of Aflas material. All O-rings in the seal section construction should be of Aflas or equivalent material. Other elastomers or proprietary products should not be used unless prior approval from Statoil. All shaft seals, both rotating and fixed faces, should be of silicon carbide construction. Seal elastomers (bellows) should be of Aflas construction. All mechanical seal faces should be of a hard abrasive resistant material such as silicon carbide, tungsten carbide or similar.

Motor
The motor should be capable of operating inside the operational envelope at the down-hole condition without comprising run life time. Motor should be of sufficient horsepower to drive the seal section, intake and pump of a given application across the range of possible well and fluid operating conditions. Motors should be supplied with internal electrical connections to be attached to base connections and pressure/temperature down-hole monitoring devices.

Tape-in motor connections or plug-in motor connections can only be supplied supported with a guarantee that failures attributed to the connection type will be sufficient grounds for a warranty claim award. A thermocouple installed in the motor winding is required in all motors for feed into the down-hole monitoring device. Motors should be capable of 15% overload (1.15 service factor). However, this overload capacity should not be utilized in sizing each motor to a pump. All motor stators should be of new and unused construction. The motor power factor and efficiency shall meet API recommended motor performance criteria.

Down-hole Sensor
The system shall communicate with surface via the main ESP cable. A down-hole sensor system is required for each ESP. This device shall be used primarily for ESP monitoring purposes. Data gathered shall also be used in determining individual well production rates and for production allocation.

Contractor should deliver the following ESP down-hole measured parameters as a minimum:

- Pump Intake Pressure (required)
- Pump Intake Temperature (required)
- Pump Discharge Pressure
- Motor Winding Temperature
- Current Leakage
- Vibration
- Formation Fluid Temperature (not influenced by motor temperature)
- Pump Fluid Flow (possible future requirement)

The following range and resolution for the measured parameters should be applicable:

- Pressure measurement range shall be 0-5000 psig/0-34,474 kpag
- Pressure accuracy shall be 0.1 % of full scale value or better
- Pressure resolution shall be 0.1 psi/0.69 kPa or better
- Temperature measurement range (motor winding) shall be 0-500˚F/0-260˚C or better
- Temperature measurement range (pump intake) shall be 0-302˚F/0-150˚C or better
- Temperature accuracy shall be 1% of full scale value or better
- Temperature resolution shall be 1.8˚F/1˚C or better

Manufacturers gauge metrology parameters for measurement (range accuracy, resolution, drift and data rates), all pressures, all temperatures, vibration, current leakage etc. should be specified by contractor. The sensor package should include a weather proof choke panel installed on the switchboard and all the necessary accessories to monitor and collect the down-hole data. The power supply for the surface panel of the sensor shall be configured to collect data when the VSD is on or off. [33]

5.3 Design of ESP at Peregrino

This chapter gives a brief description of how to perform a ESP design with a variable speed drive for a common application in an oil well. It will be described procedures that form a basis for computerized calculations that are used in the industry. A manual ESP sizing can be seen in Appendix A and B.

5.3.1 Operational Data

To be able to conduct a proper ESP design for a given well, thorough gathering of operational data has to be performed. Well productivity data is maybe the most important one since this have to be known to establish the desired flow rate from the well. This again depends on pump specifications. See API 11S4, 2008 for more information. The operational data is given in Table 13 in section 5.1.1.

5.3.2 ESP Selection

The selection process for a given well includes the following:

1. Determine pump series and pump type
2. Decide number of pump stages
3. Determine power requirement by pump
4. Check the shaft strength of the pump
5. Ensure that proper metallurgy for produced well fluid is selected

1. Pump series and pump type

It is important that the selected pump fit in the casing string of the well. In order to do so, one has to decide the correct diameter of the pump. An ESP pump in the 400 series usually means that it has an outside diameter of 4”. 562 series means 5.62” and so on [1]. According to in section 5.1.1 is the production casing for our well 9 5/8” in diameter so our selected pump should not be greater than 7”, we want to have some clearance between the equipment and the wellbore. Because well fluids have to be able to flow past the motor for cooling and we have to take into account the motor lead extension that runs outside the equipment. With other words a 700 series pump is the largest we can apply for this well, this is however as we will see later sufficient size.

One must also look at the performance curve to see that when the pump is run within the recommended operating range when delivering the desired production range. A pump should be selected such that the desired operating flow is as near as possible to the best efficiency point. See API 11S4, 2008 for more information. As described in Appendix A, the selected pump for medium well in Peregrino is a 675 series pump. This pump will be able to produce the required production rate within the recommended operating range and will fit in the well.

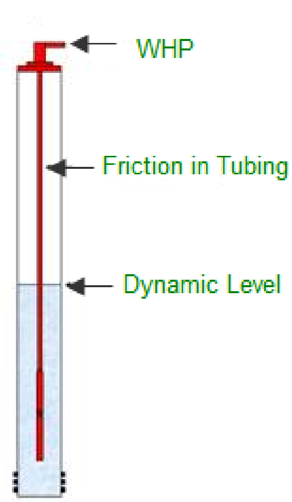

Figure 48. Total Dynamic Head [34]

2. Number of pump stages
In order to determine the required number of stages, the total head to be overcome by the pump has to be calculated. This is called the total dynamic head (TDH) and is the sum of the following components:

- The wellhead pressure (WHP)
- The frictional pressure drop that occurs in the tubing string
- The net hydrostatic pressure acting on the pump (Dynamic Level)

Next, we select the head per stage at maximum frequency (60 Hz) and maximum desired flow rate (14000 BPD). With the maximum total dynamic head, we calculate the number of pump stages required.

$$\text{No. Stages} = \frac{\text{Max TDH}}{\text{Head per stage at 60 Hz}} \quad [35]$$

The calculations of the number of pump stages can be seen in Appendix A. the result was as follows:

- **Case 1, 0% Water Cut:**
  Number of pump stages = 78

- **Case 2, 30% Water Cut:**
  Number of pump stages = 78

- **Case 3, 50% Water Cut:**
  Number of pump stages = 79

- **Case 4, 95% Water Cut:**
  Number of pump stages = 80

3. Determine power requirement by pump
Next, using the VSD Power Curve for the pump, we read off the BHP/stage at 60 Hz for maximum rate (14000 BPD). Then we calculate the required power:

BHP @ Max. Hz. = BHP/Stg. @ 60 Hz. x No. Stgs. x Specific gravity of the fluid pumped [34].

The calculations of the power requirement can be seen in Appendix A. the result was as follows:

- **Case 1, 0% Water Cut:**
  BHP = 794 (584kW)

- **Case 2, 30% Water Cut:**
  BHP = 811 (596kW)

- **Case 3, 50% Water Cut:**
  BHP = 838 (616kW)

- **Case 4, 95% Water Cut:**
  BHP = 882 (649kW)

4. Checking Shaft Strength
The amount of power the pump shaft can transmit should be checked to insure it is within the design limits stated in the manufacturers catalog. Exceeding this limitation can result in premature failure. See API 11S4 for more information. The design limits for high strength 675 series pump is according to Figure 49 1219 BHP at 60Hz. When we determined the power requirement above, the highest amount of horsepower was found to be 882 BHP. The pump will therefore not exceed the shaft limitation of 1219 BHP.

SHAFT HORSEPOWER LIMITATION

SERIES	HZ	338	375	400	450	513	562	675	725	875
SEAL Standard High Strength	60 50 60 50	171 (142) 270 (226)		287 (239) 450 (382)		500 (458) 715 (665)		890 (955) 1219 (1251)		1999 (1668) 3140
MOTOR Standard High Strength	60 50 60 50		247 (206)		432 (292)		736 (613) 1200 (1000)		1600 (1333)	

Figure 49. Design limitations [36]

5. Metallurgy
According to Table 13 in section 5.1.1 is initial sand production expected; this will require abrasion resistant pumps.

Gassy Wells
According to API 11S4 are hand calculations inadequate for gas applications. Computer software should be used to handle the more complex inflow and multiphase flow calculations. For more information see the API standard [35]. Appendix B is an example of hand calculations for the Peregrino field considering free gas. However it will be utilized a gas separator to avoid free gas at the pump intake.

Selection of the Seal
When selecting a seal section several considerations include:

- The seal must be compatible with the pump and motor
- The seal must fit in the casing string and allow clearance for the cable
- Have sufficient shaft strength to withstand maximum torque for the application
- Have sufficient expansion volume in chambers to allow expansion of motor oil

- Include a thrust bearing rated to handle the pump shaft axial thrust
- Be filled with proper motor oil
- Be compatible with the application

For more information see API 11S4, 2008.

Motor Selection
When selecting the submersible motor to match the ESP pump already chosen, one has to determine:

- The proper motor series
- The required motor power
- The right combination of motor voltage and amperage

For more information see API 11S4, 2008.

5.4 Future ESP Design

This section describes a proposed mechanical design for a Peregrino medium well. See section 5.1.1 for process data.

The main goal is to choose a design that will meet the production targets including ability to be integrated in the field process and have as long life time as possible, with other words; high reliability. To achieve this, we need to consider materials that can prevent against corrosion, erosion, emulsions, scale formation etc. Vendors can offer components of varying material quality, it is therefore important to know what kind of environment our ESP system shall be operated in so we can choose proper materials. Further we have do decide whether we want to apply compression or floater pumps.

ESP Pump

Table 14 shows the technical specifications of the proposed pump design.

Pump	
Housing type	420 SS UNS S42000
Shaft material	Nitronic 50 UNS S20910
Stage type	Mixed Flow
Stage build	Compression
Bearing material	Tungsten Carbide
Sleeves material	Tungsten Carbide
Elastomer material (O-rings)	Aflas
Impeller and diffuser material	Duplex Metal
Thrust description / thrust handling	Thrust taken up by seal bearing
Surface coatings	Corrosive resistant
Torsional & Lateral Critical Speeds	Minimum 40 - 60 Hz
Minimum number of monitoring points	9

Table 14. Proposed pump specifications

Housing Type

The surface materials of the down-hole equipment are located in a highly corrosive environment, and can contain; water, oil, gas and brine. Therefore should the housing of the pump, seal, separator, and motor be of a material that can prevent against corrosion.
420 SS UNS S42000 is of stainless steel and have a higher strength and hardness than the most common alloy 9Cr Mo [37]. However 420 SS UNS S42000 is not recommended for operating in environments with NaCl.

Shaft material
The pump shaft material must have the right properties so its remains it initial shape when it is subjected to high torque forces and temperatures. The pump shaft is also subjected to the corrosive and erosive well environment since it is in direct contact with the fluid being pumped. Nitronic 50 UNS S20910 is of stainless steel and provides therefore corrosion resistance. The material is also of high strength, which is important when it comes to stress-handling and resistance to abrasive wear. This material can also handle temperatures up to 1066˚C [38].

Stage type
ESP pump stages are categorized in radial or mixed flow design. Radial flow pumps are not able to pump flow rates higher than 3500 BPD [6]. So in the Peregrino case where the flow rates are between 5000 – 14000 BPD we have to use pump stages with the mixed flow design.

Stage build
As mentioned in chapter 3.1 two pump impeller designs are available; compression pumps and floater pumps. For Peregrino it is recommended to use compression pumps. This is because according to the process data given in chapter 5.1 it is expected that sand production will occur. Compression pumps have according to chapter 4.3 (solids production) better properties to cope with abrasive well conditions. It is believed that compression pumps offer a better reliability in high power applications because of the thrust handling. This allegation gets support from the historical data for compression vs. floater pumps in the ESP-RIFTS database. Compression pumps have according to RIFTS a MTTF of 2051 days, and floater pumps only a MTTF of 739 days. See Figure 50.

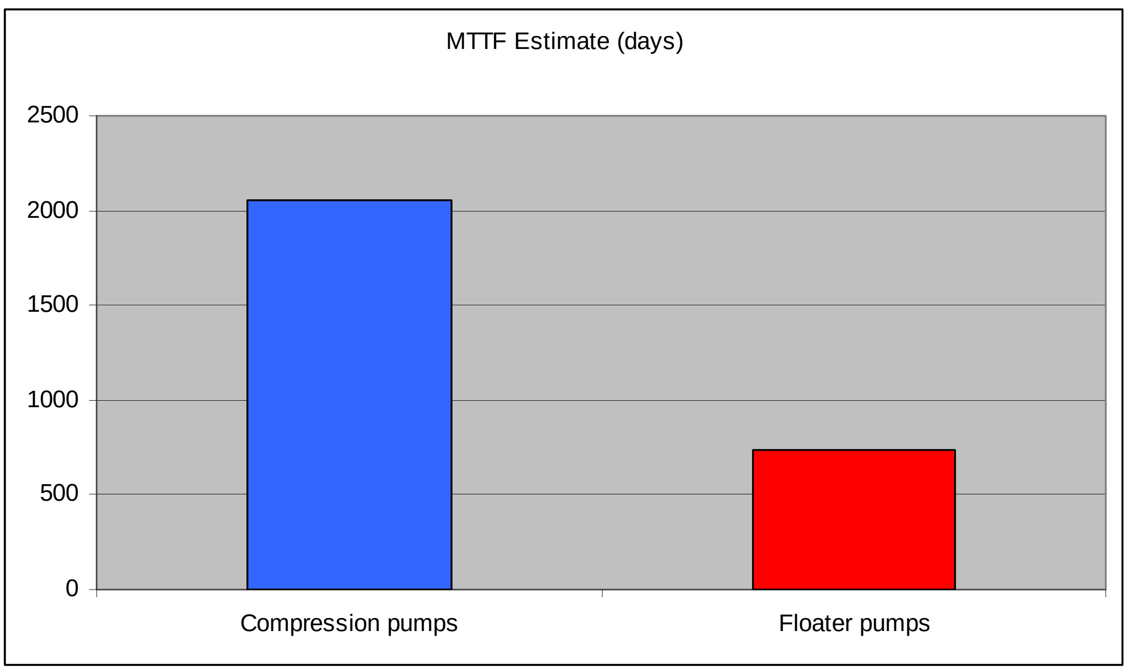

Figure 50. Compression vs floater pumps

Bearing material
As mentioned earlier in this section, it is likely that sand production will occur. This will as described in chapter 4.3 lead to abrasion damage on the pump bearings. To minimize this kind of wear is it important that the bearing material have a harder surface than the solid particles in the fluid being pumped, see Figure 41 in chapter 4.3. One Tungsten Carbide bearing in each pump stage is recommended to achieve maximum protection against axial wear, as described in chapter 4.3. The bearings must also be compliant mounted (also described in chapter 4.3).

Sleeves material
Tungsten carbide is the preferred material because of the same reasons as for bearing material. Sleeves along with the bearings will be a subject of the abrasive wear due to the sand production. As a result, in order to protect the shaft, it needs to be manufactured using the same material as the bearing.

Elastomer material (O-rings)
Aflas also called TFE/P is the preferred elastomer material. Aflas is a copolymer of tetrafluorethylene and propylene with a fluorine content of approximately 54%. The material has excellent resistance to acids, steam, hot water, brine, oil and gas. It's also resistant to all types of hydraulic fluids, all brake fluids and amine corrosion inhibitors. It can withstand temperatures between -5°C to +204°C [40]. Aflas should therefore be able to cope with the environment ESP will experience in Peregrino.

Impeller and diffuser material
The most common impeller and diffuser material is Ni-resist which is not a very hard material. See Figure 41 in chapter 4.3. The optimal material has a combination of the properties of a hard material to resist wear and a corrosion resistant material to ensure long run life. This is difficult to achieve, since when a metal gets heat treated to get the desired hardness, it loses corrosion resistance. Soft materials can have the right properties to protect against corrosion but they lack the hardness necessary for achieving long run life. Materials that combine these features are called Duplex Metal [41]. The preferred impeller and diffuser material is therefore duplex metal.

Thrust description / thrust handling
Since we have chosen the stage build to be of the compression type, the total thrust (shaft thrust and hydraulic thrust) are taken up by a robust thrust bearing in the seal section. See description of the seal thrust bearing.

Surface coating
Surface coating applied on the housing surfaces will enhance the protection of ESP equipment. The following features should be valid for the coating:

- Temperature resistant to over 200˚C, the reservoir temperature in Peregrino is according to section 5.1.1 78,6˚C.
- Hardness comparable to Tungsten Carbide so that the coating does not get scratched up easily.

Torsional & Lateral Critical Speeds
The pump must be able to operate in the frequency range of 40-60 Hz. The minimum speed of up to 40Hz is required in order to ensure proper lubrication of the thrust bearing as being lubricated by flow of oil past it. It is also has to do with the heat dissipation through this flow of oil as well. On the other hand; theoretical maximum speed of the ESP is limited by design of motor. [42]

Minimum number of monitoring points

- Intake Pressure
- Intake Temperature
- Motor Temperature
- Electronics Temperature
- Discharge Pressure
- Pump Pressure Differential
- Discharge Temperature
- Vibration
- Current Leakage

Seal Section

The most important operational features of the seal section are described in chapter 3.1.4. The proposed seal configuration is illustrated in Figure 51.

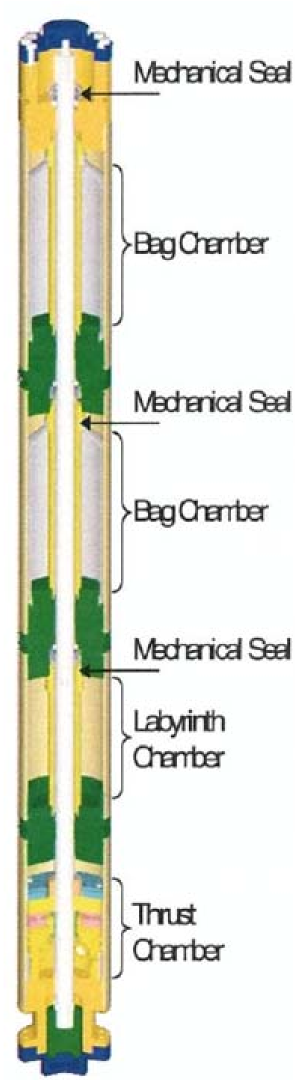

Figure 51. Seal Section. [43]

Seal Section	
Housing material	420 SS UNS S42000
Shaft material	Nitronic 50 UNS S20910
Number isolation chambers	3
Elastomer bag material	Aflas
Thrust bearing type	Slide bearing
Thrust bearing material	Silver plating
Mechanical seal material	Tungsten Carbide
Sleeves material	Tungsten Carbide
Number of mechanical seals	3

Table 15. Seal Section Specifications

Housing
The same housing material as for the pump is valid for the seal housing.
420 SS UNS S42000 stainless steel is chosen.

Shaft Material
The same shaft material as for the pump is chosen; Nitronic 50 UNS S20910 stainless steeel.

Number of isolation chambers
The seal includes three chambers, well fluid communicates from the top of the seal section to the first and second chamber. These are connected in parallel and are elastomeric bag configurations. The bags separate the well fluid from the high dielectric oil, which communicates to and is contained in the electric motor. The bags are also expandable, which allows for expansion of the motor oil when temperature increases. The third chamber is a labyrinth chamber where direct contact between the motor oil and well fluids occurs. The expansion and contraction of the oil causes the oil to follow a labyrinth path in the chamber. Since the well fluid will have a higher specific gravity than the motor oil found in the second chamber (specific gravity of motor oil are approximately 0.8), it will be trapped in the bottom of the labyrinth chamber and will not flow into the second chamber (see Figure 51). Three chambers are the maximum amount of chambers available [1]; it is believed that this will give the best protection of the motor since it involves the maximum amount of barriers.

Elastomer Bag material
Aflas is the preferred material because of the same reasons as given for elastomers for the pump. Operating temperature is an important consideration when selecting elastomer bag material, aflas is the material on the market today that can handle the highest temperatures. As stated earlier; the reservoir temperature in Peregrino is expected to be approximately 78,6˚C so the aflas material will easily withstand this temperature.

Thrust bearing type
The thrust bearing in the seal section is the main thrust bearing for the whole ESP system since we have chosen the compression type design. The preferred bearing type is a sleeve bearing which have the following advantages compared to roller bearings:

- They can handle higher thrust loads
- Less heat is generated in the bearing
- They are lubricated with an oil film that prevent metal to metal contact [1]

Thrust bearing material
To achieve a long run life for ESP in Peregrino the proper thrust bearing material should have the following properties:

- Corrosion resistant
  - As discussed earlier in the report, ESP' are operating in corrosive well environments.
- Have sufficient strength
  - The thrust bearing are subjected to high loads because thrust forces are developed on the fixed impellers which are transferred to the thrust bearing in the seal.
- Handle dirt particles
  - Solid particles in the lubrication oil can scratch and damage the thrust bearing surfaces.

Thrust bearing with silver plating is the preferred material, even though it is the most costly one. Silver bearings have all the properties described above and can handle temperatures up to 250˚C [43].

Number of mechanical seals
Each isolation chamber of the seal in Figure 51 is equipped with a rotating mechanical seal. Since there are three chambers the number of mechanical seals are three. The mechanical seals are located at the top of the chambers and prevent communication of well fluids and motor oil along the shaft.

Mechanical seal material
The mechanical seals contain a stationary seal ring fixed to the seal section housing and a rotating ring turning with the shaft, both are of tungsten carbide. These parts prevent leakage of fluids along the rotating shaft. Damaged mating surfaces initiate leaking; this is why in wells where sand production tungsten carbide on seal surfaces is preferred. The mechanical seal also contains three supplementary static seals. These are of rubber materials compressed against hard surfaces to provide the necessary sealing action as follows:

- Sealing between the housing and the stationary seal ring is performed by an O-ring of aflas material.
- An elastomeric bellow also made of aflas is fixed radially to the mechanical seal on the shaft. The bellow is forced downward by a spring and provides the seal on the rotating seal ring. [1] (Aflas is explained in the pump section above).

Sleeves
Tungsten Carbide is the preferred sleeve material for the same reason as explained for the pump sleeves.

Dual Seal
Dual seals are two seal sections connected together in tandem between the pump and the motor. Both seals have their own thrust bearing but they can not work in unison. To achieve proper cooperation between the bearings requires that the clearance between the moving parts of the thrust bearings is controlled to very close tolerances (less than 2.54 mm). This is difficult to achieve because of factors like; thermal expansion of metal parts and shaft compression. There are two ways to operate tandem seal sections:

1. All thrust is carried by the thrust bearing in the upper seal; this depends on sufficient axial gap between the shafts of the two seals. The isolation chambers of the upper seal protect the thrust bearing if the seal gets contaminated by well fluids. However, an increased wear of the thrust bearing starts and results in a downward movement of the shaft. This will lead to mechanical damage of the pump stages due to down-thrust. So in this case will the pump fail before the motor, since the lower seal protects the motor from contamination.

2. All thrust is carried by the thrust bearing in the lower seal. This is achieved if the tandem seal are installed in such a manner that the shaft of the lower seal slightly lifts the upper shaft's seal. This will assure that the two seals protect the main thrust bearing and the motor. Normally will contamination of the seal chambers start from the top and proceed downwards. The upper thrust bearing will fail first, then the lower bearing and as a result of failed thrust bearing the pump will fail. Finally well fluid gets into the motor and causes it to fail. Even though the ESP will finally fail, this is the preferred method since the time to system failure is greater than for the first case. [1]

Dual seals are preferred since it is believed that such a configuration will offer the best protection of the electric motor since it includes the maximum amount of barriers. The total amount of isolation chamber is as many as six. A research performed in ESP-RIFTS also supports that a dual seal configuration will achieve longer run life than a single seal configuration. According to Figure 52 has a dual seal system a MTTF of 994 days compared to 895 days of a single seal system.

According to Figure 53 has the proposed ESP system for Peregrino with a dual seal and a compression type pump a MTTF of 2578 days which is approximately 7 years. Compared to an ESP system comprised of a single seal and floater type pump with a MTTF of 682 days which is approximately 2 years, the proposed configuration is superior when it comes to run life. If we also include the proposed technical specifications described earlier in this chapter the run life will probably increase even more.

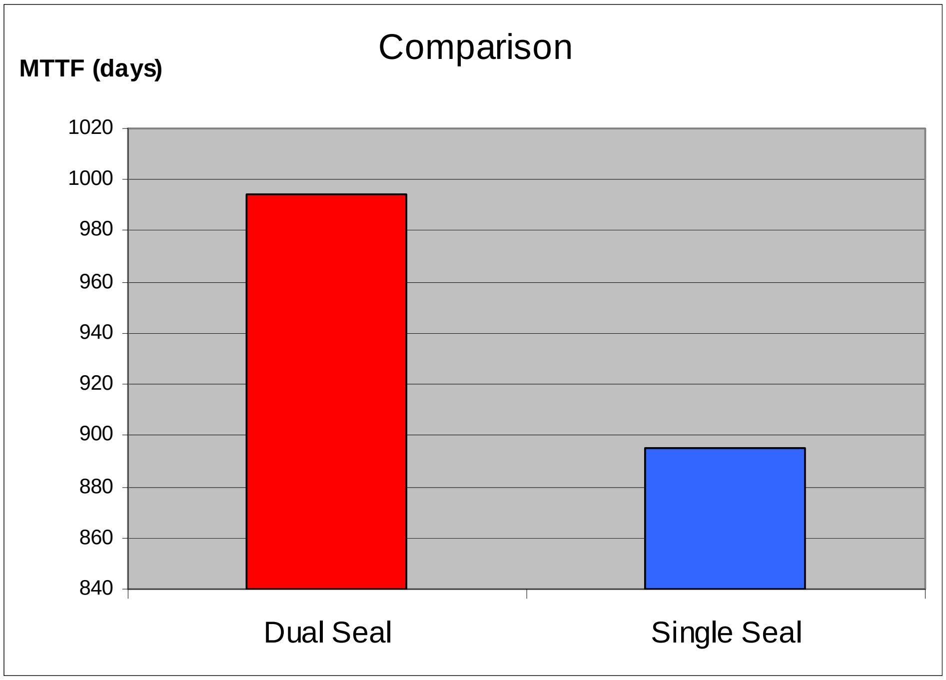

Figure 52. Run life comparison between dual and single seal configuation

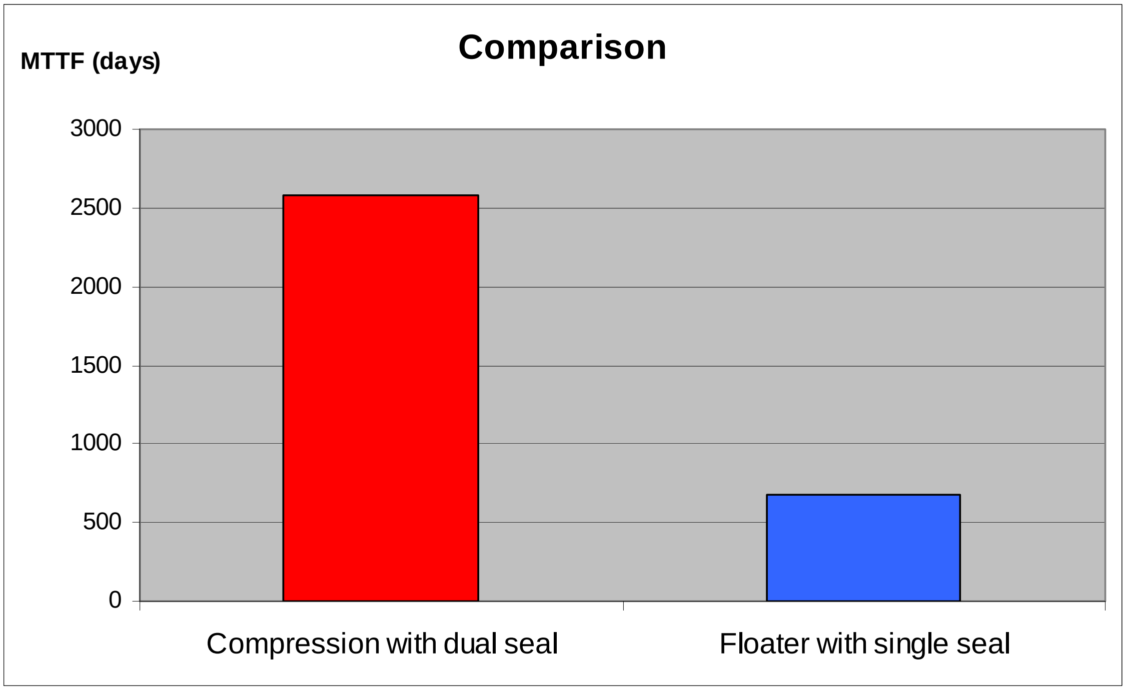

Figure 53. Compression pump with dual seal vs. floater pump with single seal

5.5 ESP Surveillance

To be able to operate the ESP in the recommended operating range and to make sure we are not running the system outside the design limitations we have to monitor several critical parameters. Normally when performing surveillance of mechanical equipment the objective are to make sure that the equipment is run inside the design limits. If the monitoring system in for example a gas plant detects that a compressor has higher operating temperature than recommended, the operator can evaluate this information and take action. In this case condition monitoring is a helpful tool to perform preventive maintenance. However, it is much more difficult to perform preventive maintenance for offshore ESP systems. ESP are normally run till they breakdown, and then replaced, with other words corrective maintenance. For the more accessible compressor, there is no problem to perform necessary maintenance/repairs, we have no ability to access the down-hole equipment and do maintenance in the case of ESP. However it is significant benefits that are gained with having a sufficient surveillance system for ESP, which will be explained in this chapter.

Motor current from ammeter chart

Surveillance of the electrical current drawn by the motor is a valuable tool, since ESP failures often are a result of an electrical error as explained in chapter 4. Surveillance of the current is achieved by use of a recording ammeter chart located in the VSD or switchboard. The ammeter monitors the input amperage to the motor by using a current transformer coupled to one of the power cables. The current is recorded as a function of time on a circular chart. The proper interpretation of ammeter chart readings, can provide operators with valuable information of the condition of the ESP, and therefore make him able to correct minor operational problems before they result in catastrophic failures [27]. Surveillance in onshore application often consists only of ammeter chart analysis. However in offshore application like Peregrino where the work-over costs are much higher (see chapter 5.6) it is strongly recommended to use advanced down-hole surveillance.

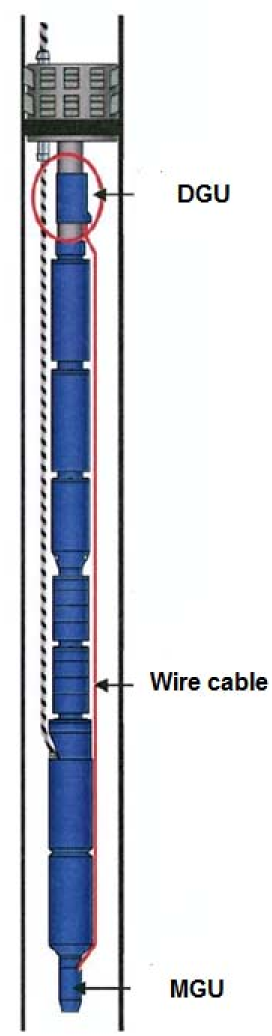

Figure 54. Down-hole surveillance. [43]

Down-hole surveillance of Peregrino

The down-hole devices are located in an instrument package which is either placed at the bottom of the motor housing or above the pump housing. When it is mounted on the motor housing it is called Motor Gauge Unit (MGU), and when mounted on the pump discharge it is called Discharge Gauge Unit (DGU). The advantage of having the package at the bottom of the motor is that its power supply and data transmission through the cable are easy accomplished. The disadvantages are that the cooling effect subjected from the produced liquid cannot be detected. This is because the MGU monitors the temperature inside the gauge units itself which are located below the motor. This problem is solved if a DGU mounted on top of the pump is applied, which measure discharge temperature and pressure. The main disadvantage of this position is the need for a dedicated communication wire between the two gauges as illustrated in Figure 54. The preferred choice is as shown in the figure, where we have both MGU and DGU installed. This will give us the ability to combine the advantages to both units. However, one major disadvantage of this configuration is that vibration measurements are too weak. This will be discussed in the last section of this chapter, but let us first discuss the meaning of monitoring critical parameters.

- **Pump Intake Pressure (PIP).** Surveillance of the PIP is performed by a venture nozzle that makes the operator able to know if for example it is free gas at the pump suction. See appendix B. Since the bubble-point pressure for the Peregrino well is 48 bar (700 psi), see Table 13 For this case the surface control system can be configured to trigger an alarm at PIP of 55 bar (800 psi) and to trip the pump if the PIP is further reduced to 52 bar. However, this is just a solution if it is decided to not utilize a gas separator. We can also use the measured PIP to compare it with design values to find out the accuracy of the ESP design.

- **Pump Discharge Pressure (PDP).** Surveillance of the PDP are also performed by an venture nozzle and will provide the operator with data of changes in specific gravity of the produced fluid. Increasing PDP is an indication on a reduction in specific gravity of the fluid, as a result of decreased water cut or increased gas fraction. See appendix B. Decreasing PDP is an indication on higher specific gravity of the fluid being pumped. The setting of an alarm should be based on calculated maximum value for discharge pressure during normal operating conditions, or the value can be set at normal operating PDP plus a margin (for example 3 bar). The value can also be determined practically in the field by performing a shut-in test, i.e. observing the measured PDP and setting the alarm level below this value [48]. Measuring of the PDP gives also indications of down-hole problems, like equipment wear. Worn impellers in the pump will for example not be able to deliver the same production rate as new ones.

- **Pump Pressure Differential (PPD).** Surveillance of the PPD can be used to ensure that the pump is run within its recommended operating range. For a given pump frequency and with a known density of the fluid being pumped, a minimum and maximum PPD corresponding to the up-thrust and down-thrust range on the ESP can be set to ensure that the pump is operated in range [46]. Consideration of Figure 55 will illustrate this as an example. The minimum and maximum operating points is according to the figure as follows:

Total head for minimum rate: 3774 ft (1150 m)
Total head for maximum rate: 1957 ft (596 m)

Converting the total head to pressure:

$$P = 0.434 \cdot TH \cdot SG_L$$

Where: P = Pressure [psi]
0.434 = Conversion factor [-]
TH = Total Head [ft]
SG_L = Specific Gravity of Liquid, set to 0.95 [-]

Minimum rate: P = 0.434 x 3774 x 0.95 = 1556 psi (107 bar)

Maximum rate: P = 0.434 x 1957 x 0.95 = 807 psi (56 bar)

These values can be programmed into the surface control system so that an alarm would trigger if PPD increase above 107 bar. A pump trip could also be set if PPD continued to increase, which is an indication that the pump is moving into a low flow or potentially damaging down-thrust condition [48].

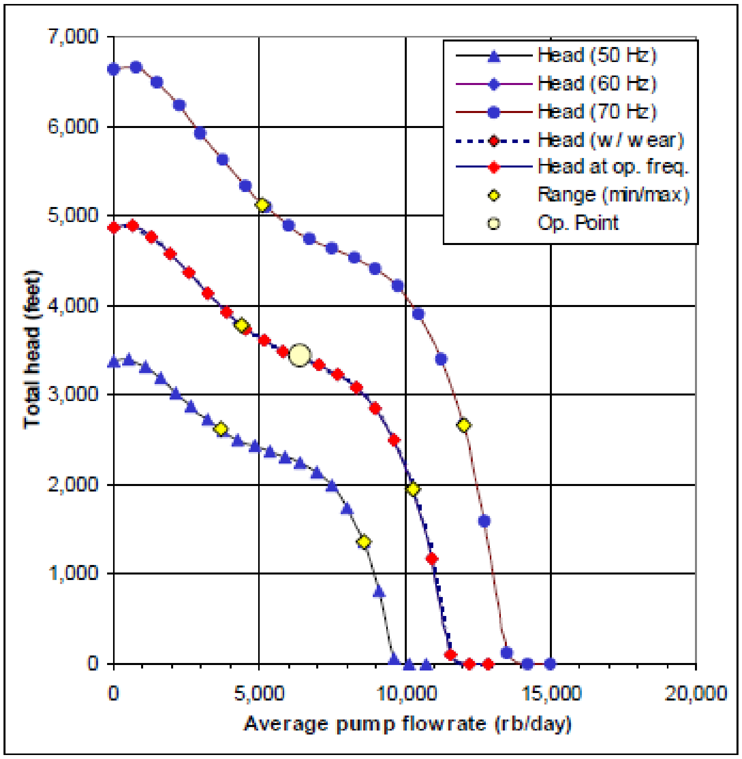

Figure 55. Efficiency range of a pump in relation to the operating point. [48]

- **Motor Temperature (MT).** Surveillance of MT includes usually motor winding temperature or motor oil temperature. Motor winding temperature increases rapidly in response to ESP problems and gives a better description of the motor condition than the motor oil temperature. This is because when the motor windings get heated, they increase the oil temperature (not the other way around). So surveillance of the motor windings is the preferred choice. An alarm should be set to activate at approximately 20˚C above normal operating temperature [48]. Surveillance of motor temperature will prevent life limiting damage from occurring. Such damage can for example be scale formation on the motor housing which reduces the cooling effect from the well fluid flowing past. Other well conditions like plugging and increased well temperature can also be detected by monitoring the motor temperature.

- **Pump Intake Temperature (PIT).** Surveillance of PIT acts as a back-up to MT and can have an alarm point at the same value as MT. However, it is most likely that the motor temperature will respond first. Surveillance of the PIT can also give indications of change in well flow-rate [48]. Some systems measure the temperature inside a gauge and use this for the intake temperature; this can result in false readings due to heat from the electronics in the gauge. The response times of these types of design are also slow, since the heat transfer need to affect a large area of metal on the gauge unit before affecting the temperature sensor. An external temperature sensor would be the preferred choice since this would give more accurate readings and faster response times. However this must be of a robust design to handle the harsh environments in the well. [42]

- **Electronics Temperature (ET).** The internal electronics (power regulators, etc) in the gauge unit will raise the temperature a few degrees above the external ambient temperature. It is important to monitor this temperature since if it rises above expected levels; this indicates that either the choke or electronic circuits are consuming more power than expected [42]. This information can be used to perform preventive actions to promote the life time of the gauge units.

- **Vibration Measurement of the pump (VM).** Surveillance of vibration includes mechanical (e.g. wear), electrical (e.g. frequency) and hydraulic (e.g. viscosity) components. This makes it difficult to interpret an exact value of vibration. However, the trend of vibration is important and can indicate problems or change in normal operating conditions like:

  - Change in motor frequency
  - Change in wellhead pressure
  - Increase in well water-cut and indication of emulsions
  - Onset or increase in solids production
  - Gas locking
  - Change in temperatures caused by severe up-thrust/down-thrust operation. [48]

  The vibration levels will vary depending on the pump speed, normally are ESP run at frequencies between 40 – 60Hz. It is not necessarily the case that the faster the speed the higher the vibration, often when a pump is run as low as 30 Hz the vibrations is greatest [16]. Hence, surveillance of vibrations will help determine the optimal operating frequency. As explained in chapter 4.2 are today's existing vibration measurement method to measure vibration in the radial and tangential direction (see Figure 42). There are three reasons to think that this method is insufficient.

  First, the accelerometer in the MGU is placed far away from the pump (see Figure 42). When mounting an accelerometer it's desirable to install the sensors as close as possible to the bearings to easier spot changes in vibration, this is not achieved when having the accelerometer installed below the motor maybe 10-30 meters from the pump.

  Second, it is not recommended to use accelerometers on pumps with slide bearings which ESP pumps use. As explained in section 4.4 accelerometers are best suitable for high frequencies (10 – 12000 Hz), which means it is recommended to use on roller bearings [49].

  Third, the existing method only measures vibrations in the X/Y direction and does not consider axial displacement. That means that wear and tear of the thrust bearings are hard to detect. [49]

Figure 56 illustrates an alternative method to perform vibration surveillance on ESP, where proximity probes are used to measure radial vibration on every bearing and one proximity probe on each shaft length that measures axial displacement. This technology does not exist today as far as the author is aware of. There can also be some challenges to be overcome when manufacturing sensors that can operate reliable in the harsh well environment. However the author thinks that this is the future way of doing vibration surveillance.

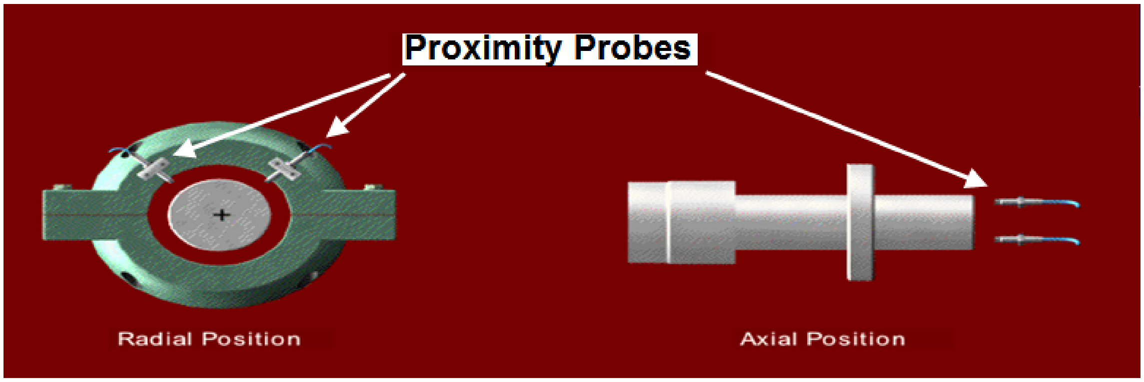

Figure 56. Alternative method. [47]

<u>**Example of ESP failure**</u>

Figure 57 shows an example of an ESP failure that could have been avoided had the proper sensor been used.

The figure shows five parameters that give reason for a failure to occur:

- **Increasing motor oil temperature.** Increasing motor oil temperature can be a result of poor cooling effect from the well fluid, because of for example scale formation on the motor housing. Or the reason can be increasing well fluid temperature. This can lead to overheating of the motor.

- **Increasing intake temperature**. Increasing intake temperature depends on the temperature of the well fluid, which can change over time. As we see from the figure the intake temperature and oil temperature are more or less equal, this give reason to think that the well fluid temperature has increased since that will cause both the oil temperature and the intake temperature to increase.

- **Decreasing discharge pressure.** Decreasing discharge pressure can be a result of a worn pump or motor. The reason can also be decreasing intake pressure. Decreasing discharge pressure means that we are not producing the desired rate.

- **Decreasing intake pressure.** Intake pressure depends on the reservoir's flowing properties; therefore a reduction in intake pressure is because of changing well conditions. If the intake pressure falls below the bubble point pressure, gas will be released from the fluid. This can lead to gas lock if no gas separator is installed.

- **Increasing vibration.** As described in chapter 4.5 are changes in vibrations dependent on several factors. Since we in this case have significant changes in flow through the pump, its reason to think that the vibrations are flow induced.

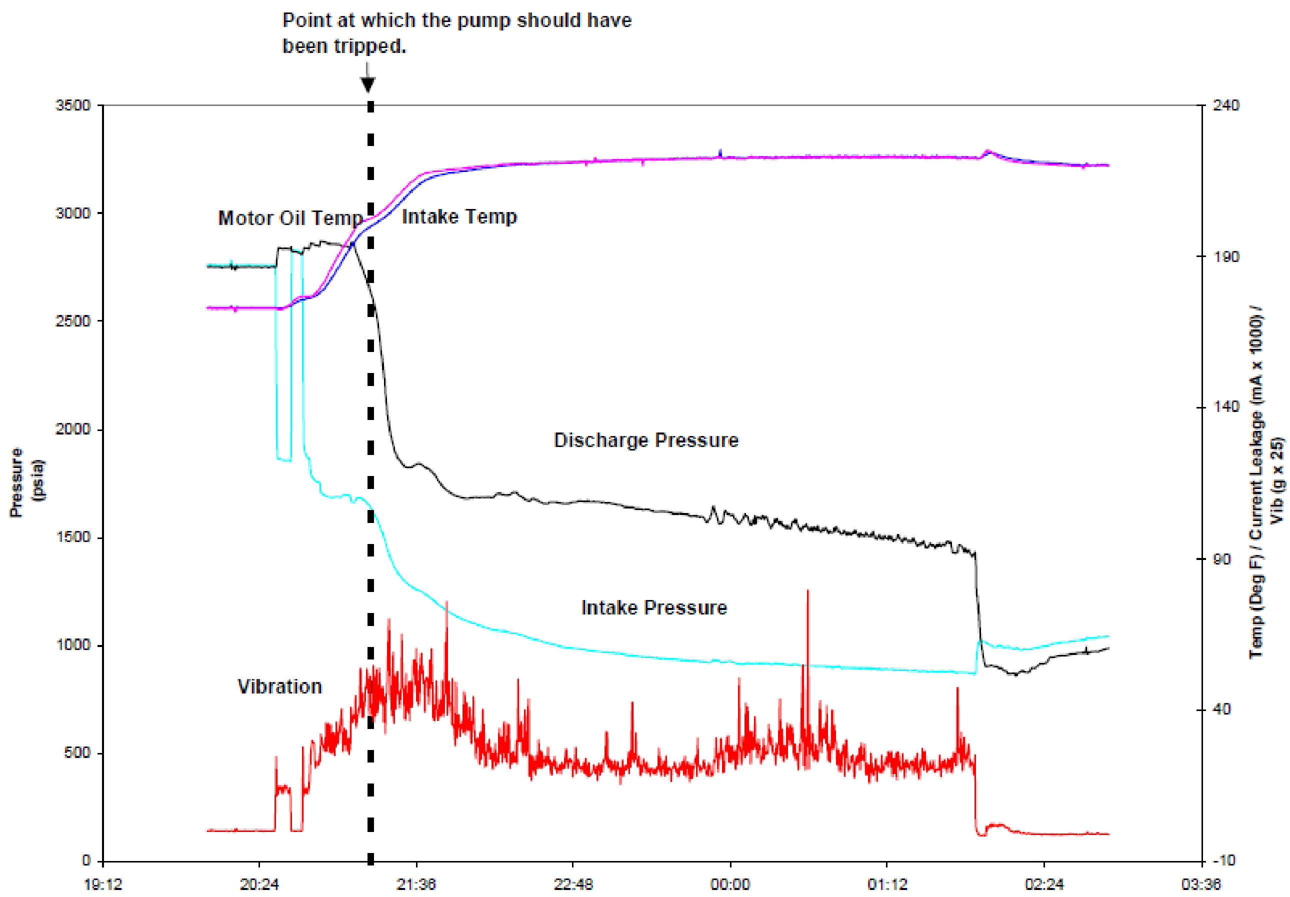

Figure 57. Example of a preventable failure. [48]

Update Rate and Instrument Communication

Update rate of the sensors are according to a leading ESP supplier as follows: Every 4 seconds, the intake pressure is updated, all other parameters are updated every 22 second [43]. For performing surveillance is a update rate of 22 seconds quite low, it is desirable with a high update rate to be able to see with accuracy what condition the machine is in. Today signals from the gauge unit are carried through the ESP power cable, the signals in the Peregrino case have to travel several kilometres before reaching the surface. The power cable is of the conventional design and offers limited transfer rates [1]. However, there is one solution to this problem; fibre optic cables. By use of fibre optic cables will the velocity and signal treatment increase significantly, this will make it possible to increase the update rate of sensors and perform real life condition monitoring. Fibre optic cables for ESP use are not at the market at present, but they are under development and we will probably soon see them in ESP applications [16].

5.6 Life Cycle Costs (LCC)

General

The life cycle cost of any piece of equipment is defined as the total "lifetime" cost to purchase install operate, maintain, and dispose of that equipment. Determining LCC involves following a methodology to identify and quantify all of the components of the LCC equation. LCC can be used as a comparison tool between possible design alternatives, the LCC process will show the most cost-effective solution within the limits of the available data.

The general LCC equation can be stated as:

$$LCC = C_{ic} + C_{in} + C_e + C_o + C_m + C_s + C_{env} + C_d \quad [50]$$

LCC = life cycle cost
C_{ic} = initial cost, purchase price (pump, system, pipe, auxiliary services)
C_{in} = installation and commissioning cost (including training)
C_e = energy costs (predicted cost for system operation, including pump driver, controls, and any auxiliary services)
C_o = operation costs (labor cost of normal system supervision)
C_m = maintenance and repair costs (routine and predicted repairs)
C_s = down time costs (loss of production)
C_{env} = environmental costs (contamination from pumped liquid and auxiliary equipment)
C_d = decommissioning/disposal costs (including restoration of the local environment and disposal of auxiliary services).

LCC for Peregrino ESP

The above equation is valid for top-side pump systems and is not tailor-made for ESP. ESP are deferring strongly from pumps used on platforms or production plants as explained in chapter 3, and there are major uncertainties to numbers regarding environmental and disposal costs. A simplified equation will be therefore be used to identify LCC drivers for ESP.

$$LCC = C_p + C_e + C_w + C_d$$

LCC = Life cycle cost
C_p = Purchase price ESP
C_e = Energy costs (predicted cost for system operation, including pump driver and controls)
C_w = Work-over costs
C_d = Down time costs (loss of production)

Financial factors must also be considered in developing the LCC:

- Present energy price
- Inflation during the pump's lifetime
- Discount rate
- Interest rate

- Expected equipment life

The following examine each element and offer suggestions on how a realistic value can be determined for use in calculating LCC.

C_p – Purchase price of ESP
The purchase price for one high power ESP with VSD are approximately 10 mill NOK [46].

C_e – Energy costs
Energy consumption is calculated by gathering data of the system output. If output is steady, the calculation is simple. If the output varies over time, a time-based usage pattern needs to be established. We will assume that the energy output is steady to simplify the calculation.

C_w – Work-over costs
A work-over (WO) is in this context a replacement of a used ESP with a new one, WO costs for offshore well with dry trees is approximately 25 MNOK [47]. It is 30 oil producing wells in the Prergrino field according to chapter 5.1 and the field has a life time of 40 years [29]. We assume that all wells are utilizing ESP during the whole life time of the field. Table 16 illustrates the effect run life has upon the WO costs during the whole field life. We can see that work-over costs are depending strongly on ESP run life. The WO is almost six times as expansive if the ESP has a run life of one year vs. five. Column four in the table is found by creating a timeline of the total field life, and then divide it up into run life sections to find the number of WO during the field life. Column five is found by multiplying number of wells with number of WO to find the total number of WO operations. Finally is the total WO cost calculated in column seven, by multiplying number of WO with WO cost pr pump.

Column	1	2	3	4	5	6	7
	Run life [years]	**No. of wells**	**Field life [years]**	**No. of WO during field life**	**Total no. of WO**	**WO cost pr pump [MNOK]**	**Total WO cost [MNOK]**
	1	30	40	39	1170	25	29250
	2	30	40	19	570	25	14250
	3	30	40	13	390	25	9750
	4	30	40	9	270	25	6750
	5	30	40	7	210	25	5250

Table 16. WO cost vs. Run Life

C_d – Downtime and loss of production costs
An offshore work-over operation takes approximately seven days [44]. The production rate for Peregrino medium well is between 7000 – 14 000BPD, see Table 18. We assume 7000BPD as production rate when calculating downtime costs. We also assume that the oil price is 70$ pr barrel which equals 420 NOK. Loss of production can therefore be calculated as:

C_d = 7 days x 7000BPD x 420NOK = 20,58 MNOK

It is however not entirely correct to call this lost production, since after the work-over is completed the ESP continues producing from the well. So a more accurate terminology would be delayed production.

In the following a LCC calculation for Peregrino will be given. The analysis will show the LCC cost pr year for 30 oil producing ESP, see Table 17.

row	**Input**	Run Life = 1 year	Run Life = 2 year	Run Life = 3 year	Run Life = 4 year	Run Life = 5 year
1	Purchase Price pr ESP [MNOK]	10	10	10	10	10
2	Purchase Price pr year [MNOK]	293	143	98	68	53
3	Energy price per kWh [NOK]	0,1	0,1	0,1	0,1	0,1
4	Power requirement pr ESP [kW]	596	596	596	596	596
5	Power requirement of 30 ESP [kW]	17880	17880	17880	17880	17880
6	Field life [years]	40	40	40	40	40
7	Number of workovers during 40 years for 30 wells	1170	570	390	270	210
8	Number of workovers for 30 wells pr year	29	14	10	7	5
9	Time consumed by each workover [days]	7	7	7	7	7
10	Time consumed by workover pr year for 30 wells [days]	205	100	68	47	37
11	Time consumed by workover pr year for 30 wells [hours]	4914	2394	1638	1134	882
12	Operating hours for 30 ESP pr year	150000	150000	150000	150000	150000
13	Real operating hours (12 - 11)	145086	147606	148362	148866	149118
14	Energy costs for 30 ESP pr year (3 x 5 x 10 x 10^-6) [MNOK]	259,41	263,92	265,27	266,17	266,62
15	Workover costs for 30 wells for total field life [MNOK]	29250	14250	9750	6750	5250
16	Workover costs for 30 wells pr year [MNOK]	731,25	356,25	243,75	168,75	131,25
17	Loss of production pr workover [MNOK]	20,58	20,58	20,58	20,58	20,58
18	Down time costs pr year [MNOK]	601,97	293,27	200,66	138,92	108,05
19	Interest rate [%]	8 %	8 %	8 %	8 %	8 %
20	Inflation rate [%]	4 %	4 %	4 %	4 %	4 %
21	Dicount factor	0,96	0,96	0,96	0,96	0,96
22	Sum yearly costs (14 + 16 + 18) [MNOK]	1333,22	913,43	709,68	573,84	505,92
23	Present value of yearly costs [MNOK]	1279,89	876,90	681,29	550,88	485,68
	Output					
24	**Present LCC-value pr year (2 + 22 - 23) [MNOK]**	**346**	**179**	**126**	**90**	**73**

Table 17. LCC pr year for Peregrino

Description of the data presented in the table:

Row 2. The purchase price for one ESP is as mentioned above approximately 10 MNOK, the purchase price for each year is therefore 10 MNOK multiplied with the number of WO pr year in row 8.

Row 3. Energy price is assumed to be 0,1 NOK pr kWh.

Row 4. The power requirement for each ESP is taken from Appendix A, case 2.

Row 7. Number of WO during 40 years for 30 wells is taken from Table 16.

Row 8. Number of WO for 30 wells pr year is found by dividing row 7 with row 6 (field life).

Row 9. Time consumed by each work-over is as mentioned above 7 days.

Row 10. Time consumed by WO pr year for 30 wells is found by multiplying row 8 with row 9.

Row 12. Operating hours for an ESP is assumed to be 5000 hours; operating hours for the whole field (30 wells) is therefore 150000 hours.

Row 13. Real operating hours is operating hours each year minus the downtime a work-over operation requires. (row 13 = row 12 – row 11)

Row 14. The energy costs are found by multiplying the power requirement with the energy price and operating hours. (row 14 = row 3 * row 5 * row 10)

Row 15. WO costs is taken from Table 16.

Row 16. WO cost pr year is found by dividing row 15 by row 6 (field life).

Row 17. Loss of production was calculated above in the downtime and loss of production section.

Row 18. Downtime costs per year are found by multiplying loss of production per work-over with the number of work-over. (row 18 = row 8 * row 17)

Row 19 and 20. The interest rate for new capital projects is assumed to be 8% and an inflation rate of 4% is considered [50].

Row 21. The discount factor is according to Table 20 in appendix c (n = 1, real discount rate = 4) 0.96.

Row 22. Sum yearly costs are found by adding the energy costs, work-over costs and downtime costs.

Row 23. Present value of yearly costs is found by multiplying the sum of yearly costs with the discount factor.

Row 24. Finally the LCC pr year is found by adding purchase price and yearly costs, then subtracting the present value.

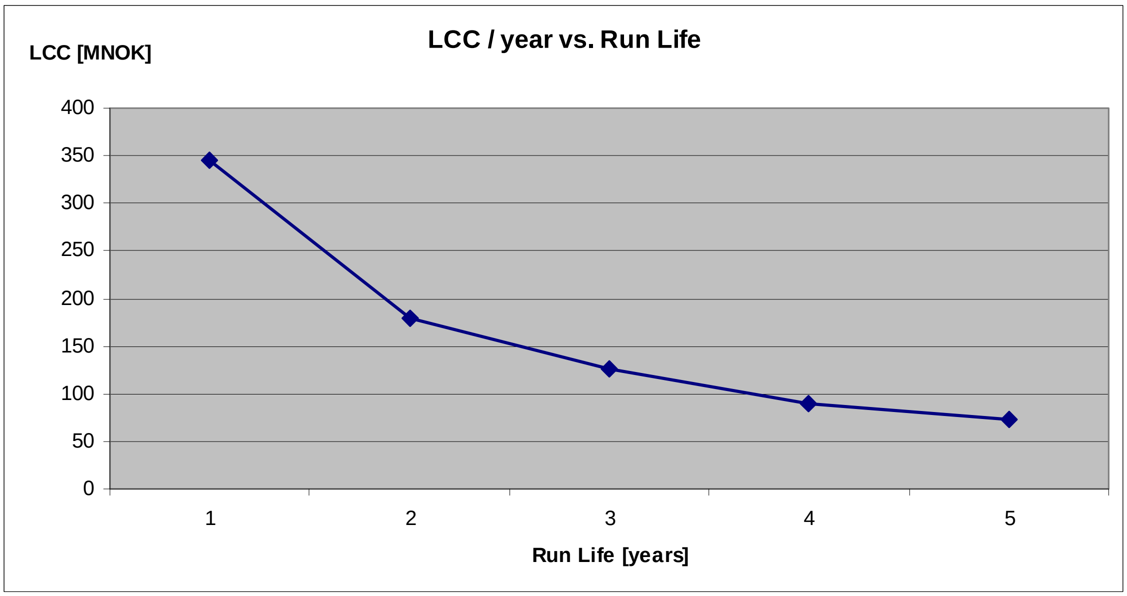

Figure 58. LCC pr year compared with run life.

Figure 58 illustrates the LCC calculation results. We can see that LCC per year decreases significantly when ESP run life is prolonged. This gives us a picture that enhanced run life of ESP will be very cost beneficial. We see that the LCC reduction is greatest from 1 to 2 years of run life, then the graph levels more and more out. This has to do with a significant reduction in WO cost when the run life grows, as a result of less WO operations. According to the analysis performed in chapter 4, is the average run life of ESP approximately 2,5 years. Figure 58 shows that if the run life is increased from 2,5 to 5 years, the operator will reduce the LCC by approximately 75 MNOK each year of production. This number is without concerning the extra cost of utilizing tailor made components and materials, to achieve a longer run life. But it is not reason to think that this cost will have detrimental affects of the LCC, when we look at the extremely high work-over costs that are dominating LCC for ESP.

6. Conclusion

It has been identified that ESP failures are often caused by electric errors because of insufficient protection of the electric motor, which results in a motor breakdown. A stronger barrier for preventing leakage of well fluids in the motor will increase run life. One of several solutions can be to install tandem seals, but also improved single seals with regards to material selection will achieve enhancement in ESP run life. To overcome operational and technical challenges, and to enhance ESP run life in Peregrino, the following main actions are recommended:

- Use compression type pumps with tandem seals
- Upgrade technical specifications with concern to material selection
- Install proximity probes for vibration measurement of the pump

The Peregrino field will be subjected to initial sand production, in addition high production rates and high water cut lead to increased sand production during the field life. Compression pumps handle solids production better than floater pump because it is no contact between impellers and diffusers under operation. Tandem seal gives the best protection of the motor since it offers a greater number of barriers than single seal designs. Material selection is an important issue to consider, especially for challenging applications like Peregrino. Component materials of varying quality and design limitations are available on the market. To improve reliability one has to choose materials which are designated to its application area. For abrasive wells as in the case of Peregrino, we have to imply materials that are harder than the solids particles that issue a threat to equipment run life. Vibration measurement of today consists of accelerometers mounted in a gauge unit blow the motor. This method is insufficient to deliver accurate condition monitoring of the pump. A better alternative would be to use proximity probes mounted close to each bearing along the pump to better detect bearing wear.

In addition, should it be put attention to the importance of training of operators so that the ESP is run inside its design limits. Full string test of the ESP system performed both at the manufacturer and on site would also result in better quality control. Development of an ISO standard that covers future applications is also very valuable for optimizing ESP. Enhancing ESP run life is necessary to achieve cost benefit in a LCC perspective. The average run life of ESP in oil wells with the power range between 300-1600 HP are 2,5 years. If the ESP used in Peregrino achieves a run life of 5 years the LCC reduction will be approximately 75 MNOK per year.

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Appendix A. Sizing with VSD for medium well in Peregrino

This section is an example on how to perform a sizing for ESP systems with a Variable Speed Drive (VSD) in oil wells. The sizing will be performed for well fluids with the following water cut: 0%, 30%, 50%, and 95%. We assume that the fluid being pumped consists of water and oil, and neglects free gas at the pump. Table 18 shows the reservoir and well data given by Statoil for medium well in the Peregrino field.

Medium Well ESP Design Parameters		
Oil Gravity	14	˚API
Specific Gravity of Water	1,05	(100,000 ppm)
Wellhead Pressure	100	Psi
Productivity Index (PI)	6,0	BPD/Psi
Water Cut	0%, 30%, 50% and 95%	
Expected Production rate	5000 - 14000	BPD
Initial Reservoir Pressure	3352	Psia
Datum	7678	ft TVD MSL
Production casing	9 5/8" #47 lb/ft	
Pump Setting Depth	6431	ft
Production Tubing	5 1/2" #17 lb/ft	

Table 18. Process data. [31]

The calculations in the following sizing procedure are in American units, and not SI units. This is because the formulas originate from API standards. However the answers will also be given in SI units.

First we have to convert oil gravity (14 ˚API) to specific gravity of oil (SG_O):

$$SG_O = \frac{141{,}5}{131{,}5 + ^\circ API} \longrightarrow SG_O = 0{,}97$$

Then we solve for the new well flowing pressure (P_{wf}) at the desired production rates (Q_d).

$$P_{wf} = P_r - \left(\frac{Q_d}{PI}\right)$$

Where: P_{wf} = Well Flowing Pressure [psi]
P_r = Reservoir Pressure [psi]
Q_d = Production Rate [BPD]
PI = Productivity Index [BPD/psi]

Minimum Desired Rate = 5000 BPD

$$P_{wf} = 3352\,psi - \left(\frac{5000bpd}{6{,}0bpd/psi}\right) = 2519\,psi \text{ (174 bar)}$$

Maximum Desired Rate = 14000 BPD

$$P_{wf} = 3352\,psi - \left(\frac{14000bpd}{6{,}0bpd/psi}\right) = 1019\,psi \text{ (70 bar)}$$

Case 1, 0% Water Cut

Specific Gravity of liquid (SG_L)

Water Cut is set to 0%

$SG_L = SG_O$

$SG_L = 0{,}97$

Pump Intake Pressure (PIP)

$$PIP = P_{wf} - \left[\frac{(\text{Datum - Pump Depth}) \cdot SG_L}{2{,}31\,ft/psi}\right]$$

Where: PIP = Pump Intake Pressure [psi]
P_{wf} = Well Flowing Pressure [psi]
Datum = Depth Reference [ft]
Pump Depth = Pump Setting Depth [ft]
SG_L = Specific Gravity of Liquid [-]
2,31 = Conversion Factor [ft/psi]

$$\text{PIP for minimum rate} = 2519\,psi - \left[\frac{(7678\text{ ft} - 6431\text{ ft}) \cdot 0{,}97}{2{,}31\,ft/psi}\right] = 1995\,psi \text{ (138 bar)}$$

$$\text{PIP for maximum rate} = 1019\,psi - \left[\frac{(7678\text{ ft} - 6431\text{ ft}) \cdot 0{,}97}{2{,}31\,ft/psi}\right] = 495\,psi \text{ (34 bar)}$$

We see that we get a higher PIP when pumping the minimum rate because of a higher well flowing pressure (P_{wf}).

Total Dynamic Head (TDH)

TDH = Net Lift + Friction Loss + Wellhead Pressure

$$\text{Net Lift} = \text{Pump Depth} - \left(\frac{\text{PIP} \cdot 2{,}31\,\text{ft/psi}}{\text{SG}_L} \right)$$

Where: Net Lift = True Vertical Depth (TVD) of the dynamic liquid level [ft]
Pump Depth = Pump Setting Depth [ft]
PIP = Pump Intake Pressure [psi]
2,31 = Conversion Factor [ft/psi]
SG_L = Specific Gravity of Liquid [-]

$$\text{Net lift for minimum rate} = 6431 - \left(\frac{1995\,psi \cdot 2{,}31\,\text{ft/psi}}{0{,}97} \right) = 1680\,ft \;\; (512\text{m})$$

$$\text{Net lift for maximum rate} = 6431 - \left(\frac{495\,psi \cdot 2{,}31\,\text{ft/psi}}{0{,}97} \right) = 5252\,ft \;\; (1601\text{m})$$

We see that we get a higher TDH when pumping the maximum rate because of a lower PIP. With other words; the total head to be overcome by the pump is increasing with the production rate.

Tubing friction loss

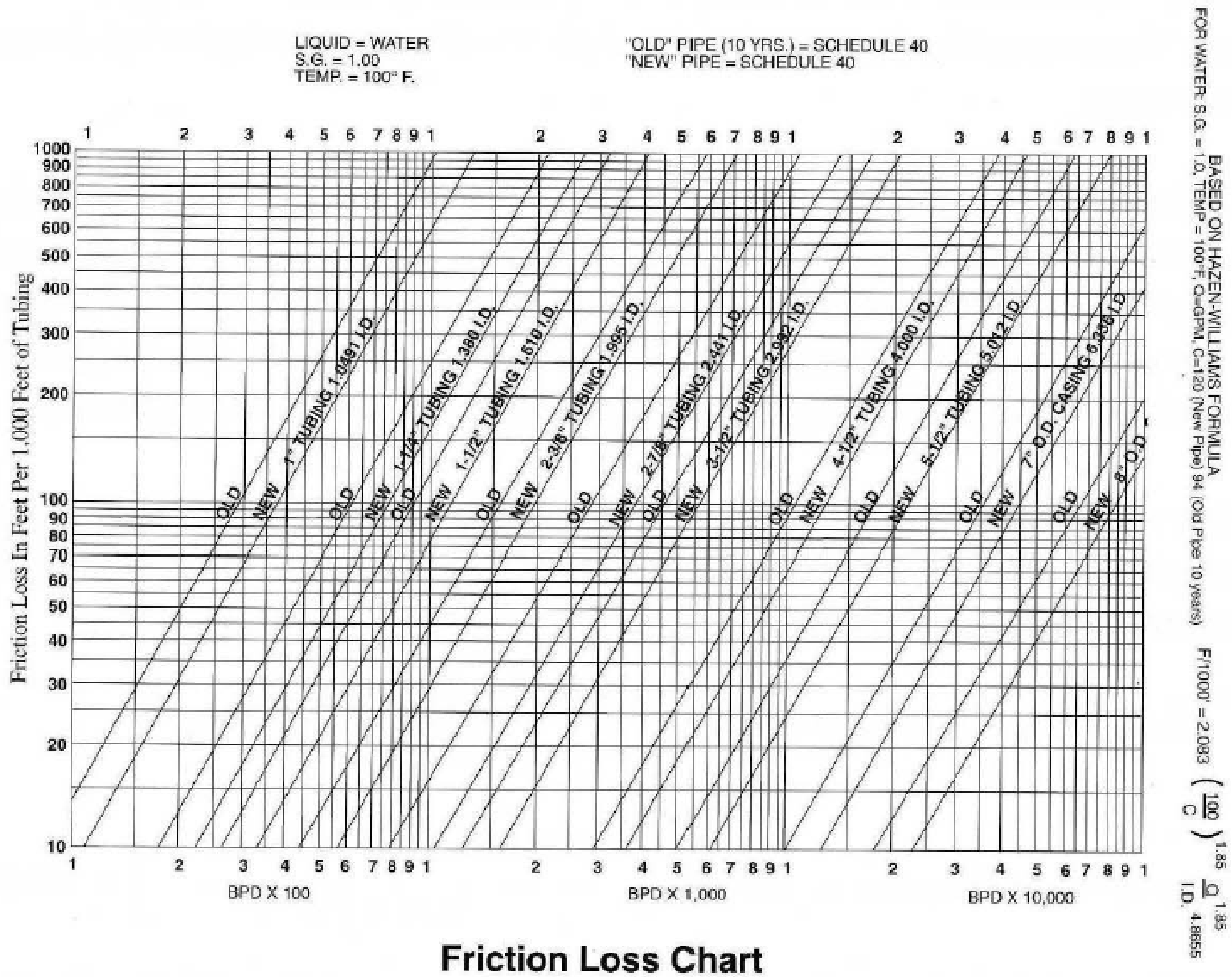

Figure 59. Friction Loss Chart. [6]

According to Figure 59 is friction loss for 5 ½" Old Tubing 10 ft. / 1000 ft. @ 5000 BPD and 75 ft. / 1000 ft. @ 14000 BPD. Pump depth is 6431 ft.

Friction loss for minimum rate $= \dfrac{6431\,ft \cdot 10\,ft}{1000\,ft} = 64\,ft$ (20m)

Friction loss for maximum rate $= \dfrac{6431\,ft \cdot 75\,ft}{1000\,ft} = 482\,ft$ (147m)

As seen in the friction loss chart both losses for old and new tubing are given. Old tubing was chosen since it is believed that this will give a more realistic value. This is because new tubing will be clean and not affected with dirt and normal wear which will occur during production. Naturally, we also see that the friction loss heavily increases with the pump rate.

Assuming that the discharge pressure head (desired wellhead pressure) is the same for both flow rates. Converting wellhead pressure into ft:

$$\text{Wellhead Pressure} = \frac{100\,psi \cdot 2{,}31\,ft\,/\,psi}{0{,}97} = 238\,ft \;\; \text{(73m)}$$

TDH for minimum rate:
1680 ft + 64 ft + 238 ft = 1982 ft (604m)

TDH for maximum rate:
5252 ft + 482 ft + 238 ft = 5972 ft (1820m)

Minimum Hydraulic Requirement	**Maximum Hydraulic Requirement**
Flow Rate 5000 BPD	Flow Rate 14000 BPD
TDH 1982 ft (604m)	TDH 5972 ft (1820m)

Now we have to select a pump that will fit in the casing, have a maximum flow rate of 14000 BPD and is near the best efficient point. Figure 60 illustrates a pump curve from one pump stage for a 675 series pump. The pump has therefore a OD of 6,75” and the yellow lines in Figure 60 shows that we are able to produce the desired production range from 5000 BPD to 14000 BPD.

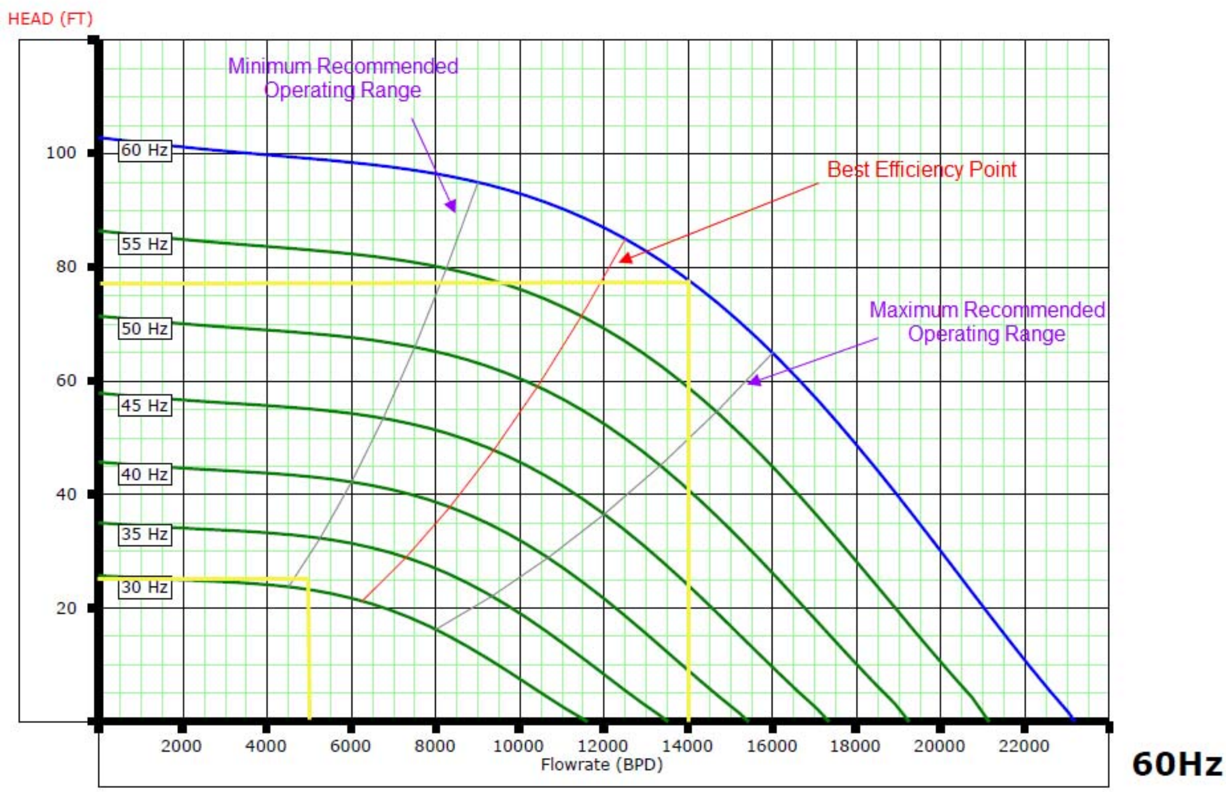

Figure 60. Pump Curve, 0% Water Cut. [51]

Next, we select the head per stage at 60 Hz and 14000 BPD, this is according to Figure 60 approximately 77 ft. /stg. With the maximum total dynamic head of 5972 ft, we calculate the number of pump stages required.

$$\text{No. Stages} = \frac{5972 \text{ ft}}{77 \text{ ft /stg}} = 78$$

To check the point of minimum hydraulic requirement, we divide the minimum TDH by the number of stages selected.

$$\text{Minimum Head / Stage} = \frac{1982 \text{ ft}}{78 \text{ stgs}} = 25 \text{ ft}$$

Then we plot the minimum head/stage (25 ft) and the minimum flow rate (5000 BPD) on the pump curve in Figure 60. This gives a minimum operating frequency of approximately 32,5 Hz. As seen in the figure, this point is just inside the recommended operating range of the pump.

Next, using the VSD Power Curve (Figure 61) for the pump, we read off the BHP/stage at 60 Hz to 10,5 HP/stg. Then we calculate the BHP at the maximum frequency:

BHP @ Max. Hz. = BHP/Stg. @ 60 Hz. x No. Stgs. x SG_L

BHP (60 Hz) = 10,5 Hp x 78 stgs x 0,97 = 794 BHP (584kW)

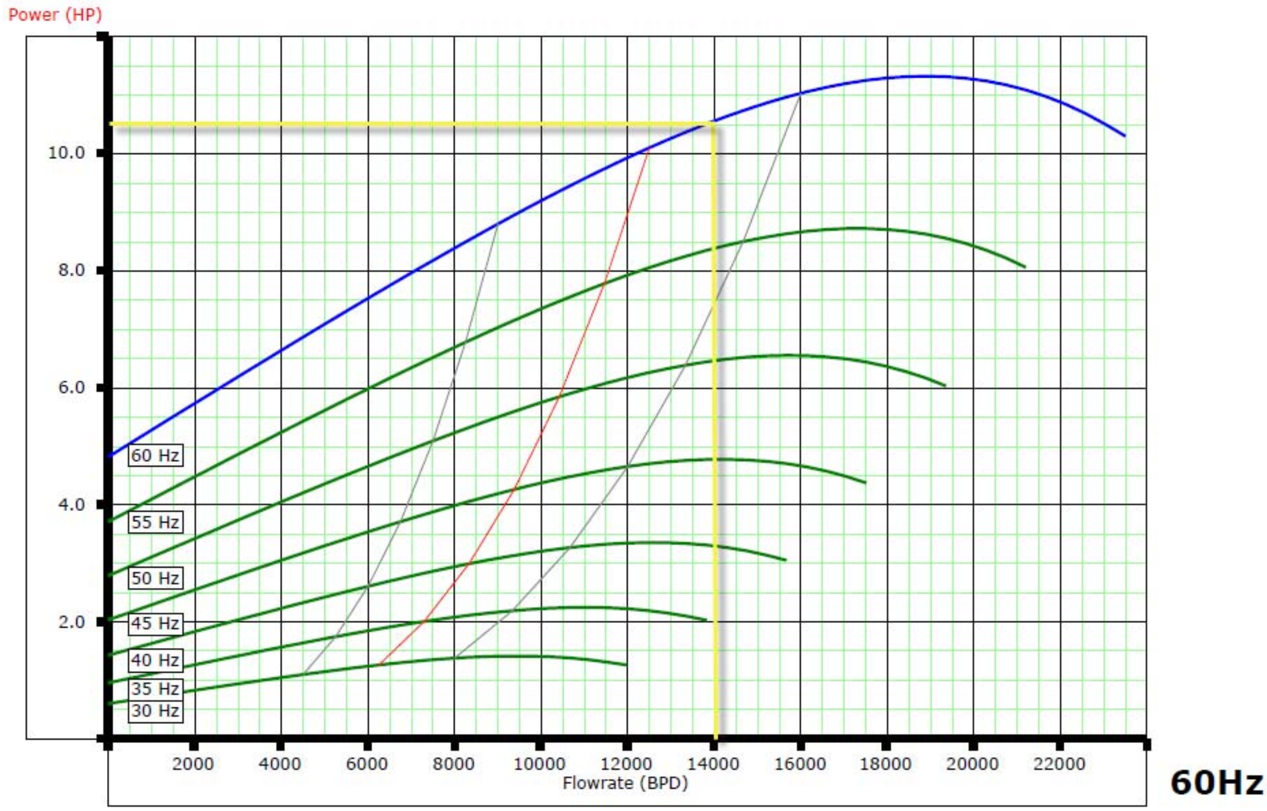

Figure 61. VSD Power Curve. [51]

Figure 61 shows that this particular pump is recommended to run at frequencies between 30 and 60 Hz. If the pump is run below 30 Hz it will cause vibrations and the pump will not run as smoothly. Down-thrust bearings will also be subjected for large forces. If the pump is run above 60 Hz it will lead to shorter lifetime because of increasing wear and tear. There is also physical limitations on the electric motor which can cause problems when exceeding 60 Hz.

For case 2, 3 and 4 only the calculations will be given, since the same procedure as described in case 1 is followed.

Case 2, 30% Water Cut

Water Cut is set to 30%

$$SG_L = SG_W \cdot SG_O$$

Where: SG_L = Specific Gravity of Liquid [-]
SG_W = Specific Gravity of Water x % of water cut [-]
SG_O = Specific Gravity of Oil x % oil [-]

$$SG_L = (0{,}3 \cdot 1{,}05) + (0{,}7 \cdot 0{,}97) = 0{,}99$$

$$\text{PIP for minimum rate} = 2519\,psi - \left[\frac{(7678\,\text{ft} - 6431\,\text{ft}) \cdot 0{,}99}{2{,}31\,ft/psi}\right] = 1985\,psi \;\; (137\text{bar})$$

$$\text{PIP for maximum rate} = 1019\,psi - \left[\frac{(7678\,\text{ft} - 6431\,\text{ft}) \cdot 0{,}99}{2{,}31\,ft/psi}\right] = 485\,psi \;\; (33\text{bar})$$

$$\text{Net lift for minimum rate} = 6431 - \left(\frac{1985\,psi \cdot 2{,}31\,\text{ft/psi}}{0{,}99}\right) = 1799\,ft \;\; (548\text{m})$$

$$\text{Net lift for maximum rate} = 6431 - \left(\frac{485\,psi \cdot 2{,}31\,\text{ft/psi}}{0{,}99}\right) = 5299\,ft \;\; (1615\text{m})$$

$$\text{Wellhead Pressure} = \frac{100\,psi \cdot 2{,}31\,ft/psi}{0{,}99} = 233\,ft \;\; (71\text{m})$$

TDH for minimum rate:
1799 ft + 64 ft + 233 ft = 2096 ft (639m)

TDH for maximum rate:
5299 ft + 482 ft + 233 ft = 6014 ft (1833m)

Minimum Hydraulic Requirement
Flow Rate 5000 BPD
TDH 2096 ft (639m)

Maximum Hydraulic Requirement
Flow Rate 14000 BPD
TDH 6014 ft (1833m)

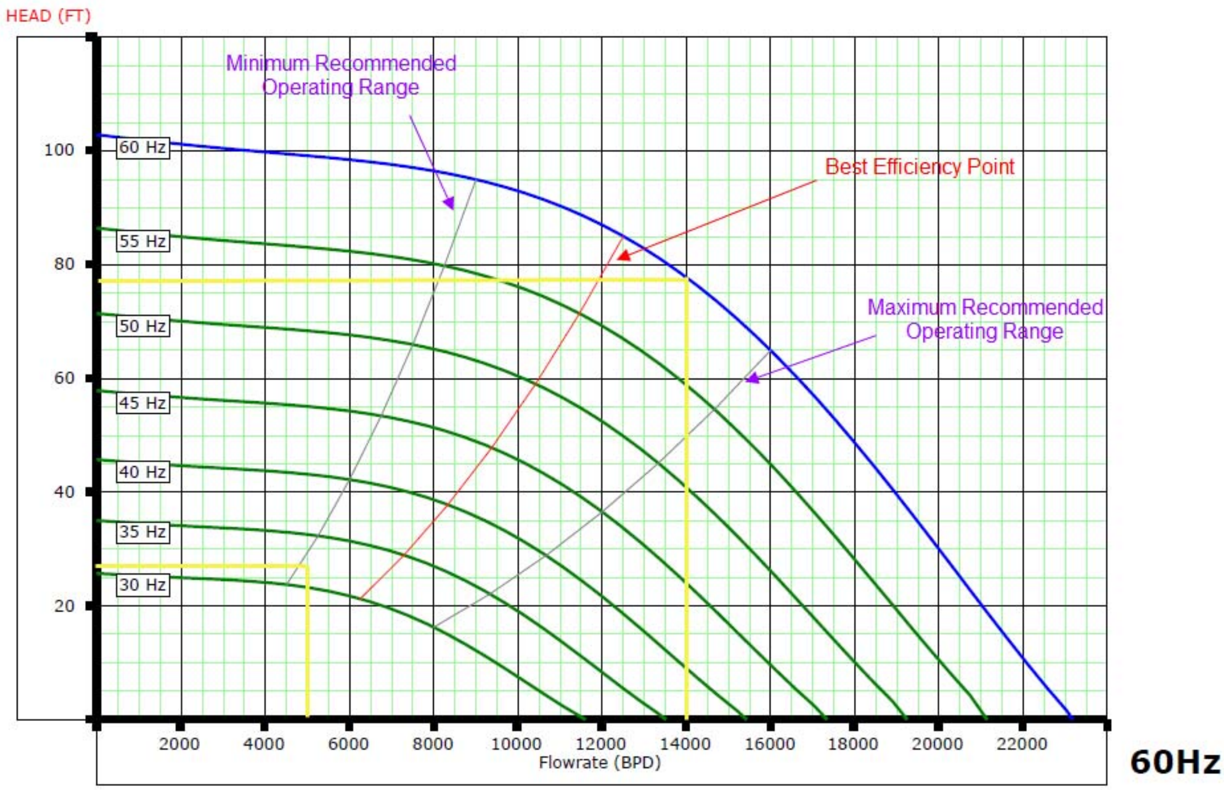

Figure 62. Pump Curve, 30% Water Cut. [51]

$$\text{No. Stages} = \frac{6014\text{ ft}}{77\text{ ft /stg}} = 78$$

To check the point of minimum hydraulic requirement, we divide the minimum TDH by the number of stages selected.

$$\text{Minimum Head / Stage} = \frac{2096\text{ ft}}{78\text{ stgs}} = 27\text{ ft}$$

BHP (60 Hz) = 10,5 Hp x 78 stgs x 0,99 = 811 BHP (596kW)

Case 3, 50% Water Cut

Water Cut is set to 50%
$SG_L = (0{,}5 \cdot 1{,}05) + (0{,}5 \cdot 0{,}97) = 1{,}01$

$$PIP = P_{wf} - \left[\frac{(\text{Datum - Pump Depth}) \cdot SG_L}{2{,}31\, ft / psi} \right]$$

$$\text{PIP for minimum rate } = 2519\, psi - \left[\frac{(7678\,\text{ft} - 6431\,\text{ft}) \cdot 1{,}01}{2{,}31\, ft / psi} \right] = 1979\, psi \ \ (136\text{bar})$$

$$\text{PIP for maximum rate } = 1019\, psi - \left[\frac{(7678\,\text{ft} - 6431\,\text{ft}) \cdot 1{,}01}{2{,}31\, ft / psi} \right] = 474\, psi \ \ (33\text{bar})$$

$$\text{Net lift for minimum rate } = 6431 - \left(\frac{1979\, psi \cdot 2{,}31\,\text{ft/psi}}{1{,}01} \right) = 1905\, ft \ \ (581\text{m})$$

$$\text{Net lift for maximum rate } = 6431 - \left(\frac{474\, psi \cdot 2{,}31\,\text{ft/psi}}{1{,}01} \right) = 5347\, ft \ \ (1630\text{m})$$

$$\text{Wellhead Pressure } = \frac{100\, psi \cdot 2{,}31\, ft / psi}{1{,}01} = 229\, ft \ \ (70\text{m})$$

TDH for minimum rate:
1905 ft + 64 ft + 229 ft = 2198 ft (670m)

TDH for maximum rate:
5347 ft + 482 ft + 229 ft = 6058 ft (1846m)

Minimum Hydraulic Requirement
Flow Rate 5000 BPD
TDH 2198 ft (670m)

Maximum Hydraulic Requirement
Flow Rate 14000 BPD
TDH 6058 ft (1846m)

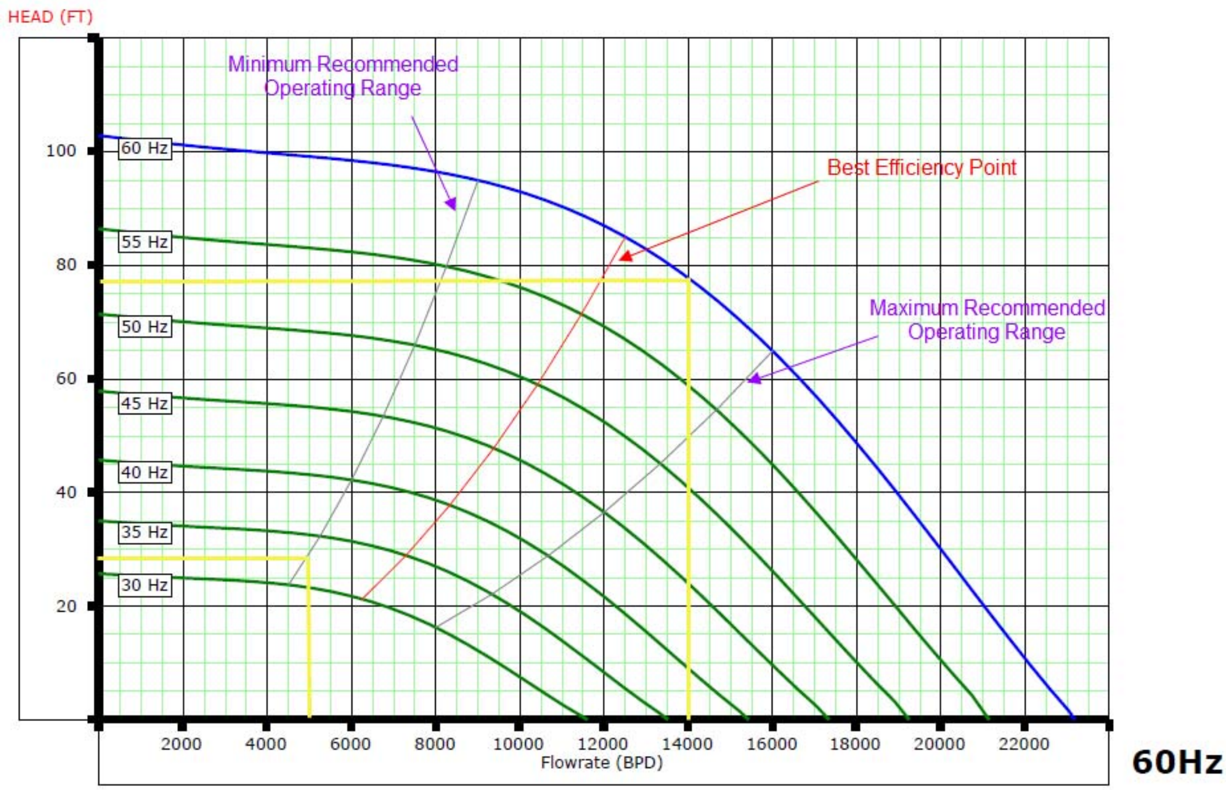

Figure 63. Pump Curve, 50% Water Cut. [51]

$$\text{No. Stages} = \frac{\text{Maximum Total Dynamic Head}}{\text{Head / Stage}}$$

$$\text{No. Stages} = \frac{6058\text{ ft}}{77\text{ ft /stg}} = 79$$

$$\text{Minimum Head / Stage} = \frac{2198\text{ ft}}{79\text{ stgs}} = 28\text{ ft}$$

BHP (60 Hz) = 10,5 Hp x 79 stgs x 1,01 = 838 BHP (616kW)

Case 4, 95% Water Cut

Water Cut is set to 95%

$SG_L = (0{,}95 \cdot 1{,}05) + (0{,}05 \cdot 0{,}97) = 1{,}05$

PIP for minimum rate $= 2519\,psi - \left[\dfrac{(7678\text{ ft} - 6431\text{ ft}) \cdot 1{,}05}{2{,}31\,ft/psi}\right] = 1952\,psi$ (135bar)

PIP for maximum rate $= 1019\,psi - \left[\dfrac{(7678\text{ ft} - 6431\text{ ft}) \cdot 1{,}05}{2{,}31\,ft/psi}\right] = 452\,psi$ (31bar)

Net lift for minimum rate $= 6431 - \left(\dfrac{1952\,psi \cdot 2{,}31\text{ ft/psi}}{1{,}05}\right) = 2137\,ft$ (651m)

Net lift for maximum rate $= 6431 - \left(\dfrac{452\,psi \cdot 2{,}31\text{ ft/psi}}{1{,}05}\right) = 5437\,ft$ (1657m)

Wellhead Pressure $= \dfrac{100\,psi \cdot 2{,}31\,ft/psi}{1{,}05} = 220\,ft$ (67m)

TDH Minimum Rate:
2137 ft + 64 ft + 220 ft = 2421 ft (738m)

TDH Maximum Rate:
5437 ft + 482 ft + 220 ft = 6139 ft (1871m)

Minimum Hydraulic Requirement
Flow Rate 5000 BPD
TDH 2421 ft (738m)

Maximum Hydraulic Requirement
Flow Rate 14000 BPD
TDH 6139 ft (1871m)

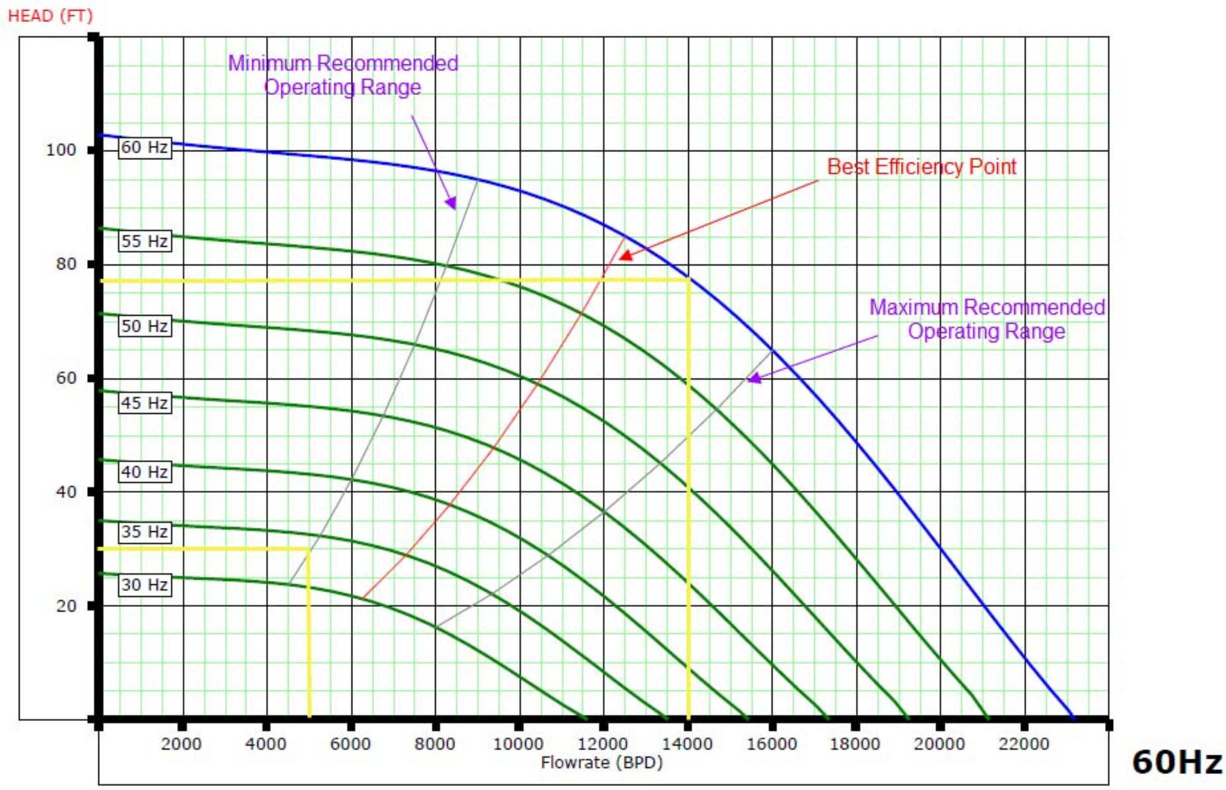

Figure 64. Pump Curve, 95% Water Cut. [51]

$$\text{No. Stages} = \frac{6139\ \text{ft}}{77\ \text{ft /stg}} = 80$$

To check the point of minimum hydraulic requirement, we divide the minimum TDH by the number of stages selected.

$$\text{Minimum Head / Stage} = \frac{2421\ \text{ft}}{80\ \text{stgs}} = 30\ \text{ft}$$

BHP (60 Hz) = 10,5 Hp x 80 stgs x 1,05 = 882 BHP (649kW)

Results

Case 1, 0% Water Cut:
SG_L = 0,97
BHP = 794 (584kW)
Number of pump stages = 78

Since we in this case are pumping the lightest fluid (SG_L = 0,97), we also need the smallest motor effect (584 kW). As seen in Figure 60 we are just inside the recommended operating range when pumping the minimum pump range of 5000 BPD.

Case 2, 30% Water Cut:
SG_L = 0,99
BHP = 811 (596kW)
Number of pump stages = 78

30% water cut results in heavier fluid (SG_L = 0,99) which requires higher motor effect (596 kW) than in case 1. A seen in Figure 62 we are just inside the recommended operating range when pumping 5000 BPD.

Case 3, 50% Water Cut:
SG_L = 1,01
BHP = 838 (616kW)
Number of pump stages = 79

50% water cut results in heavier fluid (SG_L = 1,01) which requires higher motor effect (616 kW) than in case 2. As seen in Figure 63 we are just touching the minimum recommended operating range when pumping 5000 BPD.

Case 4, 95% Water Cut:
SG_L = 1,05
BHP = 882 (649kW)
Number of pump stages = 80

95% water cut results in the highest motor effect requirement (649kW) since we are pumping the heaviest fluid (SG_L = 1,05). As seen in Figure 64 are we also in this case just touching the minimum recommended operating range when pumping 5000 BPD.

Case 1 gives the best operating conditions, because we are nearest the best efficiency point through the whole operating range from 5000 BP to 14000 BPD. Case 4 is the worst scenario since we are almost outside the recommended operating range and furthest away from the best efficiency point. However, as seen from the pump curves are there very small distinctions between the four cases. The conclusion is therefore that the pump selected would be appropriate for this well.

Appendix B. Sizing with VSD for medium well in Peregrino, considering gas

Case 5, 30% Water Cut with free gas (GOR = 70)

This section is an example on how to perform a sizing for ESP systems with a VSD, considering oil, water and gas at the pump. The sizing will be performed for well fluids with 30% water cut. Table 19 shows the reservoir and well data given by Statoil for a particular well in the Peregrino field.

Medium Well ESP Design Parameters		
Oil Gravity	14	˚API
Specific Gravity of Water	1,05	(100,000 ppm)
Productivity Index (PI)	6,0	BPD/Psi
Specific Gravity of Gas	0,77	Rel to air
Gas Oil Ratio	70	scf/STB
Oil Bubble point	700	Psia @ 183
Reservoir Temperature	173,5	deg. F
Wellhead Pressure	100	Psi
Water Cut	30 %	
Expected Production rate	5000 - 14000	BPD
Initial Reservoir Pressure	3352	Psia
Datum	7678	ft TVD MSL
Pump Setting Depth	6431	ft
Production casing	9 5/8" #47 lb/ft	
Production Tubing	5 1/2" #17 lb/ft	

Table 19. Process data [31]

The calculations in the following sizing procedure are in American units, and not SI units. This is because the formulas originate from API standards. However the answers will also be given in SI units.

First we have to determine the total fluid mixture, inclusive of water, oil and free gas that will be ingested by the pump. Then determine the solution gas/oil ratio (R_S) at the pump intake pressure by substituting the pump intake pressure for the bubble-point pressure (P_b) in Standing’s equation:

$$R_S = Y_g \cdot \left(\frac{P_b}{18} \cdot \frac{10^{0.0125 \cdot {}^\circ API}}{10^{0.00091 \cdot T}} \right)^{1.2048}$$

Where: R_S = Gas/oil ratio at the pump intake [-]
Y_g = Specific Gravity of Gas [-]
P_b = Bubble-point pressure [psi]
˚API = Oil Gravity [-]
T = Reservoir Temperature [˚F]

$$R_S = 0.77 \cdot \left(\frac{700}{18} \cdot \frac{10^{0.0125 \cdot 14}}{10^{0.00091 \cdot 173.5}} \right)^{1.2048} = 66$$

Next, we determine the formation volume factor (B_o) using the R_S from above and Standing's equation as follows:

$$B_o = 0.972 + 0.000147 \cdot F^{1.175}$$

Where: B_o = Formation Volume Factor [-]

$$F = R_S \cdot \left(\frac{Y_g}{Y_o} \right)^{0.5} + 1.25 \cdot T$$

$$F = 66 \cdot \left(\frac{0.77}{1.2} \right)^{0.5} + 1.25 \cdot 173.5 = 269.7$$

Then we can calculate B_o;

$$B_o = 0.972 + 0.000147 \cdot 269.7^{1.175} = 1.08$$

Next, we determine the gas volume factor (B_g) as follows:

$$B_g = \frac{5.04 \cdot Z \cdot T}{P}$$

Where: B_g = Gas Volume Factor [-]
Z = Compressibility factor of gas [-]
T = Reservoir Temperature (rankine temperature) [˚F]
P = Pressure at standard conditions [1atm]

The absolute zero version of the Fahrenheit scale is the Rankine scale. Add 460 degrees to Fahrenheit temperatures to obtain the Rankine temperature.

Assuming 0.85 Z factor;

$$B_g = \frac{5.04 \cdot 0.85 \cdot (460 + 173.5)}{1.014} = 2.67$$

Then we determine the total volume of fluids and the percentage of free gas released at the pump intake:

- Using the producing GOR, and oil volume, calculate the total volume of gas (T_G);

$$T_G = \frac{BPD \cdot GOR}{1000}$$

Where: T_G = Total Volume of Gas [mcf]
BPD = Maximum desired production range x % of oil [BPD]
GOR = Producing GOR [-]

$$T_G = \frac{(14000 \cdot 0.7) \cdot 70}{1000} = 686 \text{ mcf } (19.4 \text{ m}^3)$$

- Using the solution GOR (R_S), at the pump intake, calculate the solution gas (S_G);

$$S_G = \frac{BPD \cdot R_S}{1000}$$

$$S_G = \frac{(14000 \cdot 0.7) \cdot 66}{1000} = 647 \text{ mcf } (18.3 \text{ m}^3)$$

- The difference between T_G and S_G represents the volume of free gas (F_G) released from solution by the decrease in pressure from bubble-point pressure of 700 psi, to the pump intake pressure of 393 psi. The pump intake pressure was calculated in Appendix A, case 2 (30% WC).

$$F_G = T_G - S_G$$

$$F_G = 686 - 647 = 39 \text{ mcf } (1.1 \text{ m}^3)$$

- The volume of oil (V_o), at the pump intake:

$$V_o = BPD \cdot B_o$$

$V_o = (14000 \cdot 0.7) \cdot 1.08 = 10584$ BOPD (1683 m^3/d)

- The volume of free gas (V_g) at the pump intake:

$$V_g = F_G \cdot B_g$$

$$V_g = 39 \cdot 2.67 = 104 \text{ BGPD } (2.9 \text{ m}^3/\text{d})$$

- The volume of water (V_w), at the pump intake:

V_w = Total fluid volume x % water

$$V_w = 14000 BPD \cdot 0.3 = 4200 \text{ BWPD } (667.7 \text{ m}^3/\text{d})$$

- The total volume (V_t) of oil, water and gas, at the pump intake, can now be determined:

$$V_t = V_o + V_g + V_w$$

$$V_t = 10584 BOPD + 104 BGPD + 4200 BWPD = 14888 \text{ BFPD } (2366.9 \text{ m}^3/\text{d})$$

- The ratio, or percentage of free gas present at the pump intake to the total volume of fluid is:

$$\% \text{ Free Gas} = \frac{V_g}{V_t}$$

$$\% \text{ Free Gas} = \frac{104 BGPD}{14888 BFPD} \cdot 100 = 0.69\%$$

Because this value is less than 10% by volume, it has little effect on pump performance, however the gas can effect the well fluid composite specific gravity at the pump.

- The composite specific gravity, including gas, can be determined calculating the total mass of produced fluid (TMPF) from the original data given:

TMPF = {(BOPD x SG_o + BWPD x SG_w) x 62.4 x 5.6146} + (GOR x BOPD x SG_g x 0.0752)

TMPF = {(14000 x 0.7 x 0,97 + 4200 x 1.085) x 62.4 x 5.6146} + (70 x 14000 x 0.7 x 0.77 x 0.0752) = 4966709 lbs /day (2252862 kg/day)

Now we can calculate the specific gravity of the fluid being pumped:

$$SG_L = \frac{TMPF}{BFPD \cdot 5.6146 \cdot 62.4}$$

$$SG_L = \frac{4966709}{14888 \cdot 5.6146 \cdot 62.4} = 0,95$$

Now we can proceed with following the procedure described in Appendix A for 30% water cut.

$$\text{PIP for minimum rate} = 2519\,psi - \left[\frac{(7678\,\text{ft} - 6431\,\text{ft}) \cdot 0,95}{2,31\,ft/psi}\right] = 2006\,psi \ \ (138\text{bar})$$

$$\text{PIP for maximum rate} = 1019\,psi - \left[\frac{(7678\,\text{ft} - 6431\,\text{ft}) \cdot 0,95}{2,31\,ft/psi}\right] = 506\,psi \ \ (35\text{bar})$$

$$\text{Net lift for minimum rate} = 6431 - \left(\frac{2006\,psi \cdot 2,31\,\text{ft/psi}}{0,95}\right) = 1553\,ft \ \ (473\text{m})$$

$$\text{Net lift for maximum rate} = 6431 - \left(\frac{506\,psi \cdot 2,31\,\text{ft/psi}}{0,95}\right) = 5201\,ft \ \ (1585\text{m})$$

$$\text{Wellhead Pressure} \ = \frac{100\,psi \cdot 2,31\,ft/psi}{0,95} = 243\,ft \ \ (74\text{m})$$

TDH for minimum rate:
1553 ft + 64 ft + 243 ft = 1860 ft (567m)

TDH for maximum rate:
5201 ft + 482 ft + 243 ft = 5926 ft (1806m)

Minimum Hydraulic Requirement
Flow Rate 5000 BPD
TDH 1860 ft (567m)

Maximum Hydraulic Requirement
Flow Rate 14000 BPD
TDH 5926 ft (1806m)

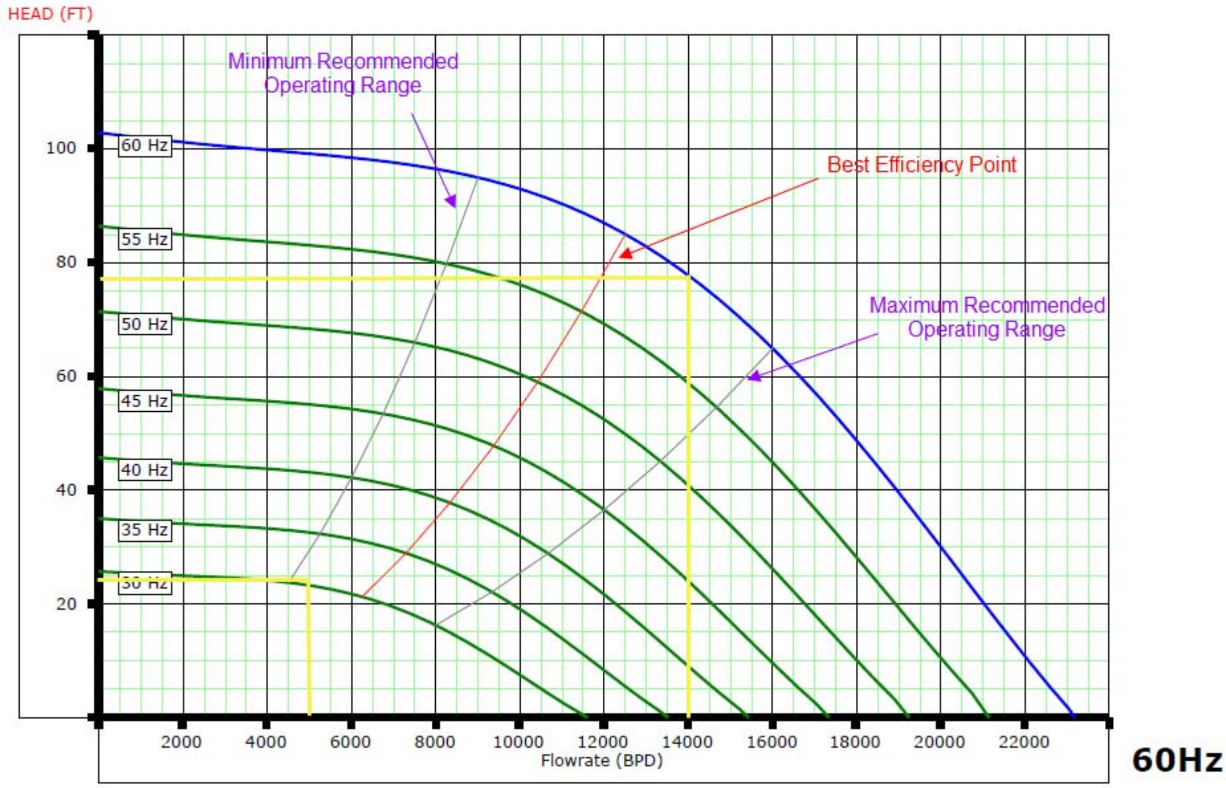

Figure 65. Pump Curve for Case 5. [51]

$$\text{No. Stages} = \frac{5926\ \text{ft}}{77\ \text{ft /stg}} = 77$$

To check the point of minimum hydraulic requirement, we divide the minimum TDH by the number of stages selected.

$$\text{Minimum Head / Stage} = \frac{1860\ \text{ft}}{77\ \text{stgs}} = 24\ \text{ft}$$

BHP (60 Hz) = 10,5 Hp x 77 stgs x 0,95 = 768 BHP (565kW)

Case 6, 30% Water Cut with free gas (GOR = 140)

To further investigate the effect of free gas in the production fluid we will consider a GOR of 140 instead of 70,

- Using the producing GOR, and oil volume, calculate the total volume of gas (T_G);

$$T_G = \frac{BPD \cdot GOR}{1000}$$

Where: T_G = Total Volume of Gas [mcf]
BPD = Maximum desired production range x % of oil [BPD]
GOR = Producing GOR [-]

$$T_G = \frac{(14000 \cdot 0.7) \cdot 140}{1000} = 1372\ \text{mcf}\ (38.9\ \text{m}^3)$$

- Using the solution GOR (R_S), at the pump intake, calculate the solution gas (S_G);

$$S_G = \frac{BPD \cdot R_S}{1000}$$

$$S_G = \frac{(14000 \cdot 0.7) \cdot 66}{1.000} = 647\ \ \text{mcf}\ (18.3\ \text{m}^3)$$

- $F_G = T_G - S_G$

$$F_G = 1372 - 647 = 725\ \text{mcf}\ (20.5\ \text{m}^3)$$

- The volume of oil (V_o), at the pump intake:

$$V_o = BPD \cdot B_o$$

$$V_o = (14000 \cdot 0.7) \cdot 1.08 = 10854 \text{ BOPD } (1682.7 \text{ m}^3\text{/d})$$

- The volume of free gas (V_g) at the pump intake:

$$V_g = F_G \cdot B_g$$

$$V_g = 725 \cdot 2.67 = 1936 \text{ BGPD } (307.8 \text{ m}^3\text{/d})$$

- The volume of water (V_w), at the pump intake:

V_w = Total fluid volume x % water

$$V_w = 14000 BPD \cdot 0.3 = 4200 \text{ BWPD } (667.7 \text{ m}^3\text{/d})$$

- The total volume (V_t) of oil, water and gas, at the pump intake, can now be determined:

$$V_t = V_o + V_g + V_w$$

$$V_t = 10854 BOPD + 1936 BGPD + 4200 BWPD = 16990 \text{ BFPD } (2701.1 \text{ m}^3\text{/d})$$

- The ratio, or percentage of free gas present at the pump intake to the total volume of fluid is:

$$\text{\% Free Gas} = \frac{V_g}{V_t}$$

$$\text{\% Free Gas} = \frac{1936 BGPD}{16990 BFPD} \cdot 100 = 11.4\%$$

- The composite specific gravity, including gas, can be determined calculating the total mass of produced fluid (TMPF) from the original data given:

  TMPF = {(BOPD x SG_o + BWPD x SG_w) x 62.4 x 5.6146} + (GOR x BOPD x SG_g x 0.0752)

  TMPF = {(14000 x 0.7 x 0,97 + 4200 x 1.085) x 62.4 x 5.6146} + (140 x 14000 x 0.7 x 0.77 x 0.0752) = 4999209 lbs /day (2267603 kg/day)

Now we can calculate the specific gravity of the fluid being pumped:

$$SG_L = \frac{TMPF}{V_t \cdot 5.6146 \cdot 62.4}$$

$$SG_L = \frac{4999209}{16990 \cdot 5.6146 \cdot 62.4} = 0.84$$

Then we can proceed with following the procedure described in case 5.

$$\text{PIP for minimum rate} = 2519\,psi - \left[\frac{(7678\,\text{ft} - 6431\,\text{ft}) \cdot 0.84}{2{,}31\,ft/psi}\right] = 2066\,psi \;\; (142\text{bar})$$

$$\text{PIP for maximum rate} = 1019\,psi - \left[\frac{(7678\,\text{ft} - 6431\,\text{ft}) \cdot 0.84}{2{,}31\,ft/psi}\right] = 566\,psi \;\; (39\text{bar})$$

$$\text{Net lift for minimum rate} = 6431 - \left(\frac{2066\,psi \cdot 2{,}31\,\text{ft/psi}}{0.84}\right) = 750\,ft \;\; (228\text{m})$$

$$\text{Net lift for maximum rate} = 6431 - \left(\frac{566\,psi \cdot 2{,}31\,\text{ft/psi}}{0.84}\right) = 4875\,ft \;\; (1486\text{m})$$

$$\text{Wellhead Pressure} \; = \frac{100\,psi \cdot 2{,}31\,ft\,/\,psi}{0.84} = 275\,ft \;\; (84\text{m})$$

TDH for minimum rate:
750 ft + 64 ft + 275 ft = 1089 ft (332m)

TDH for maximum rate:
4875 ft + 482 ft + 275 ft = 5632 ft (1717m)

Minimum Hydraulic Requirement	**Maximum Hydraulic Requirement**
Flow Rate 5000 BPD	Flow Rate 14000 BPD
TDH 1089 ft (332m)	TDH 5632 ft (1717m)

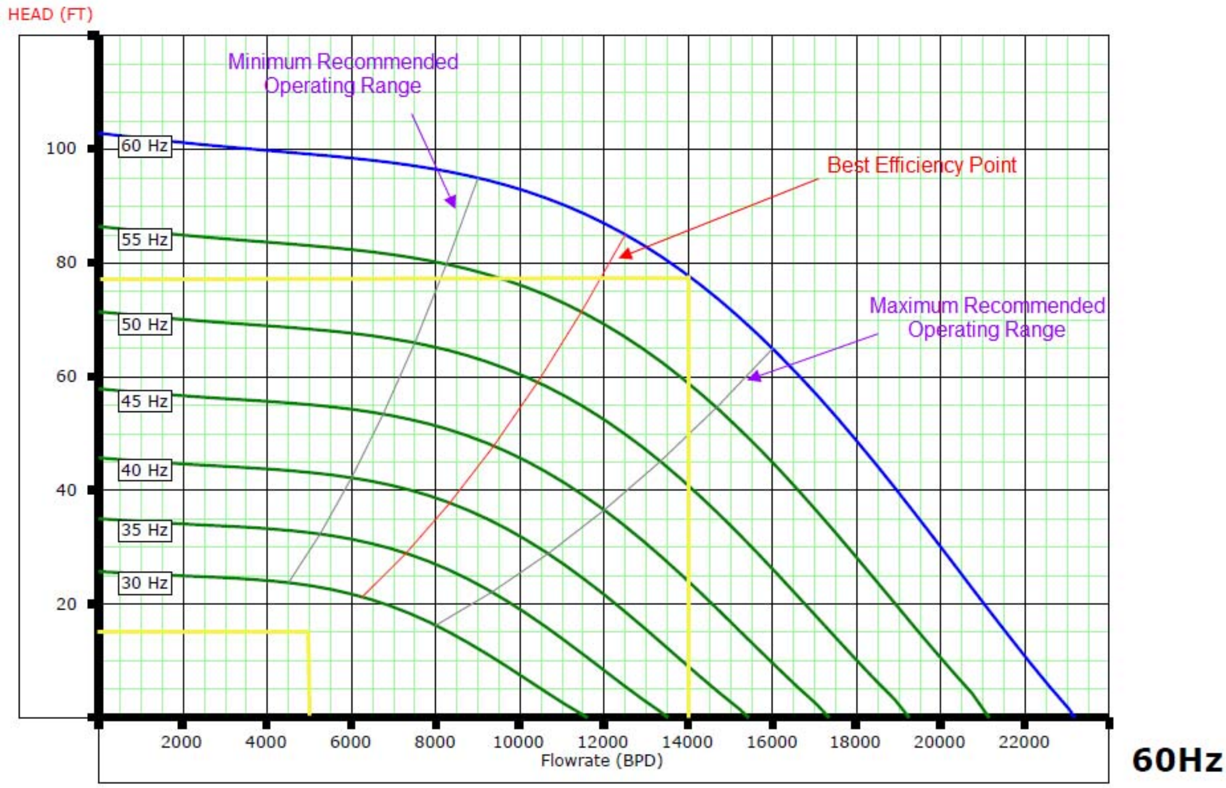

Figure 66. Pump Curve for Case 6. [51]

$$\text{No. Stages} = \frac{5632\ \text{ft}}{77\ \text{ft /stg}} = 73$$

To check the point of minimum hydraulic requirement, we divide the minimum TDH by the number of stages selected.

$$\text{Minimum Head / Stage} = \frac{1089\ \text{ft}}{73\ \text{stgs}} = 15\ \text{ft}$$

BHP (60 Hz) = 10,5 Hp x 73 stgs x 0.84 = 644 BHP (473kW)

Results

Case 5, 30% Water cut with free gas (GOR = 70)
SG_L = 0.95
BHP = 768 (565kW)
Number of pump stages = 77
Free gas in the fluid = 0.69%

Compared to case two in Appendix A, can we see that the specific gravity is a bit lower (0,95 against 0,99) and hence require less power (565 kW against 596 kW) which result in one pump stage less than for case 2 (77 against 78). According to Figure 66 are we just inside the recommended operating range when producing 5000 BPD. This proves that it's not necessary to utilize a gas separator when we are producing fluids with a gas/oil ratio of 70. It should however be installed a separator to be on the safe side, since well conditions never are static and will change over time.

Case 6, 30% Water cut with free gas (GOR = 140)
SG_L = 0.84
BHP = 644 BHP (473kW)
Number of pump stages = 73
Free gas in the fluid = 11.4%

As seen from the results is the specific gravity reduced significantly when considering a GOR of 140 instead of 70. This is because gas has a lower specific gravity than oil and water. A lighter fluid to be lifted means less power required, only 473 kW. As seen in Figure 66, are we below the frequency range when pumping 5000 to 8100 BPD. A solution is to install a gas separator before the pump-intake to be able to pump the desired production range with this particular pump. The presence of free gas at the pump also depends on the pump intake pressure (PIP). If the PIP is higher or equal to the bubble-point pressure, no free gas enters the pump. The gas is kept inside the fluid. If the PIP falls below the bubble-point pressure than gas is evolved from the oil and we get free gas at the pump intake.

Appendix C. Calculation charts for LCC

	Real discount rate (interest rate minus inflation rate, in percent)												
No. of years (n)	−2	−1	0	1	2	3	4	5	6	7	8	9	10
1	1.02	1.01	1.00	0.99	0.98	0.97	0.96	0.95	0.94	0.93	0.93	0.92	0.91
2	2.06	2.03	2.00	1.97	1.94	1.91	1.89	1.86	1.83	1.81	1.78	1.76	1.74
3	3.12	3.06	3.00	2.94	2.88	2.83	2.78	2.72	2.67	2.62	2.58	2.53	2.49
4	4.21	4.10	4.00	3.90	3.81	3.72	3.63	3.55	3.47	3.39	3.31	3.24	3.17
5	5.31	5.15	5.00	4.85	4.71	4.58	4.45	4.33	4.21	4.10	3.99	3.89	3.79
6	6.44	6.22	6.00	5.80	5.60	5.42	5.24	5.08	4.92	4.77	4.62	4.49	4.36
7	7.60	7.29	7.00	6.73	6.47	6.23	6.00	5.79	5.58	5.39	5.21	5.03	4.87
8	8.77	8.37	8.00	7.65	7.33	7.02	6.73	6.46	6.21	5.97	5.75	5.53	5.33
9	9.97	9.47	9.00	8.57	8.16	7.79	7.44	7.11	6.80	6.52	6.25	6.00	5.76
10	11.19	10.57	10.00	9.47	8.98	8.53	8.11	7.72	7.36	7.02	6.71	6.42	6.14
15	17.20	16.27	15.00	13.87	12.85	11.94	11.12	10.38	9.71	9.11	8.56	8.06	7.61
20	24.89	22.26	20.00	18.05	16.35	14.88	13.59	12.46	11.47	10.59	9.82	9.13	8.51
25	32.85	28.56	25.00	22.02	19.52	17.41	15.62	14.09	12.78	11.65	10.67	9.82	9.08
30	41.66	35.19	30.00	25.81	22.40	19.60	17.29	15.37	13.76	12.41	11.26	10.27	9.43

Table 20. Discount factor for yearly expenditures. [50]

Appendix D. Scope of work

Tittel på oppgave: Artificial Lift -Electrical Submerged Pump, best practice and future demands within subsea applications
Oppgavetekst (Scope of work) med aktiviteter og leveranser: Mye av eksisterende oljereserver er olje med høy viskositet som ligger i krevende reservoar. Det er svært viktig at utvinning av denne oljen foregår på en mest mulig kostnadseffektiv og miljøvennlig måte. Nedihullspumper, såkalte ESP'er (Electrical submerged pumps), har vært brukt i en årrekke for dette formålet. I fremtiden stilles det krav til høy effekt, store reservoartrykk og -temperatur. Oppgaven skal gi en oversikt over status i teknologi for pumpesystem med fokus på oppbygging, design, drift og regularitet. Både hovedteknologi og eventuell eksisterende alternative og kostnadseffektive teknologier skal identifiseres og diskuteres. Oppgaven skal føre frem til et forslag til design for ESP pumpesystem for bruk både i subsea oljebrønner og på landbaserte tungoljefelt. **Omfang** 1) Det skal først gis en kort beskrivelse av oppbygging og virkemåte av forskjellige typer ESP'er inkl. kontrollsystem. Beskrivelsen skal også inkludere en identifisering av drift og vedlikehold. Det skal tas utgangspunkt i ESP pumpesystem som brukes i subsea brønner, men oppgaven skal også omfatte bruk av ESP'er i landbaserte tungoljefelt. 2) Det skal foretas et begrenset litteratursøk / litteraturstudium for å finne ut hva som er publisert både med hensyn på teori og praksis for ESP'er. Videre bør det legges spesiell vekt på erfaringsdata fra relevante anlegg (pumper som er i drift) 3) I del tre av oppgaven skal kandidaten foreslå beste praksis for ESP pumpesystem i et utfordrende tilfelle. Det skal etableres beste praksis for: - Pumpespesifikasjoner - Tilstandsovervåking - Livssykluskostnader og inntjening Kandidaten skal innsende til instituttet en detaljert arbeidsplan, samt et skjema over disponeringen av den tid kandidaten har til rådighet for oppgavens bearbeidelse. Dokumentasjon av arbeidet gjøres mest mulig i form av en forskningsrapport med sammendrag både på norsk og engelsk. Videre kreves det konklusjon, referanseliste, innholdsfortegnelse etc.. Ved utarbeidelse til teksten skal kandidaten legge vekt på å gjøre teksten oversiktlig. Ved bedømmelsen legges det stor vekt på at resultatene er grundig bearbeidet og at numeriske resultater presenteres tabellarisk / grafisk på en oversiklig måte og diskuteres utførlig. Det legges vekt på at alle benyttede kilder, inklusive muntlige kilder, oppgis fullstendig.
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