SIZING AND SELECTION CRITERIA FOR SUBSEA MULTIPHASE PUMPS

A Thesis Presented to

The Faculty of the Department of Engineering Technology

University of Houston

In Partial Fulfillment

of the Requirements for the Degree

Master of Science in Engineering Technology

By

Manoochehr Bozorgmehrian

May 2013

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ABSTRACT

The main application of subsea multiphase pumps is in mature assets; when the reservoir pressure is not high enough to boost the oil flow, or in the green fields or tie-backs where the host facilities are far from the well head. In all subsea boosting projects, the pumps are expected to work non-stop for a long period of time (3-7 years) without any intervention. To satisfy this requirement alone, a highly reliable integrated system including electric motors, sealing parts, couplings, bearings and connections is necessary.

There are no published standards for subsea pump selection and sometimes there is a big gap between proven experience and what is claimed by the manufacturers. An independent research study that covers all parameters and priorities can be helpful. In the present research, all available operator or manufacturer's reports and published papers were reviewed and a criterion was developed based on these steps:

1- Since the evaluation of the various lifting and boosting methods is the prerequisite of pump selection and sizing, the conventional subsea separation system was compared to a multi-phase pumping one.
2- All applicable types of subsea pumps including Helico-Axial (HAP), Twin Screw Pump (TSP) and Electrical Submersible Pump (ESP) were compared, and their pros and cons were discussed. The auxiliary equipment of each one such as conditioning tank and bypass line were also explained. The characteristic curves and applicable operating ranges of each pump were analyzed.

3- Sometimes different suppliers offer a wide range of products, which need to be verified for specific working conditions; therefore all parameters in pump type selection such as Gas Volume Fraction (GVF), water cut, differential pressure, head, flow rate, fluid viscosity, RPM, sand content, water depth, Opex, Capex, reliability and asset life were discussed.

4- Material selection as a critical part of the pump selection was studied thoroughly. In this regard, all applicable types of corrosion and cracking including CO_2 corrosion, Pitting and Crevice corrosion, Galvanic corrosion, Microbial Induced Corrosion (MIC), Hydrogen Induced Cracking and Sulfide Stress Cracking were studied to determine which part of the pump is susceptible to which types of corrosion. Material selection was conducted based on this survey and applicable standards such as Norsok M001 and NACE MR0175. Finally, applicable material grades were introduced.

5- Above mentioned parameters and the latest project records have been applied in designing a practical procedure which determines the priority of the parameters and the sequence of pump selection process. Applicable calculations and requirements of API 610, API 17A and DNV A-203 standards have also been incorporated. Measures were introduced to prevent some operational problems reported in historical records.

6- The procedure begins with the study of flow regime to calculate the differential pressure of the pump. Then it leads to calculating the shaft power. Other steps are dedicated to the specific pump type, whether it is Helico-Axial or twin screw. The calculations are more dependent on the vendor data for TSP; however, the formulas that can lead to proper sizing such as axial velocity, efficiency, slip and power were provided. Balancing, sealing and coupling standards that are applicable to both types were also explained at the end of the study.

TABLE OF CONTENTS

LIST OF ACRONYMS

ANSI	American National Standard Institute
API	American Petroleum Institute
ASME	American Society of Mechanical Engineers
BHFP	Bottom Hole Flowing Pressure
BHP	Brake Horse Power
BPD	Barrel Per Day
BPED	Barrel Per Day Equivalent
CFC	Corrosion Fatigue Cracking
DNV	Det Norske Veritas
DP	Differential Pressure
DSH	Data Sheet
EIA	Energy Information Administration
EOR	Enhanced Oil Recovery
ESP	Electric Submersible pump
FAT	Fabrication and Assembly Test
FEED	Front End Engineering Design
GVF	Gas Volume (Void) Fraction
HAP	Helico-Axial Pump
HIC	Hydrogen Induced Cracking
IRR	Inflow Performance Relationship
Lg	Long Term

MTBF	Mean Time Between Failure
NPSH	Net Positive Suction Head
Nq	Specific Speed
P&ID	Piping & Instrument Diagram
PCP	Progressive Cavity Pump
PFD	Process Flow Diagram
PI	Productivity Index
PMS	Piping Material Spec
PQR	Procedure Qualification Records
RPM	Revolutions per Minute
SCC	Stress Corrosion Cracking
SG	Specific Gravity
SSC	Sulfide Stress Cracking
T.B	Tieback
TSP	Twin Screw Pump
VSD	Variable Speed Drive
WOR	Water Oil Ratio
WPS	Welding Procedure Specification

LIST OF FIGURES

LIST OF TABLES

CHAPTER 1

INTRODUCTION

1.1 Introduction

Continuous depletion of onshore oil and gas resources led oil companies to begin developing offshore assets in 1940, which have been developing during previous decades. Now offshore oil and gas produce more than 30% of oil and gas on the market. An Energy Information Administration (EIA) research report that represents the future pattern of oil production is given in Table 1. It shows that until 2030, a significant share of oil and gas production will be from offshore reservoirs.

Table 1: Oil Production Capacity in Future (Sandrea, 2009)

	EIA Model	Oil Production	Capacity	Model
Year	Total	Total	Onshore	Offshore
2010	73.9	71.2	42.3	28.9
2020	82.1	70.0	42.3	27.8
2030	93.1	62.5	42.3	20.2

Relentless production of onshore and shallow water fields has convinced the operators to focus on deep water areas where a combination of floating and subsea production units is used to extract hydrocarbons. Higher oil prices, especially after, the year 2000, have made this investment more economical. This trend was further reinforced in recent years due to lower political risks in deeper water than the locations with unstable business conditions in shallow water.

The ultra-deep water and some deep water fields (~5,000-10,000 ft depth) have had the above mentioned advantages, however when the well, is 10,000-15,000 ft depth, even if the reservoir has a normal pressure (1500 psi), it might not be able to move the oil to the host facilities.

Subsea pumping technologies are being preferred more and more in subsea tiebacks, subsea boosting and subsea processing. Subsea trees have lower potential recovery rates and are more expensive than dry trees. Therefore, any increase in the efficiency obtained from processing the products on the seabed instead of the platform or from reducing flow assurance issues can increase the recovery rates and profits, (Ioanna Karra, 2013).

As is expected in the start-up of a project, the oil has its own natural pressure, which provides enough boosting pressure to move from the reservoir through the well bore up to the surface. Over time, however, the natural reservoir pressure drops, which results in wells that are not able to flow, leaving a great amount of hydrocarbon still in place. In this situation, some form of artificial lift is required to drive the fluid to the surface and gain the maximum recovery of oil. Subsea boosting is an effective and economical tool to lift a large volume of fluids from great depths under different well conditions.

The main application of subsea pumps is in mature assets which have lost their own natural pressure, or in wells located far from host facilities (e.g. in ultra-deep water), when the reservoir pressure is not high enough to boost the oil flow. As shown in Figure

1, the other application is in the conventional separation systems used to transfer single phase liquids to topside facilities, or to inject water into the basin.

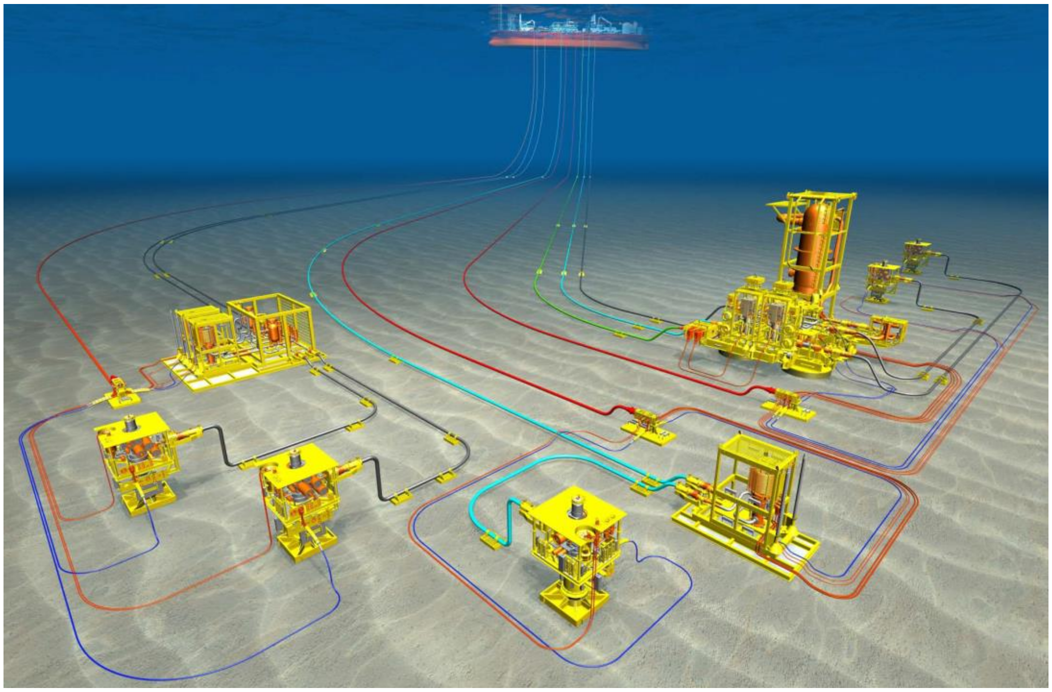

Figure 1: Subsea Processing Systems (Technip, 2011)

There are also several green fields that don't have enough boosting pressure to transfer oil to platform or shore facilities, or that are located far from host facilities. It means that they need a type of artificial lift such as subsea boosting and pumping from the beginning of the project.

All of these previously mentioned cases are reasons that have caused the development of subsea boosting and pumping. It is a new and growing field all over the world.

There are several different types of pumps for onshore applications. However, just a few of them are applicable in subsea services such as Helico-Axial Pumps (HAP), Twin

Screw Pumps (TSP) and Electric Submersible Pumps (ESP). The main reasons which limit subsea pumps to a few types are mentioned below:

1- Naturally onshore and topside pumps are designed to work in a limited working region with specific differential pressure, flow rate and Specific Gravity (SG). In offshore application it is quite normal to have wide ranges of production, Gas Volume Fraction (GVF), Water Cut (WC) and consequently Specific Gravity (SG) and Density.
2- Pumps in onshore and topside locations are accessible and a normal maintenance program can be followed for them, while a subsea pump is installed to work non-stop for 3-5 years. This high level of reliability and availability eliminates several types of topside pumps.
3- Subsea pumps are dealing with almost all types of corrosion from inside and outside. It is because of the high corrosion rate of sea water and production fluids, acids, etc. Although the pump casing is isolated from sea water, it is designed to withstand sea water in case of failure of insulation.

Where several standards and technical specifications have been prepared for onshore and topside pumps, no standard for subsea applications is available.

1.2 Research Objectives

This research focuses on different types of subsea pumps, and their advantages and disadvantages. It then lists all applicable parameters, such as differential pressure, GVF, viscosity, flow rate, well life, Opex and Capex, in the sizing and selection of a subsea pump. Finally it introduces a procedure to size the pump considering all parameters and based on the priority of different factors. In this research, corrosion and material selection as important concerns are also studied.

CHAPTER 2
MULTIPHASE BOOSTING

2.1 Introduction and Background

Multiphase boosting has been an alternative for subsea separation for a long time. Both methods have advantages and disadvantages which will be discussed further. Since each concept can result in different types of pump, both subsea separation and multiphase boosting have been studied in parallel.

No official document such as API (American Petroleum Institute), ASME (American Society of Mechanical Engineers) or ANSI (American National Standard Institute) standard has been published for subsea pumps. This survey was started with the review of available references such as API 610, API 682, API 676, NACE MR 0175, Pump handbook (Brennan, 2008) and Sulzer handbook (Sulzer, 2003), for centrifugal and screw pumps. Then based on the other available material such as records of different projects, the procedure has been developed. Some of these records are explained here. Dustan Gilyard, and Earl B Brookbank (Dustan Gilyard, 2010) describe the challenges and uncertainties in the Perdido project which resulted in big and complicated test facilities for newly designed ESPs to evaluate different scenarios and select the best and most reliable one. Fantoft, Hemndriks, and J. Elde from Tordis project (Fantoft, 2006) from Tordis project, also talk about important challenges such as sand handling. Ben D. Gould and Marcelo Clemente (Gould, 2011), (Clemente Marcelo, 2009) describe the down-hole application of Progressive Cavity Pump (PCP) and ESP pumps. Also there are papers from main manufacturers including Fromo and GE that explain the technical abilities of their products.

Schlumberger cooperating with Texas A&M and Clausthal universities (Hua, 2012) categorized the main types of subsea and down hole pumps such as TWP, PCP and HAP and mentioned some of the effective parameters.

2.2 Multiphase Boosting Advantages

Gas lift and water flooding are two solutions to drive more hydrocarbons. Each method is directly related to asset characteristics, cost and available technologies. and sometimes a combination of two methods is used in the field. Each approach has its own advantages and disadvantages. Gas lift increases the GVF and requires an expensive compressor package topside. More important is that gas lift is not available in the start-up of a project. Water flooding increases the water cut of the reservoir and separation requires installing heavy equipment.

Different wells even in one reservoir might have different GVF, water content and sand production which normally varies during the asset life. It is pretty common to observe an increase in the water cut and GVF during the asset life as well as a decrease in reservoir pressure. Separation facilities used to be installed topside, but new technologies have enabled companies to install separation facilities on the seabed. When they are installed subsea, the pumps are intended to handle clean fluids, however in recent projects (Dustan Gilyard, 2010), to achieve the highest level of reliability, it is important to design pumps for sand handling scenarios as a part of the qualification process. This means that even in a subsea separation unit, the pumps are relatively multiphase and need to be prepared for sand handling.

In some other projects, it would be an option to route sand to the topside instead of re-injecting to the well.

Providing electricity to the pumping systems is more reasonable and efficient than delivering gas to gas-lift systems, which require an expensive compressor and auxiliary equipment. The high-volume capacity, wide operating range, and efficiencies up to 40% higher than the gas-lift process make pumping systems more attractive for some deep water subsea wells.

All in all, the advantages of subsea multiphase boosting can be listed as the following (Dustan Gilyard, 2010) (Heyl, 2008):

1- Accelerates and increases the production
2- Stabilizes flow in wells that cannot naturally produce to remote facilities
3- Extends subsea tieback distance
4- Reduces well intervention cost
5- Reduces subsea development cost
6- Operates in an environmentally friendly way
7- Permits oil and gas production in a harsh environment
8- Eliminates offshore flaring and saves the relevant costs
9- Reduces backpressure at wellhead
10- Reduces the shear and decreases emulsion formation, which leads to lower flow assurance issues
11- Reduces the size of topside facilities

12- Does not require semi-submersible rigs and can be deployed simply with common vessels. Reduces both the overall cost of installation and intervention and deferred production resulting from a waiting period for a rig.
13- Provides a backup system to maximize run life and minimize deferred production, when configured properly.
14- Is not as space constrained as in-well pumps.
15- System alternatives use existing infrastructure to house the systems, which also significantly reduces overall development costs.
16- Production from several wells can be boosted with only one seabed booster system, (Y. Bai, 2010).

As per the previously mentioned advantages of subsea multiphase boosting, it is one of the best methods to increase the production rate. In the next chapter available types of multiphase pumps will be discussed.

Figure 2 shows how a green field or a brown one can take the advantages of the boosting. It should be mentioned that boosting system requirements must be studied and predicated in FEED and verified in the detail stage of a green field project, because providing required facilities such as power and control connections as well as required piping on a brown field will result in a huge intervention cost.

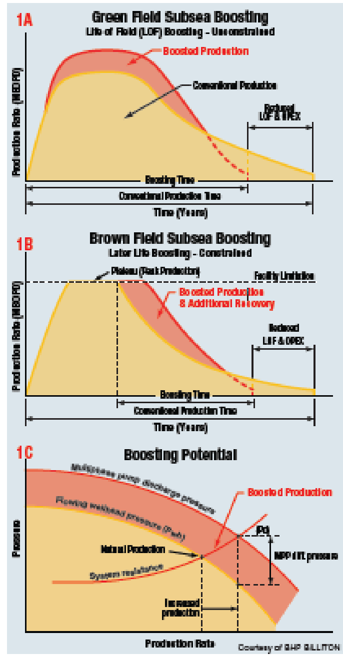

Figure 2: Advantages of Subsea Boosting in Green or Brown Fields and Boosting Potential (Subsea Processing Poster, 2013)

CHAPTER 3
SUBSEA PUMP TYPES

3.1 Introduction

In order to have an idea about the available types, the main subsea pumps are introduced in this chapter. Based on the subsea system design requirements such as GVF, water, viscosity, differential pressure and sand content, different pumps are used. In the special cases when the well produces oil with low GVF (less than 10%), simple one-phase pumps can transfer oil to the host facilities, but it is a rare condition. In contrast, in almost all cases subsea pumps are designed to handle a wide range of differential pressure, flow rate, GVF and water cut.

Two main categories of subsea pumps are shown in Figure 3: The Positive Displacement Pump and the Rotodynamic Pump. The Rotodynamic category consists of ESP and HAP; where TSP is a positive displacement pumps. Subsea pumps can also be categorized in two main applications, “subsea” and “down hole”, which have different requirements.

In the current market, ESP application is mainly for sea bed applications, where the ESP is mounted inside a caisson and installed on the seabed. In this case, ESP is a choice for low GVF. Newly designed ESPs with modified impellers can handle higher GVF.

Based on the above discussion, each pump is introduced hereafter.

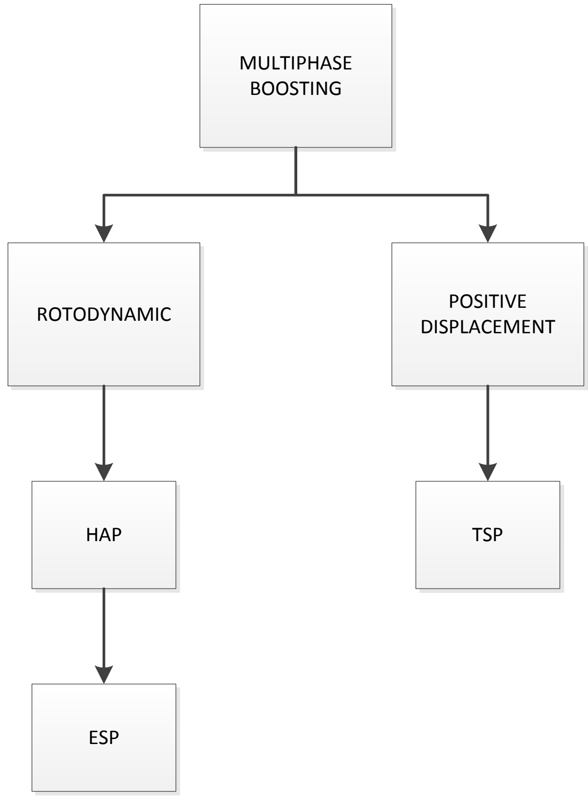

Figure 3: Applicable Multiphase Pump Types

3.2 Helico-Axial Pumps

An engineered centrifugal pump for high capacity and relatively low differential pressure should have high Specific Speed (nq) (7,000-20,000) (Engineering Toolbox, 2013) and utilize axial impellers. A simple description for HAP is a highly specialized rotodynamic pump with an open impeller installed on a single shaft requiring two mechanical seals. As shown in Figure 4 the design includes a compression cell that contributes to compress gases and mix with fluid to handle higher GVF rates.

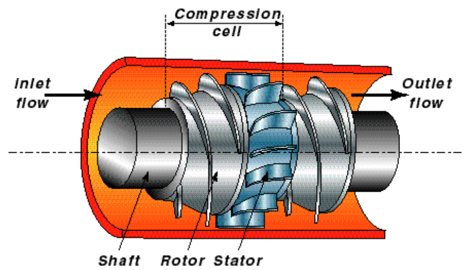

Figure 4: Helico-Axial Pump (Framo, 2013)

The other name of this type is " Poseidon pump", as the name of the first project that used these pumps (Arnaudeau, 1988, Statoil). HAP covers the differential pressure less than 2900 psig and total flow rate (oil, water and gas) at suction conditions ranging from 50,000 bpd to 450,000 bpd) with the RPM of 3500 to 6,500. GVF up to 70-80 % can be handled by this type. Figure 5 represents the effect of GVF on the operating ranges of a HAP.

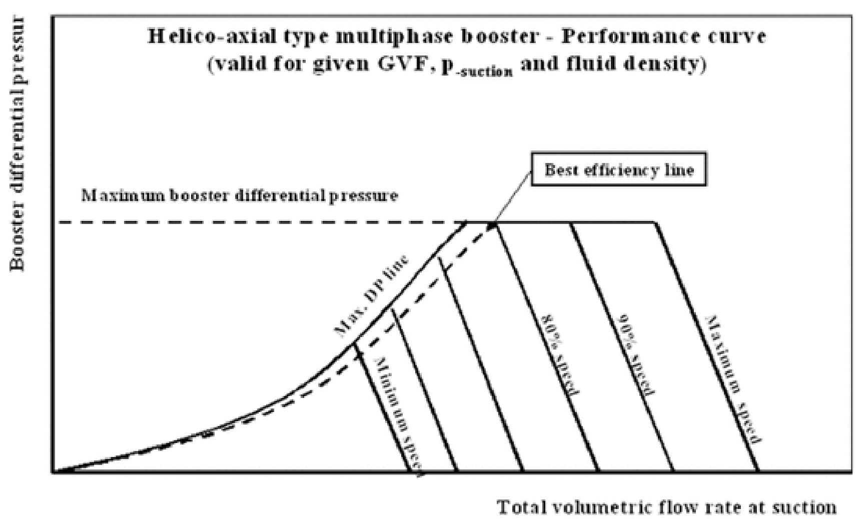

Figure 5: Operating Ranges of a Sample HAP for Different GVF (Schlumberger, 2013)

The main components of a HAP package are shown in Figure 6 and explained below:

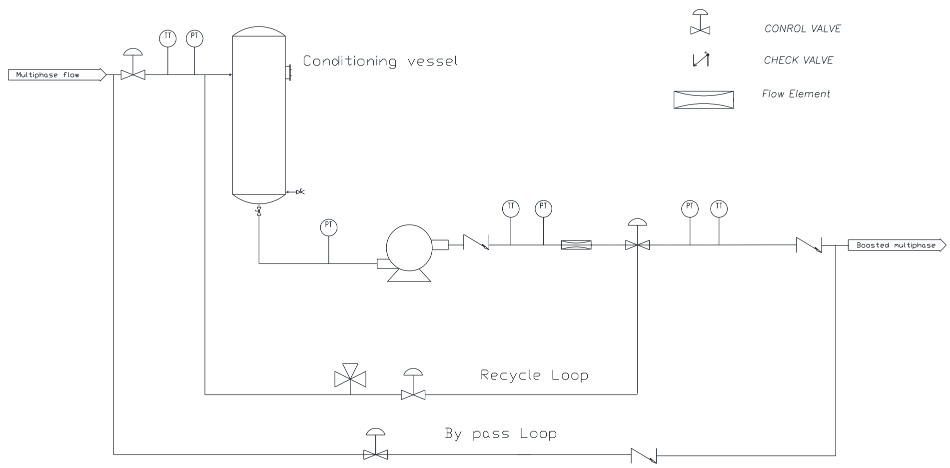

Figure 6: Typical Process Flow Diagram (PFD) For HAP

<u>Recycle line:</u> Utilizing a flow control valve, flow meter and pressure differential indicator (PDI), it controls the minimum flow across the pump. The recycle line protects the pump from running without fluid and consequently overheating. The other application of this system is to adjust the pump output in the wide range of operating conditions.

<u>Bypass flow line</u>: For any special conditions such as:

- Pigging
- operating under well pressure

<u>Conditioning vessel:</u> This component works as a mixer to dampen the slugs and provide a more uniform flow at the pump inlet. The recycle line normally is returned at the point before the conditioning tank. Therefore better control on GVF can be achieved.

<u>Recycle Vessel:</u> This tank is used to work as a separator to ensure that only liquid recycles in the pump and to help to improve ability to handle GVF. In advanced models, the recycle vessels have been eliminated.

By optimizing the design of conditioning and recycle tanks, manufacturers claim to have the ability to handle very high GVF (90%), however it hasn't been approved yet in the field. In fact, the conditioning tank and recycle line prevent the pump from seeing 100% GVF. Even in a very high GVF, minimum amount of liquid is recirculated in the pump. It is required to provide a liquid seal in TSP, and to remove heat in HAP. Minimum flow is also an important factor that reduces the vibrations.

<u>Check valve:</u> To prevent fluid from coming back from the riser and forcing the pump to rotate in the reverse direction, a check valve (one direction valve) is installed as downstream the package. Pump also has a built-in check valve.

<u>Isolating valves:</u> Isolate the pump package from the flow lines to be ready for intervention.

The advantage and disadvantages of HAP pump are listed in Table 2:

Table 2: Helico-Axial Pump, Pros and Cons

Helico-Axial pumps	
Advantages	**Disadvantages**
High capacity (50,000 bbl/d to 450,000 bbl /d) and	High shear
Good operational range and flexibility due to high speed range	Slugging problem
High differential pressure (< 2900 psig) and high pressure rise	Not a good choice for viscous fluids
Self-adapting to flow changes	Not a good choice for low flow rates
Series and parallel operation is possible	Not able to operate in low suction pressure
70 mm <Impeller diameter< 400 mm	
Low potential for erosion in case of solid handling	

3.3 Twin Screw Pumps

Single, Twin or Triple screw pumps have been used since the beginning of the oil and gas industry. As shown in Figure 7 and 8**,** single end arrangement provides higher pressures, while double end design reduces the axial loads on the bearings. A better way to balance the axial loads is utilizing a balancing piston in a double volute hydraulically balanced pump. Timing gears are used to transmit the power from the motor to the driven shaft and synchronize rotations to keep the standard tolerance between the screws and prevent erosion. Internal gears are more applicable to clean fluids that have lubricating properties. External gear design is more common and reliable in oil and gas heavy duty applications. A separate lubrication system is designed for bearings and gears in these applications. Among all types, only twin screw, double end, external gear arrangement pumps are used in subsea with the require adjustments for this environment.

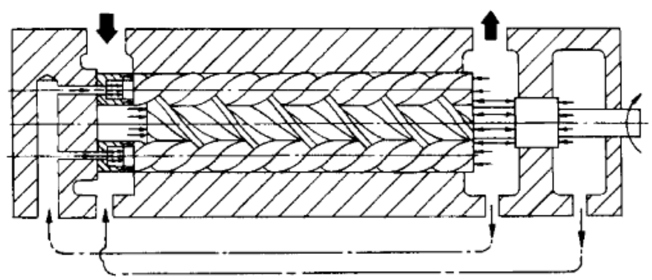

Figure 7: Balancing Line in a Single End Screw Pump (Karassik, 2008)

Figure 8: Twin Screw Pump with Double End (Bornemann, 2013)

Two screws force the movement of the pumped fluid. The screw pump doesn't create pressure; it just transfers a quantity from inlet to outlet. The standard space between the screws is called the cavity and is the unit for the flow rate. The sealing between the screws is provided by a liquid that is mandatory for proper operation of a TSP. A liquid management system is necessary to provide minimum liquid for circulation in any operating condition. In high differential pressure and low viscosity applications, some amount of liquid is allowed to return from discharge to suction, which is called slip (S).

Displaced volume (V_D) is a theoretical volume displaced per revolution of the rotor and is dependent on physical characteristics of the design. TSP has a theoretical flow rate (Q_t) and actual delivered flow (Q), which are defined below (Karassik, 2008):

$$Q_t = \frac{V_D n}{231} \quad \text{(US units)} \tag{3.1}$$

$$Q_t = 6 \times 10^{-8} V_D n \ \text{(SI units)} \tag{3.2}$$

$$Q_t = Q + S \tag{3.3}$$

These pumps can handle high viscos products with low shear that is not possible for HAP. The Speed is directly derived by suction lift and viscosity. The pump doesn't pull the fluid inside; suction pressure is necessary to push that inside. However in a subsea application, which is located close to the wellhead, it would not be a problem. The suction lift shouldn't be underestimated, because it may result in the pump operating at lower speeds. It increases the cost because of a larger low speed pump and driver.

The higher the GVF, the smaller the screw to provide a sufficient number of locks (space). Figure 9 shows a subsea multiphase TSP pump. A typical performance curve of a TSP is also shown in Figure 10.

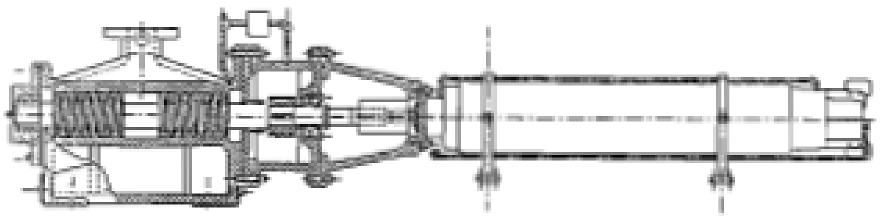

Figure 9: Subsea Twin Screw Pump (Cooper)

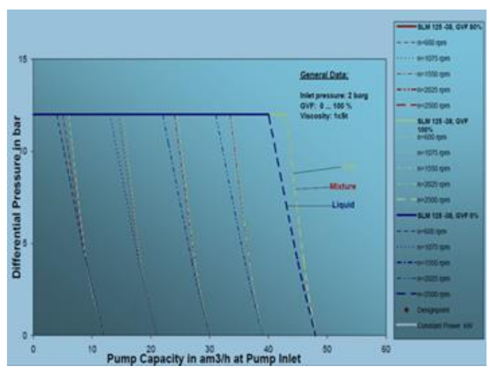

Figure 10: Twin Screw Pump, Typical Performance Curve (Hua, 2012)

The maximum differential pressure in different capacities is almost constant. The efficiency of the pump is the product of volumetric (η_v), mechanical (η_m) and hydraulic (η_{hyd}) efficiencies.

$$\eta_p = \eta_{hyd}\eta_m\eta_v \tag{3.4}$$

To calculate the efficiency for a screw pump, the most important factor is the volumetric efficiency. The two other efficiencies are high enough and close to 1. Mechanical efficiency is lower at low viscosity and high speed. The hydraulic efficiency is lower at high viscosity and low speed operation. Generally the losses are governed by changes in viscosity and speed, not by the pressure. The worst case is the highest viscosity case.

The volumetric efficiency for low flow rates can be estimated using Figure 11. The formula gives a better estimate for higher capacities.

$$\eta_v = (Q_t - S)/Q_t \tag{3.5}$$

Where, Q_t, S and Q are theoretical capacity, slip and capacity respectively.

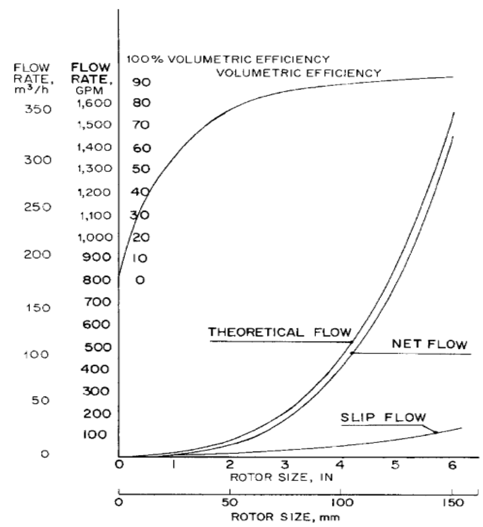

Figure 11: Volumetric Efficiency (Karassik, 2008)

The figure is not applicable to subsea application in that the capacity is in a higher range, it can be useful just as a guide. It should be noted that Q_t and S are proportional to ID with the power of three and two respectively.

Referring to the above mentioned relation between the diameter of the rotor, Q_t and S, when the volumetric efficiency is low, increasing the rotor size will increase that successfully.

When GVF increases, the slip decreases until the inlet volume flow rate is equal to the pump discharge (slip=0). This situation causes a constant inlet volume flow rate, independent to the DP. It means even a small amount of liquid can provide a liquid seal. Figure 12 shows the performance characteristics of a multiphase pump in different GVF.

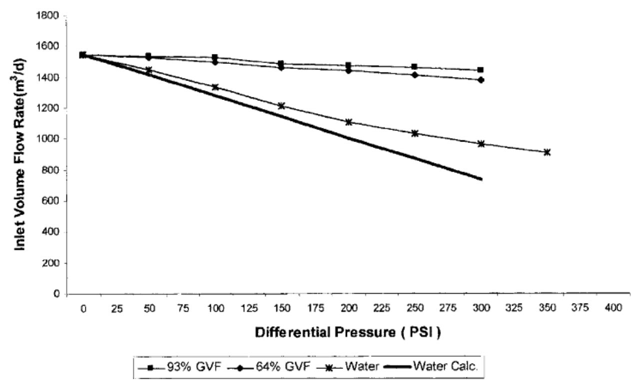

Figure 12: Typical Performance Characteristics of a Screw Pump in Different GVF (Brennan, 2008)

The pump is not susceptible to fluid density; therefore it can handle slugs much better than HAP and in most cases doesn't need a conditioning tank in the inlet side. Other components are shown in Figure 13 and explained hereafter.

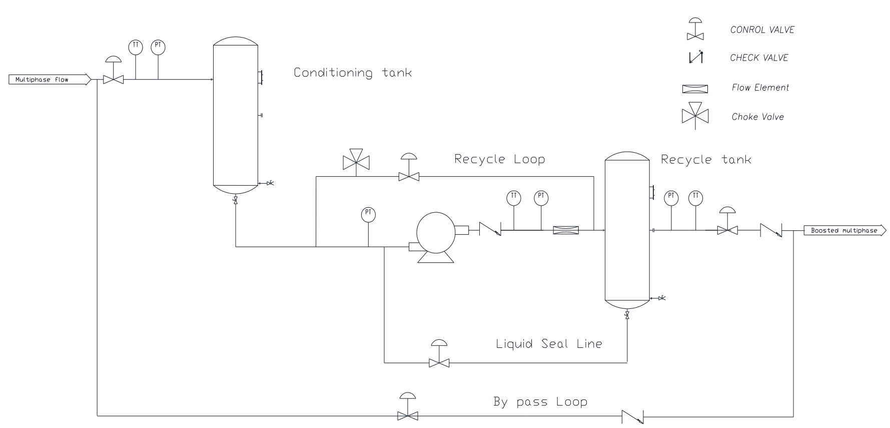

Figure 13: Typical Process Flow Diagram (PFD) For TSP

The applications of check valves, isolating valves and by pass lines are the same as that for HAP. The main difference is the necessity of a Liquid Seal line that was explained before. Advantages and disadvantages of TSP are listed in Table 3.

Table 3: Twin Screw Pump, Pros and Cons

Twin screw pumps	
Advantages	**Disadvantages**
Handles higher GVF	Series operation is not possible
Good choice for viscous fluids	Resize over life of asset might be needed
Normally no need for buffer tank	Low to moderate capacity (10,000 to 440,000 bbl /d), due to low speed 600 to 1800rpm (barely 3600 rpm)
	Slipping increase with higher discharge line resistance

TSP is designed for vertical and horizontal installation, while HAP is only available in vertical design. Figure 14 shows the vertical types of HAP and TSP with more details.

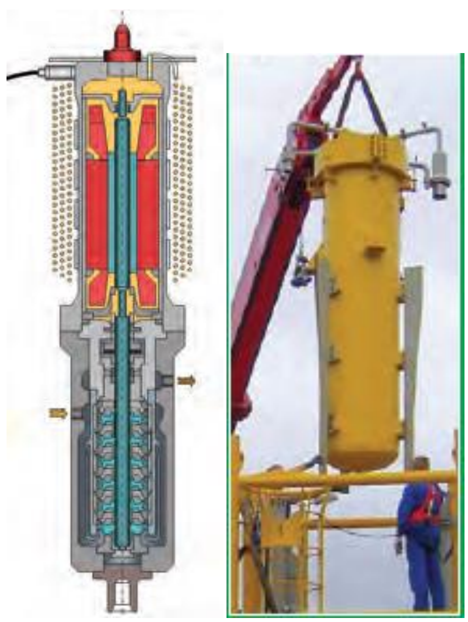

Figure 14: HAP and TSP in Vertical Design (Subsea Processing Poster, 2013)

3.4 Electrical Submersible Pumps

The ESP pump modular consists of a driver unit and a pumping unit. The driver is installed upstream of the pump to be cooled by passing liquid. The driver can be either an electric motor or a water turbine. Figure 15 represents an ESP tuned for gassy wells. The pump itself is not designed to handle high GVF, because the electric motor is cooled by the passing liquid. ESP is an option at a flow rate range from less than 1,000 BFPD to 20,000 BFP and in the future of down-hole applications; for example in ultra-deep wells (10,000 ft) to drive the fluid to the seabed. After boosting fluid to the seabed, HAP or TSP can drive it to the host facilities. ESP in series installation can provide enough differential pressure to boost fluids to host facilities. Table 4 lists the advantages and disadvantages of ESP.

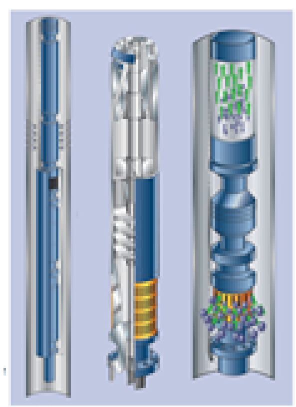

Figure 15: Advanced ESP Design (E&P, 2013)

Table 4: Electrical Submersible Pump, Pros and Cons

Electrical submersible pumps	
Advantages	**Disadvantages**
Series operation is possible	Low sand production capability
Good choice for low viscosity	Not suitable for viscous fluids
High differential pressure in series operation	High shear , potential emulsions

3.4.1 Multi Vane Centrifugal Pumps

For GVF around 20 percent, a mixer is installed upstream of ESP or the shape of the impeller is changed to multi vane type (Hua, 2012). The advanced design, split vane impellers with enlarged balance holes as shown in Figure 16, make ESP suitable for gassy wells and increase the GVF capability of ESP up to 40%.

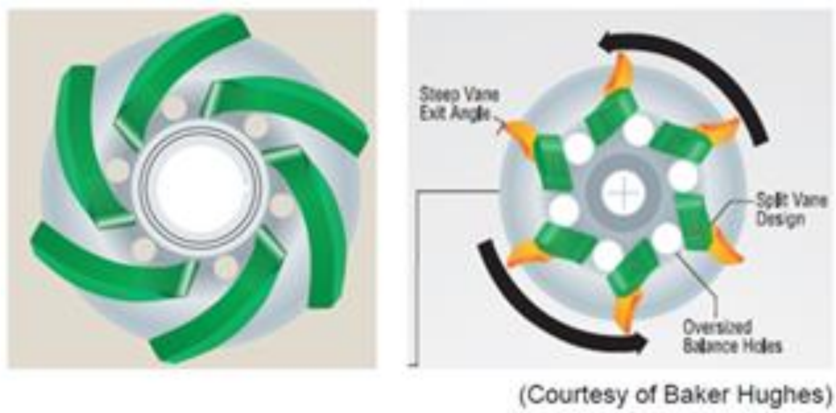

Figure 16: Split Vane Impeller (Hua, 2012)

3.5 Centrifugal Pumps

When the gas GVF is relatively low (<10%), for example downstream of the separation system or in water injection applications, it is possible to handle that with centrifugal pumps. These pumps are normal centrifugal pumps which have been modified for subsea applications. They are used for water injection or hydrocarbon transfer from topside or subsea locations to the host facilities. The pump as shown in Figure 17 is a common centrifugal pump that is susceptible to gas handling. Gas bubbles collect in the low pressure side of the impeller and cause gas locking. As a consequence the flow rate will be stopped.

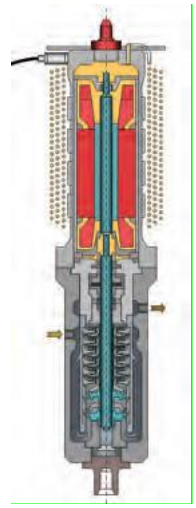

Figure 17: Electrical Centrifugal Pump (Subsea Processing Poster, 2013) .

CHAPTER 4
EFFECTIVE SELCECTION PARAMETERS

4.1 Introduction

In order to conduct correct design and pump selection, it is important to have a clear picture of all affecting parameters. The main factors such as differential pressure (or Head), flow rate and viscosity are working the same as those on topside pumps; however the effects of multiphase flow and viscosity correction should be considered in calculations. The other difference is that these parameters are fluctuating during both short term and long term production. Inherently the reservoir pressure drops during the asset life and the pump is needed to provide more boosting pressure. Lower reservoir pressure causes lower flow rate. In case of any slug, a rapid change in characteristics is also expected. Slug can increase the torque on the pump shaft and the motor.

The important parameters for subsea and down-hole pump selection are as listed below. The corrosion, which is the most influencing factor in material selection, will be discussed at the end as a separate section.

4.2 Differential Pressure

Similar to any other pump application, the equipment is intended to drive the fluid. The pump must overcome all friction and gravity losses as well as internal mechanical, volumetric and hydraulic losses.

4.2.1 Pressure Loss Calculation in Single Phase Flow

A pressure loss is the result of friction and gravity losses. Differential pressure (or head) due to friction losses in single phase flow can be calculated from this equation, (Bratland, 2010):

$$\Delta P = \frac{f.l.\rho.V^2}{2.ID} \tag{4.1}$$

Where, f , l, ρ, V, ID are surface roughness coefficient, line length, density, fluid velocity and internal diameter respectively.

The surface roughness, density, pipe length and velocity increase the pressure loss, while larger ID decreases the losses. Figure 18 shows the effect of surface roughness on the single phase flow.

Higher surface roughness such as in flexible pipes increases the friction loss and requires higher power. Sometimes in ultra-deep water it is inevitable to use flexible composite risers. Also, increasing the ID of the riser may be an option to reduce the friction loss, but it must be considered that a larger riser increases the top tension of the riser and rig capacity. Another consequence of a large riser is lower flowing speed which may result in flow assurance issues such as wax and hydrate formation.

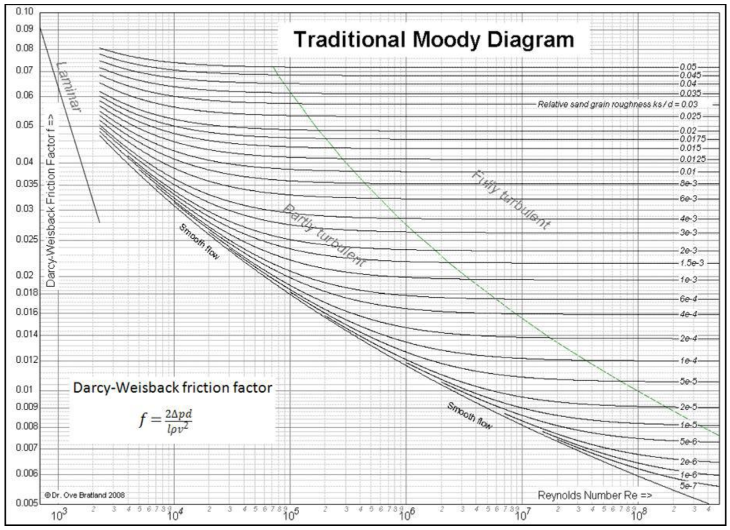

Figure 18: Moody Diagram (Bratland, 2010)

Gravity loss in single phase flow also can be calculated using this formula:

$$\Delta P_{grav.} = \rho.g.h \tag{4.2}$$

Where, h and ρ are true depth and density respectively. g is equal to 9.81 m/s^2 in SI units.

There are also some important local pressures that play significant roles in the calculations:

Reservoir pressure (Pres.): Estimated pressure of the reservoir

Bottom Hole Flowing Pressure (BHFP): Pressure exactly above the completion

The relation between bottom hole pressure and reservoir pressure is explained here (Bratland, 2010):

$$PI = \frac{Q_{Tube}}{\mathrm{P}_{res.} - BHFP} \tag{4.3}$$

Where PI, Q_{tube}, $P_{res.}$ and BHFP represent productivity index, tubing Flow rate, reservoir pressure and Bottom Hole Flowing Pressure respectively.

The Productivity index of a reservoir is the ability to produce and is a valuable measure to predict the future performance of the well. It is a tool to estimate the profile of the reservoir pressure, well head pressure and the time that boosting would be required.

The third one is topside pressure, or the destination pressure.

Top side pressure: The fluid received at the topside or host facilities should be in the certain range that downstream equipment is designed for. This pressure and temperature of this condition is around 100-500 psi and 120-140 °F respectively.

Wellhead Pressure: Pressure measured at the subsea well head. Sometimes it is given and sometimes it must be calculated based on the productivity index and BHFP. In this case, frictional losses and gravity losses in the tubing must be considered too.

Having the frictional, gravitational losses and well head pressure, it is possible to calculate the pump differential pressure. Other losses such as that in valves, flow meter, and bends can be added to the formula.

$$\Delta P_{pump} = P_{topside} + \Delta P_{friction} + \Delta P_{gravity} + \Delta P_{losses} - P_{wellhead} \tag{4.4}$$

It is also possible to estimate the well production profile (flow rate and pressure) utilizing Pipesim, OLGA or other software. All these estimations must be done at the FEED stage of the project and finalized in the Detail stage. Failing to study the required facilities, such as topside power and control facilities or umbilical connectors in the FEED stage, will result in a significant cost increase in the future.

As stated in chapter 3, HAP can handle the differential pressure to 2900 psi and TSP to 1500 psi. Having the required pressure is one of the key factors to select the pump.

4.2.2 Pressure Loss Calculation in Multiphase Flow

Using software such as OLGA, Pipesim or applying analytical methods, the type of the flow regime and required pressure to boost fluid from the wellhead to the host facilities can be calculated. The size of the flow line, gas and liquid flow rates are given from flow assurance study. In the worst case, in the FEED stage of a green field project, the line size can be modified based on the boosting requirements.

To do the analytical solution, the Beggs and Brill correlation can be used according to the following steps (Saleh, Fluid Flow Handbook, 2002):

$If\ \ N_{FR} < L_1 : Segregated\ \ flow$

$If\ \ L_1 < N_{FR} \le L_2 : Intermittent\ \ flow$

$If\ N_{FR} > L_2\ and\ N_{FR} \ge L_1 : Distributed\ \ flow$

$$L_1 = \exp(-4.62 - 3.757X - 0.481X^2 - 0.0207X^3) \tag{4.5}$$

$$L_2 = \exp(1.061 - 4.602X - 1.609X^2 - 0.179X^3 + 0.635X^5) \tag{4.6}$$

N_{FR} is dimensionless number and is called Froude number as:

$$N_{FR} = \frac{V_m^2}{gD} \quad (Froude\ Number) \tag{4.7}$$

$$X = \ln(\lambda) \tag{4.8}$$

$$\lambda = \frac{Q_L}{Q_L + Q_G} \tag{4.9}$$

Where λ and Q are "Input liquid content" and "Volumetric flow rate" respectively. Q_L and Q_G are "Volumetric flow rate" for liquid and gas respectively. D is pipe inside diameter and V_m is mixture velocity.

To calculate gravitational friction factor, liquid hold up (H_L) must be computed:

$$H_L(\theta) = H_L(0)\psi \tag{4.10}$$

Where $H_L(0)$ is "liquid holdup fraction" at horizontal condition (angle=0 degree) and $H_L(\theta)$ is "liquid holdup fraction" at angle θ. ψ also is the inclination factor. Liquid holdup in horizontal fraction for different flow regimes is defined here:

$$H_L(0) = \frac{0.98\lambda^{0.4846}}{N_{FR}^{0.0868}} \; for\ segregated\ flow \tag{4.11}$$

$$H_L(0) = \frac{0.845\lambda^{0.5351}}{N_{FR}^{0.0173}} \; for\ \mathrm{int}ermittent\ flow \tag{4.12}$$

$$H_L(0) = \frac{1.065\lambda^{0.5824}}{N_{FR}^{0.0609}} \; for\ distributed\ flow \tag{4.13}$$

and

$$\psi = 1 + C[\sin(1.8\theta) - 0.333\sin^3(1.8\theta)] \tag{4.14}$$

θ is angle of inclination and c is defined based on the Table 5.

Table 5: C Coefficient for Angle Correction Factor (Karassik, 2008)

Flow Pattern (Horizontal)	Upward Inclination	Downward Inclination
Segregated	$C=(1-\lambda)\ln\left[\frac{0.011N_{LV}^{3.539}}{\lambda^{3.768}N_{FR}^{1.614}}\right]$	$C=(1-\lambda)\ln\left[\frac{4.7\,N_{LV}^{0.1244}}{\lambda^{0.3692}N_{FR}^{0.5056}}\right]$
Intermittent	$C=(1-\lambda)\ln\left[\frac{2.96\lambda^{0.305}N_{FR}^{0.0798}}{N_{LV}^{0.4473614}}\right]$	$C=(1-\lambda)\ln\left[\frac{4.7\,N_{LV}^{0.1244}}{\lambda^{0.3692}N_{FR}^{0.5056}}\right]$
Distributed	0.0	$C=(1-\lambda)\ln\left[\frac{4.7\,N_{LV}^{0.1244}}{\lambda^{0.3692}N_{FR}^{0.5056}}\right]$

The gravitational pressure losses per unit length is defined here:

$$(\frac{dP}{dZ})_g = \frac{\rho_m.g.Sin\theta}{144g_c} \tag{4.15}$$

Where

$$\rho_m = H_L\rho_L + (1.0 - H_L)\rho_G \tag{4.16}$$

and ρ_m is mixture density as a function of gas and liquid holdup.

The frictional pressure drop losses per unit length also formulated below:

$$-(\frac{dP}{dZ})_f = \frac{f_{tp}G_mV_m}{2g_cID} \tag{4.17}$$

f_{tp} , G_m , V_m and ID are “two phase friction factor”, “mixture mass flux”, ”mixture velocity” and internal diameter respectively. Using no slip friction factor, two phase friction factor, f_{tp} can be calculated:

$$\frac{f_{tp}}{f_n} = e^s \tag{4.18}$$

f_n, which is called “no slip friction factor” in a smooth pipe, can be calculated using the Moody diagram and Reynolds’ number:

Where S is defined as:

$$S = \ln(y) / \{-0.0523 + 3.182\ln(y) - 0.8725[\ln(y)]^2 + 0.01853[\ln(y)]^4\} \qquad (4.19)$$

and

$$y = \frac{\lambda}{[H_L(\theta)]^2} \qquad (4.20)$$

For the interval 1.0<y<1.2, S is unbounded and defined as:

$$S = \ln(2.2y - 1.2) \qquad (4.21)$$

Having the frictional, gravitational losses and well head pressure, it is possible to calculate the pump differential pressure. Other losses such as that in valves, flow meter and bends can be added to equation (4.4) to calculate pump differential pressure.

4.3 Flow Rate

The second main pump parameter is flow rate. Flow rate normally changes the size of the pump but not necessarily the design. Looking at the capacity ranges in Table 6 gives an idea about the capability range of each pump type:

Table 6: Ranges of Flow Rates for Different Pump Types

	Pump type	Barrel Per Day (bpd)
1	HAP	50,000-450,000
2	TSP	10,000-440,000
3	ESP	1,000-20,000 max

Using the affinity laws it is possible to check the effects of the impeller diameter or motor RPM on the capacity.

According to the affinity law, for one single pump, if the impeller diameter kept constant, then:

$$(Q_1 / Q_2) = (n_1 / n_2) \tag{4.22}$$

$$(h_1 / h_2) = (n_1 / n_2)^2 \tag{4.23}$$

$$(P_1 / P_2) = (n_1 / n_2)^3 \tag{4.24}$$

and also if the shaft speed held constant it can be observed that:

$$(Q_1 / Q_2) = (D_1 / D_2) \tag{4.25}$$

$$(h_1 / h_2) = (D_1 / D_2)^2 \tag{4.26}$$

$$(P_1 / P_2) = (D_1 / D_2)^3 \tag{4.27}$$

Where, Q, P and h are flow rate, power and impeller diameter respectively.

The ESP doesn’t have enough space to increase the diameter, and therefore cannot handle higher capacities, so rotation speed must be increased. This results in a larger size motor with the power of 3. It means ESP is not a good choice in this case.

For high capacities or for the wide range of capacities, HAP is the best option. HAP is able to change the rotation speed from 3600 to 6500 rpm. To compensate even wider ranges, Variable Speed Drive (VSD) motors can be used.

4.4 Gas Volume Fraction

GVF is defined as the ratio of gas flow rate over liquid plus gas flow rates. When gas bubbles enter a generic centrifugal pump, they accumulate in the low pressure side of the impeller and cause gas lock. The air bubbles fill the impeller and the pump cannot drive the fluid. An ordinary centrifugal pump is not intended to handle GVF. However, for subsea application where the pump is intended to handle the fluctuating condition of the well, GVF must be considered in design. One of the methods to handle higher, GVF as explained in Chapter 3, is to use multi vane impellers which are applicable to ESPs. Utilizing this modification the ESP pump is able to handle higher GVF up to 40-50 %.

HAP, which is an axial flow pump, uses the advantages of its design, which doesn't cause gas lock. TSP, which is independent to the density of inlet fluid, is the least susceptible pump to GVF. The manufacturers take some measures to fulfill this requirement. The applicable methods are listed here:

1- Installing mixer upstream of the ESP
2- Installing conditioning tank and recycle line on the HAP
3- Installing recycle tank and liquid seal line for TSP

Differential pressure, flow rate and GVF have been explained up to now. Figure 19 gives a rough estimation about the pump type based on flow rate and GVF. It is not possible to select the pump accurately without taking into consideration the viscosity. Viscosity is the parameter that can change the selection easily. It will be discussed in the upcoming section.

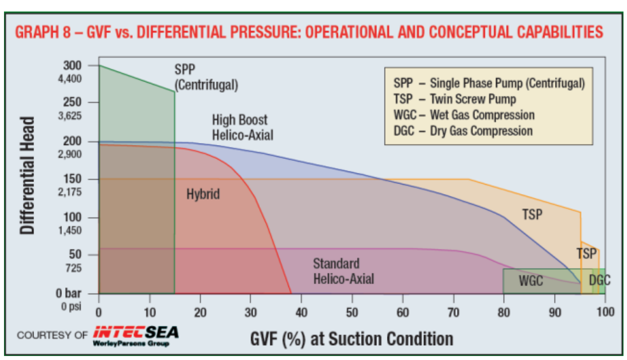

Figure 19: Concept Selection Based on GVF and Flow Rate (Subsea Processing Poster, 2013)

4.5 Viscosity

Viscosity is one of the main parameters in pump design and selection. Viscosity of the produced oil can vary from light oil (API>40) to extra heavy oil (API<23). Some pumps such as ESPs work much better with low viscosities, while some others such as twin screw pumps are better for heavy oils. TSP can be used for very high viscous fluids. For the viscosities more than 350 cp, no other pump type can handle the fluid. The efficiency of TSP increases with higher viscosities. It is possible to reduce the heavy oil viscosity by adding Kerosene, but it will be a very expensive solution, (Schobert).

4.6 Operating Temperature

High temperature operations (250- 350 °F), increase the risk of some elastomers failure and reduce the cooling characteristics of the fluid. In this case, the well fluid cannot be used to cool the mechanical seal and bearings, and barrier fluid is supplied from the topside and is independent to the well fluid. Low temperature application increases the risk of hydrate, wax and paraffin formation and other flow assurance issues. Very low temperatures have negative effects on materials and may cause brittle fracture. The effects of temperature on corrosion will be discussed in an upcoming section.

4.7 Low Suction Pressure

As a general rule, the wells have high pressures and suction pressure, therefore net positive suction head NPSH is not a problem. However in some cases especially for mature assets, this issue might be raised. In this case, HAP cannot tolerate low suction pressure. If the reservoir is going to this condition, TSP might be a better choice. Finally the ESP that can be installed down hole just above the fraction is the last solution if the well doesn't produce sand and the fluid is low viscous type.

4.8 Water Oil Ratio

Water oil ratio is the parameter that defines the water content. Water production can increase over time depending upon the reservoir drive mechanism, thus if water drives, more water will be delivered to the topside over time. Water flooding in order to reach higher productivity ratios increases the WOR and the density of the produced fluids. Higher density requires higher power. It means the pump sizing must be conducted for the highest possible density.

4.9 Sand Production

Similar to viscosity, sand is governing parameters. The size and shape of sands and amount of it must be studied. HAP can handle sands because of its axial flow design. TSP can handle limited sand when equipped with hardened screws. ESP is not intended to carry sand production.

4.10 Asset Life

Although in some cases it is possible to rerate the pumps, since it is very expensive to design, test, install, operate and intervene, it is preferred to design the system for the longest operating time, for example 8-10 years. Sizing a pump requires rigorous study of reservoir, fluid composition, production index and reduction rates.

4.11 OPEX and CAPEX

Capex for all subsea boosting projects are high. From the other side, only a few manufacturers such as GE, Sulzer, Flowserve, Baker Hughes, Bornemann, Framo, Schlumberger and Leistsritz are available in the market which reduces available choices. If possible, it would be a good idea to select a pump among the available products instead of designing and testing a new one. This is a cost saving decision because the operator or the engineering contactor can track the records of similar installed products in the other projects. There is also no need to do several tests to ensure reliability of the system. The manufacturer doesn’t need to invest millions of dollars to design a new model, and they can supply the equipment in a relatively short period. The other important advantage is the earlier production, which is the goal of every project and if the pumps are on the critical path, this can make a significant effect on the project costs.

Subsea architecture, number of wells and future plans should be mentioned when selecting the boosting system in the FEED stage of the project. Sometimes it is not possible to insert several ESP inside several wells. It should be better to collect wells’ output in the manifold located on the sea bed and install the pump over there.

The size and weight of the equipment and accessories governs the size and the capacity of the vessel required to install them. Lower weight and the sizes, (normally less than 80 tons) can lead to lower installation costs.

4.12 Reliability

The pumps and auxiliaries are intended to work non-stop for about 5 years. Selecting all modules and materials accurately increases reliability and would require less intervention and OPEX (Bass, 2006).

Operators prefer a pump to work for a period of 5 years without maintenance. To satisfy the highest reliability criterion, incorporating all operation and shutdown scenarios and reservoir characteristics must be considered. In some projects such as BP King in Gulf of Mexico, one pump is installed as a spare module to increase the reliability and guarantee the production in case of failure. The most common problems associated with the installed pumps according to available records are mentioned in the following:

4.12.1 Bearing Failure

Bearings are intended to tolerate high trust and radial loads. In case of slug flow, they have to be designed to handle huge fluctuating loads on the pump shaft and the motor shaft. Special attention should be paid to designing an appropriate conditioning vessel to mitigate slug issues and provide uniform flow.

4.12.2 Leakage

Leakage of production fluids to the environment or leakage of sea water into the equipment, piping, electrical and control module can result in environmental issues, equipment failure or even production shut-in. Limited leakage of barrier fluid from the sealing system to the process side is normal, but in the reverse direction, it may cause severe damage to bearings and the electric motor. Mechanical seals must be selected precisely and must be redundant to reduce the risk (Necker, 2002). Diamond coated seal face is a newly developed technology (Pennwell, 2013) that reduces the friction and the temperature between the contacting faces and increases the life of the seal. Because of the higher ductility of Sintering materials and Ceramics, they are more applicable than silicon carbide for subsea application. A condition monitoring system with several sensors to measure the pressure, temperature and flow rate including ones to detect any leakage is a real time facility to increase the reliability.

To prevent leakage from piping or vessels, all weld lines must be tested completely according to an approved WPS and PQR.

4.12.3 Motor, Control and Power Transmission Failure

Motor, VSD system and power transmission lines are critical facilities. Failure in each component can result in production failure. Proper starting and cooling systems for heavy oil applications are highly recommended. The redundancy in the control system is a solution to increase the reliability of the system. The VSD system should be compatible with the range of production in the expected life.

In order to increase the reliability, the subsea pump is tested on several levels. All components such as motor or mechanical seals are tested in the supplier's shop. The complete system tests such as mechanical running test, performance test, complete unit test or extended "Fabrication and Assembly Test" (FAT) are performed under the simulated condition in a simulation tank. For new products, comprehensive tests are conducted to define the "Mean Time Between Failure" (MTBF) records of a new subsea pump.

4.13 Corrosion and Erosion

Water is one of the common production fluids in which the presence of CO_2 or hydrogen ions can increase the acidity of the fluid. Also hydrogen sulfide in the presence of water shows its corrosive effects. Furthermore, the reservoirs contain corrosive fluid such as hydrofluoric acid and other packer fluids that cause corrosion and cracking. Sand production increases erosion corrosion. From the outside, the pump assembly is in contact with the sea water which contains chloride as a factor that prevents the passive layer from rebuilding. Sea water itself as an excellent electrolyte also causes Pitting and Crevice corrosion.

Although painting, coatings and insulation are methods to prevent the surface from contact with sea water, all components must be designed to tolerate this severe condition in case of insulation failure.

Oxygen concentration even less than 10 ppm can accelerate corrosion significantly. Sulfur (S) is also another element with corrosive effects that have been discovered in recent decades. The presence of dirt and no working cathodic protection system (sacrificial anodes), may cause microbiologically induced corrosion (MIC).

Corrosion is one the most important problems, especially in the sea water and subsea applications where there is no access to the equipment for a long time.

Basically corrosion occurs when three factors operate together:

1- Electrolyte medium such as water
2- Susceptible material such as iron

3- Corrosive agent such as hydrogen ion or oxygen

Other than corrosive agents such as CO_2, H_2S, and oxygen, several other factors that affect corrosion rates or cracking are explained hereafter.

4.13.1 pH

pH specifies the acidity or alkalinity of the medium. In down-hole applications, the combination of aqueous CO_2 and hydrogen sulfide with water can make an acid solution and reduce the pH. For example, Figure 20 represents the effect of low pH on Sulfide Stress Cracking. As shown on the graph, the effect of hydrogen sulfide is increased significantly when the pH reduces. Partial pressure (concentration) of hydrogen sulfide more than 0.05 psi is a critical limit to consider the service as a sour and paying more attention to material selection is necessary.

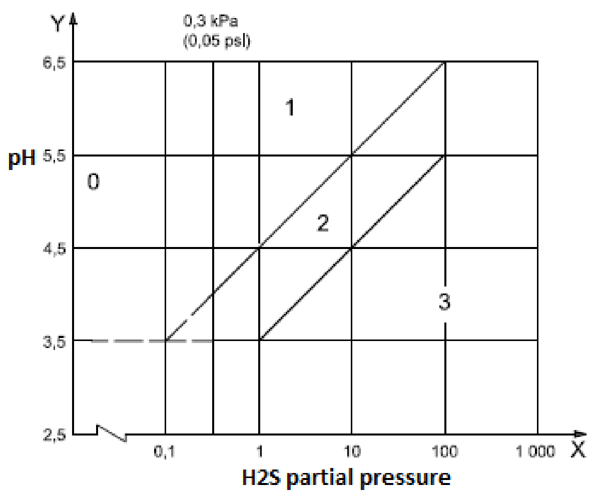

Figure 20: Effect of pH on SSC in Carbon and Low Alloy Steels (MR0175,Petroleum and Natural Gas Industries – Materials for use in H2S-containing Environments in Oil and Gas Priduction, 2003)

4.13.2 Temperature

The effect of temperature is a challenging discussion. As a general rule, increasing the temperature is detrimental to corrosion due to the increased rate of chemical reactions (Craig, 2006); however after some time the corrosion products cover the surface and reduce the rate. Some references explain the unlimited direct relation between the corrosion and temperature rise (Craig, 2006); however, there are exceptions in the CO_2 environment and corrosion rate decreases when the temperature passes 150-160 °F (70-80 °C) (Bajvani, 2011).

4.13.3 Stress

Stress is a supporting factor to corrosion. In general, stress corrosion cracking (SCC) happens if a susceptible material works under tensile stress and in a corrosive environment, (Jones, 1996). In the sour service, “Sulfide Stress Cracking (SSC)” happens by a similar mechanism. The only difference is the source of hydrogen ions, which is hydrogen sulfide.

Corrosion Fatigue Cracking (CFC) is the other type of corrosion that increases the rate of corrosion significantly. Fatigue life shows a dramatic decrease in a corrosive environment. Figure 21 represents the change in S-N curve, in corrosive medium.

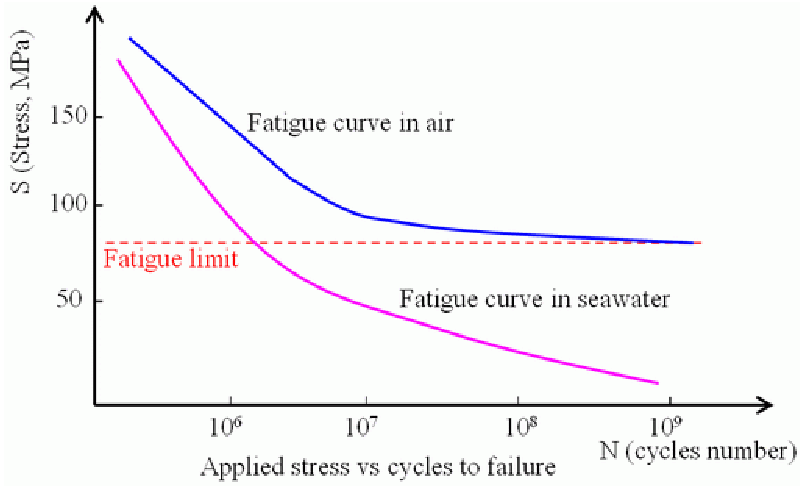

Figure 21: Reduction of Fatigue Life in Sea (Corrosion Consulting, 2013)

4.13.4 Velocity

From one point of view, high velocity doesn't let the material react and corrode, but as a general rule, increasing velocity in the presence of oxygen can increase the corrosion, especially when the fluid carries solid particles like sand. In addition, high velocity can break down the protective layer of rust and accelerate the corrosion. However, most graphs do not show the effect of low velocity on the corrosion rate. As shown in Figure 22, when the wetted surface is not clean, biofilms can form and in contact with CO_2 or hydrogen sulfide can increase the corrosion. In long shut-in conditions or in bypass lines where stagnation points are formed, localized corrosion such as pitting can happen. High velocity in high production rates, especially in the valves or orifices can be the reason for erosion corrosion.

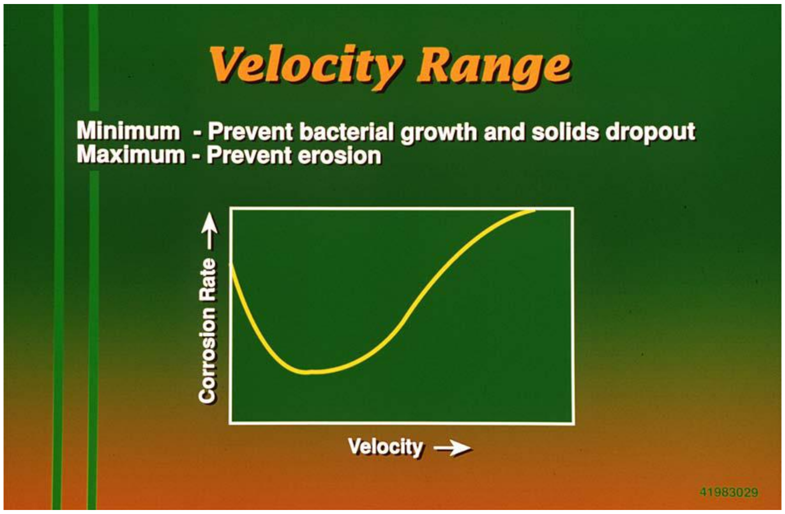

Figure 22: Effect of Velocity on Corrosion (MR0175,Petroleum and Natural Gas Industries – Materials for use in H2S-containing Environments in Oil and Gas Priduction, 2003)

The types of corrosion can be classified in nine categories (Fontana, 1985).

4.13.5 Uniform Corrosion

One type of corrosion is when a part is in contact with the solution uniformly and the metals have uniform microstructure. Uniform corrosion can be observed in the pump body, steel structures of the pump module and piping. To mitigate uniform corrosion, coating and painting are applied. For structures, cathodic protection is widely used in subsea applications.

Figure 23 represents a uniform corrosion in the presence of water that naturally contains oxygen (aerated water). At the active sites (anodes), electrons leave the iron atoms and move into the surrounding environment as ferrous ions. This process is represented as:

$$2Fe \rightarrow 2Fe^{2+} + 4e^{-} \quad (4.28)$$

The electrons that remain in the iron body flow to the cathodes and react with aqueous oxygen . A common reaction is:

$$2H_2O + O_2 + 4e^- \rightarrow 4OH^- \tag{4.29}$$

To maintain electrical neutrality, the ferrous ions move through the water to the cathodic sites where they react and form iron hydroxides (rust):

$$2Fe^{2+} + 4OH^- \rightarrow Fe(OH)_2 \tag{4.30}$$

This initial precipitated hydroxide tends to react further with oxygen to form higher valence oxides.

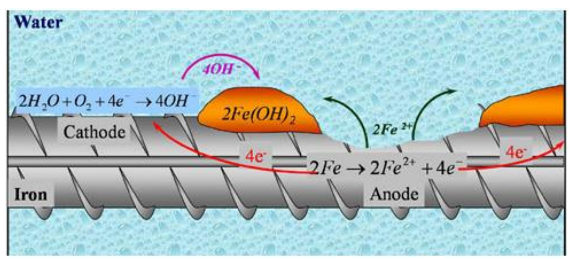

Figure 23: Uniform Corrosion (PCA, 2013)

The corrosions that are more important are localized ones such as pitting or crevice. The failure due to localized corrosions happens earlier than the uniform corrosion (Craig, 2006).

4.13.6 Hydrogen Damage

Sometimes hydrogen ions are available in the sea water or the well medium. Corrosion or rusting, and electroplating also generates hydrogen ions. Hydrogen ions also are released as a result of a cathodic protection. Therefore, cathodic protection can be used to limited potential. Although cathodic protection systems are not the only source of hydrogen ions, it must be taken into consideration in a corrosion study. Another effect of hydrogen on a material as shown in Figure 24 occurs under the tensile stress and is called Hydrogen Induced Stress Cracking (HISC). Atomic hydrogen (H) permeates into the steel and nucleates to the hydrogen molecule in the defects and inclusions of the steel. This condition will result in several high pressure void spaces that make the steel susceptible to cracking or blistering. Hydrogen attack is detrimental to ductility and may result in a failure due to embrittlement.

In the presence of hydrogen ions, if the conditioning tanks and recycle vessels are just protected by a cladding layer, the substrate material of a clad vessel should be resistant to hydrogen attack as well.

Figure 24: Hydrogen Induced Cracking (Masteel, 2013)

4.13.7 Environmental Induced Cracking

Toxic gases like H_2S that are present in the environment can lead to death if an individual is exposed to high concentrations of those for a certain period of time. Environmental Induced Cracking (EIC), is a general term for a brittle mechanical failure that results from a synergism between tensile stress and corrosive environment . Three main factors, which are mentioned here, should act simultaneously to cause EIC to occur:

1- Corrosive environment
2- Tensile stress (not compressive)
3- Susceptible material

H_2S in the presence of water can cause uniform corrosion, but the cracking is the most serious type of corrosion that must be controlled. H_2S and CO_2 will be discussed hereafter:

4.13.7.1 Sulfide Stress Cracking

Figure 25 shows an example of sulfide stress cracking. The process of corrosion requires wet hydrogen sulfide, which results in Sulfide Stress Cracking (SSC) as a type of Stress Corrosion Cracking (SCC). Other factors such as pH, hydrogen sulfide ion concentration and temperature affect the process.

The combination of material under tensile stress in the corrosive environment is necessary to initiate the cracking.

Higher strength materials are more susceptible to cracking. Therefore practices and standards such as NACE MR0175 have recommendations to limit the hardness of different materials, which are reported in Hardness Rockwell C number (HRC).

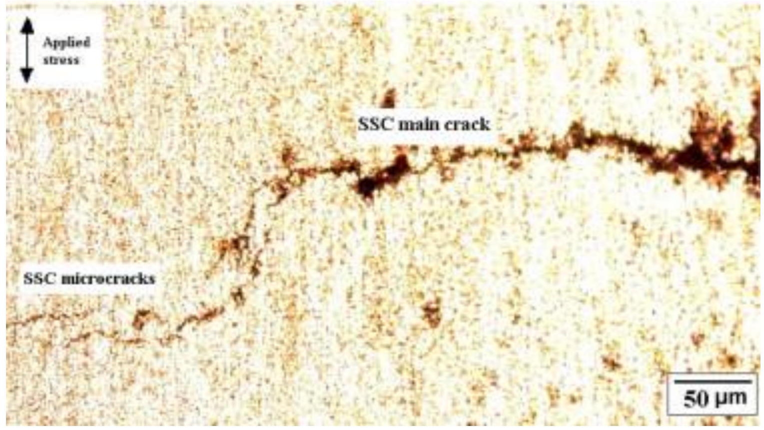

Figure 25: Longitudinal Section of an X100 Specimen Loaded at 65%YS Showing main SSC Crack (right of image) and Small Microcracks (left of image) (Al-Mansoura, 2009)

Figures 26 and 27 show the partial pressure of hydrogen sulfide more than 0.05 psi as a criterion for sour service operation in single or multiphase flows. Cracking happens in the presence of water; however the effects of pH, temperature and total presure are also important.

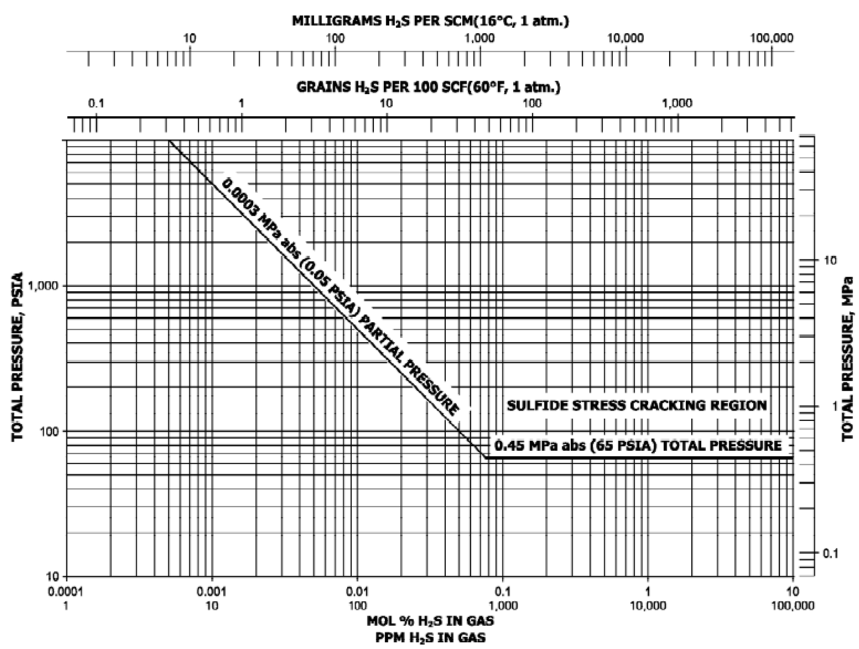

Figure 26: Sour Gas System in Single Flow (MR0175,Petroleum and Natural Gas Industries – Materials for use in H2S-containing Environments in Oil and Gas Priduction, 2003)

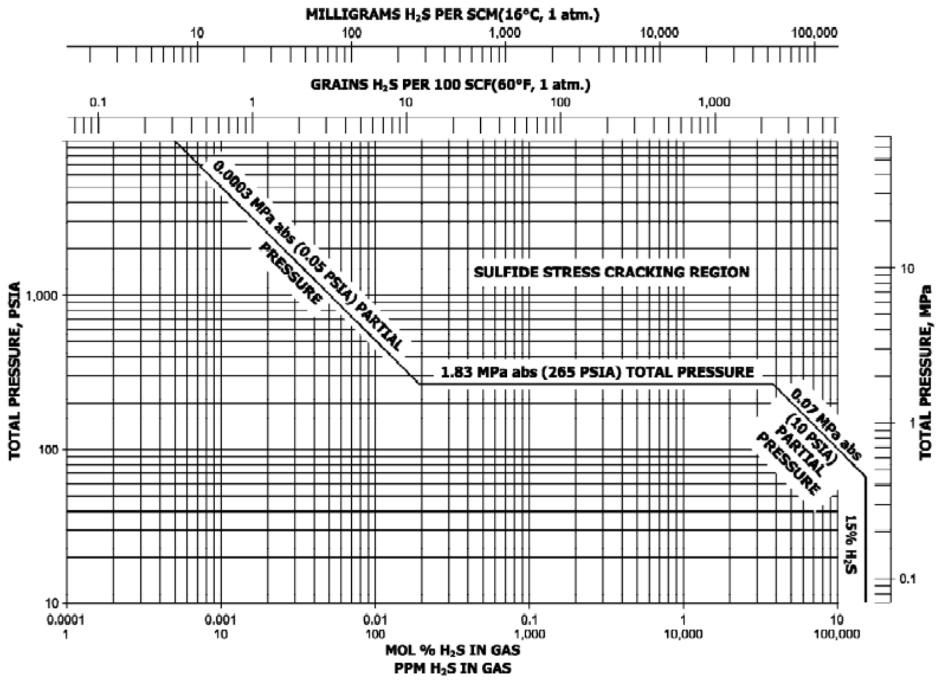

Figure 27: Sour Gas System in Mixed Flow (MR0175,Petroleum and Natural Gas Industries – Materials for use in H2S-containing Environments in Oil and Gas Priduction, 2003)

In the wet environment, hydrogen sulfide affects steel based on the following reactions (Singer, 2011). At the beginning, H_2S dissolves and dissociates. In the next step the product of the reaction dissociates and the process continues:

$$H_2S_{(g)} \Leftrightarrow H_2S_{(aq)} \tag{4.31}$$

$$H_2S_{(aq)} \Leftrightarrow H^+_{(aq)} + HS^-_{(aq)} \tag{4.32}$$

$$HS^-_{(aq)} \Leftrightarrow H^+_{(aq)} + S^{2-}_{(aq)} \tag{4.33}$$

In the next step, H_2S reduces to the following:

$$2H_2S_{(aq)} + 2e^- \rightarrow H_{2(g)} + 2HS^-_{(aq)} \tag{4.34}$$

Then FeS is formed by precipitation and finally solid state reaction:

$$Fe^{2+}_{(aq)} + S^{2-}_{(aq)} \Leftrightarrow Fes_{(s)} \tag{4.35}$$

$$Fe_{(s)} + H_2S \rightarrow FeS_{(s)} + H_2 \tag{4.36}$$

Sulfide (S^{2-}) is called poisoning ion, because it doesn't let two hydrogen ions combine and form H_2; instead hydrogen ions are reduced to atomic hydrogen, which is absorbed into the steel leading to hydrogen embrittlement.

H_2S uniform corrosion is not as serious as its cracking effects due to released hydrogen ions. Hydrogen ions reduce the pH and cause hydrogen attack. However since the source of hydrogen is hydrogen sulfide, it is called Sulfide Stress Cracking (SSC). The NACE MR 0175 standard outlines specific materials and processes suitable for the sour service application. Methods to mitigate the H_2S corrosion are limited to material selection. Inhibitors that can increase the pH or galvanizing methods that can cover the steel face may also reduce the cracking. H_2S scavenger is another practice to reduce H_2S corrosion (Nguyen Phuong Tung, 2001).

4.13.7.2 CO_2 Corrosion

Carbon dioxide (CO_2) is found in oil and gas fields in different concentrations as an impurity. Similar to H_2S, Dry CO_2 is not a corrosive medium to metals and alloys. However, in the presence of water, carbonic acid (H_2CO_3) is formed which causes severe corrosion on the infrastructure. Carbon dioxide corrosion occurs according the following steps (Singer, 2011):

Water dissociation:

$$H_2O \Leftrightarrow H^+_{(aq)} + OH^-_{(aq)} \tag{4.37}$$

Carbon dioxide dissolution:

$$CO_{2(g)} \rightarrow CO_{2(aq)} \tag{4.38}$$

Carbon dioxide hydration (slow step):

$$CO_{2(aq)} + H_2O_{(1)} \leftrightarrow H_2CO_{3(aq)} \tag{4.39}$$

Carbon acid dissociation:

$$H_2CO_{3(aq)} \leftrightarrow H^+_{(aq)} + HCO^-_{3(aq)} \tag{4.40}$$

Bi-carbonate ion dissociation:

$$HCO^-_{3(aq)} \leftrightarrow H^+_{(aq)} + CO^{2-}_{3(aq)} \tag{4.41}$$

Proton reduction

$$2H^+_{(aq)} + 2e^- \rightarrow H_{2(g)} \tag{4.42}$$

Carbonic acid reduction:

$$2H_2CO_{3(aq)} + 2e^- \rightarrow 2HCO^-_{3(aq)} + H_{2(g)} \tag{4.43}$$

Iron oxidation:

$$Fe_{(s)} \rightarrow Fe^{2+}_{(aq)} + 2e^{-} \tag{4.44}$$

Iron carbonate precipitation (if supersaturated):

$$Fe^{2+}_{(aq)} + CO_3^{2-} \leftrightarrow FeCO_{3(s)} \tag{4.45}$$

The corrosive effect of carbon dioxide is dependent on CO_2 concentration, material type, pH, temperature and water chemistry (TWI, 2013). The formed carbonic acid reduces the pH. Higher concentrations of CO_2 increase general corrosion and pitting of carbon and stainless steel alloys. Higher partial pressure and temperature will increase the corrosion rates. Figure 28 shows the effect of temperature on CO_2 corrosion in different partial pressures.

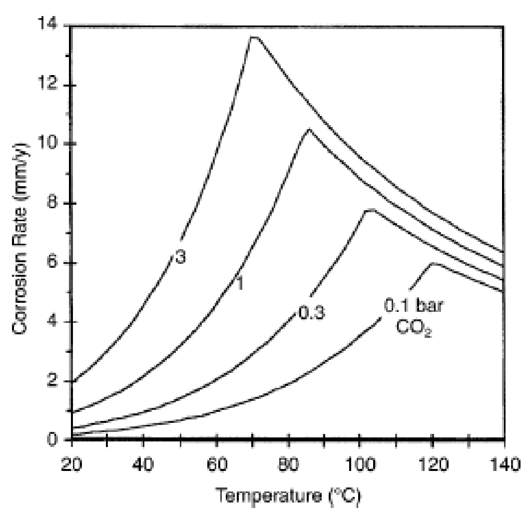

Figure 28: Effect of Temperature on Carbon Dioxide Corrosion in Different CO2 Partial Pressure (Kermani, 2003)

On the other hand, lower pH increases the acidity and corrosion. It has been reported that temperatures in excess of 150-160 °F, (70-80°C) can cause CO_2 cracking (Bajvani, 2011).

Special attention must be made in piping design and valve and orifice selection, because corrosion occurs at locations where CO_2 condenses and increases partial pressure.

Since some of the reservoirs have CO_2 and H_2S simultaneously, it is also necessary to study this type of combined corrosion. Figure 29 represents the effect of H_2S and CO_2:

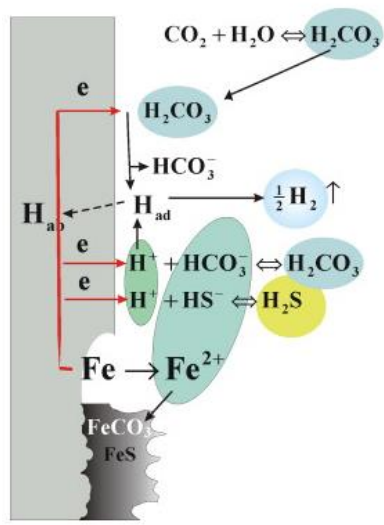

Figure 29: Effect of CO_2 and H_2S on Composition (Banasu, 2007)

4.13.8 Galvanic Corrosion

When two or more dissimilar metals are located in an electrolyte and connected electrically, galvanic corrosion occurs. The metal with least resistance to corrosion, called active metal, works as an anode, while the other one as a noble material becomes cathode. For Aluminum, Nickel and stainless steel alloys a thin layer, called the passive layer, covers the metal, which increases resistance to corrosion significantly. Common metals in galvanic series are given in Table 7 (Craig, 2006)**.** The metals are listed in the table based on corrosion potential from anodic to cathodic materials.

Special attention must be paid to the temperature and the area of anode vs. cathode which are the factors affecting galvanic corrosion.

Table 7: Galvanic Series for Sea Water (Craig, 2006)

Magnesium
Zinc (hot-dip, die cast, or plated)
Tin (plated)
Stainless steel 430 (active)
Lead
Steel 1010
Iron (cast)
Stainless steel 410 (active)
Copper (plated, cast, or wrought)
Nickel (plated)
Chromium (Plated)
Stainless steel 301, 304, 310, 410, 430, (active)
Stainless steel 316L (active)
Bronze 220
Copper 110
Stainless steel 201 (active)
Carpenter 20 (active)
Stainless steel 316, 321 (active)
Stainless steel 301, 304,321, 316L (passive)
Stainless steel 301 (passive)
Titanium
Silver
Gold
Graphite

The other important rule is to consider high anode to cathode ratios. For example, in a welded connection between a stainless steel flange and a carbon steel pipe, corrosion starts from the weld line on the carbon steel side. A bronze pump connected to a steel pipe would be common in offshore industries. In this case, the steel works as an anode and is prone to corrosion. Therefore, the pump must be coated to reduce the cathode area (Craig, 2006).

Weld lines in the pump piping systems are also susceptible to galvanic corrosion. In the weld zones, high residual stress in boundary layers is prone to react with corrosive environment and release the energy.

The heat from the welding process changes the microstructure of the weld metal and heat affected zone. Dissimilar microstructure creates the anode and cathode and galvanic corrosion occurs. Cathodic protection is usually applied to control galvanic corrosion.

4.13.9 Crevice Corrosion and Pitting

Stainless steels and aluminum are corrosion resistant materials because of the passive film that forms on their surface. The thin oxide layer forms inherently on the alloy surface and protect it from corrosion. The one disadvantage of the passive film is susceptibility to localized breakdown, which will cause accelerated dissolution of the metal. This phenomena is called pitting corrosion on the open surfaces and crevice corrosion at the confined voids such as bolt head or beneath the gaskets. When the pit is initiated, the unpitted surface works as a cathode and the pit becomes an anode. A large cathode to anode ratio in a low pH environment creates a galvanic cell and accelerates corrosion dramatically. The pit can grow to the whole section and prediction of the life of the part becomes difficult. Figure 30 shows an example of a deep pitting on a metal surface. In subsea applications, pitting normally occurs on the surfaces in contact with sea water; however records show pitting corrosion even inside the piping. Pitting accelerates if the environment contains choride, CO_2, H_2S and dissolved oxygen.

Figure 30: Deep Pits in Metal (Merus, 2013)

The mechanism for crevice corrosion is similar to pitting corrosion, but more applicable to two surfaces in contact, either between two metals or gasket and metal. The variation in oxygen between crevice and surrounding environment will result in a potential cell. The crevice corrosion depends on the size of crevice.

It is recommended to use welded connections instead of flanged connections to reduce the risk of crevice corrosion. A stagnation point as a source of pitting and crevice corrosion should be avoided as much as possible in the piping, valve and connections.

4.13.10 Erosion Corrosion

As explained in section 4.12.5, high velocity can increase erosion especially in the reservoirs that contain sand, gas or droplets in production fluid. Impingement occurs in the locations such as bends that change the direction of fluid. As shown in Figure 31, the droplets or bubbles will hammer against the metal surface and can cause rapid failure.

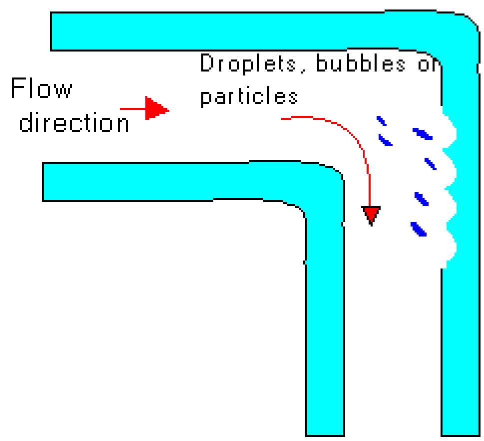

Figure 31: Erosion (Impingement) Corrosion (Alu, 2013)

Since there is a risk in carrying bubbles, droplets and sand in the piping of multiphase pumps, it is very important to filter sands, and it is recommended to use a demister in conditioning tank to remove water drops and design smooth bends in piping.

4.13.11 Intergranualr Corrosion

Interganular corrosion takes place when the chemistry between the grain and the grain boundaries changes. The corrosion starts from grain boundaries and propagates faster than the metal matrix. This kind of corrosion is more predominant in austenitic stainless steels. To reduce the risk of intergranular corrosion, carbon content should be kept in the range of 0.03 to 0.08 C. If the steel temperature is held between 800-1500 °F, (425 to 815 °C), chromium carbide precipitates in grain boundaries and depletes chromium from vicinity of grain boundaries and may result in a intergranular attack (Jones, 1996). Figure 32 shows a sample of intergranular cracking in an impeller.

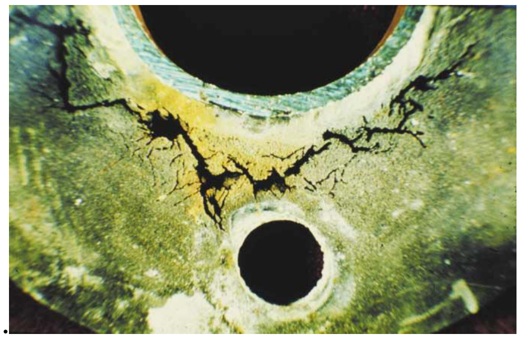

Figure 32: Intergranular Cracking on Impeller (CPC, 2013)

4.13.12 Microbial Induced Corrosion

Since the organisms in the sea water can live and survive in salinity levels up to 40 mg/L, (Lehigh, 2013). Microbial Induced Corrosion (MIC) must be considered in subsea application. In the stagnation points or low velocity sections of the flow lines and equipment especially in long shun-in periods, bacteria and other microbes accumulate and deposit. It occurs more when the temperature is from 40 to 122 °F, (10 to 50 ℃) and the pH is in the range of 4 to 9. Elements such as magnesium, nitrogen, phosphorous, and sulphates as a nutrition for bacteria can accelerate MIC. On the other hand, oxygen is trapped under the deposit and accelerates the corrosion in pitting and galvanic form. Microbial activities produce acids (and sometimes H_2S) which increase the corrosion rate (A.K. Samant, 1998). All corrosion types which influence subsea pumps have been discussed are listed in Table 8.

Table 8: Applicable Corrosion Types to Different Parts of Subsea Pump

Corrosion type	Corrosive agent	Effect	Susceptible parts	Mitigation methods
Uniform corrosion	Water containing H, H_2S, O2, CO_2	Material degradation, Thickness reduction	Structure, piping, conditioning tank, recycle tank, Process parts	1-Cathodic protection 2-Coating 3-Painting 4-Oxygen Scavenger 5-Proper material
Hydrogen corrosion (damage)	H+ and Atomic Hydrogen (from cathodic protection or H_2S containing env.)	Hydrogen embrittlement	Process parts , piping	1-Proper material 2-Proper welding procedure 3-H_2S Scavenger 4-Controled cathodic protection
		HISC	All process part, piping	
		Hydrogen Induced Cracking	Structure, piping, conditioning tank, recycle tank, Process parts	
EIC	Water containing H_2S+CO_2+ Tensile stress	Sulfide Stress Cracking (SSC) Stress Corrosion Cracking (SCC	All process parts, piping	1-H_2S Scavenger 2-Inhibitors 3- Proper material selection
Galvanic corrosion	Dissimilar material	Material degradation in the joints	Flanges, plugs, Gaskets, welding, piping, structure	1-Cathodic protection 2- Proper material selection 3- Proper coating or painting
Crevice corrosion and Pitting	Sea water, Water containing H_2S+CO_2	H_2S, CO_2, Chloride and O2 accelerate pitting	Piping, Flanges , Seals	1- Avoid flange connection 2- Avoid traps and stagnation points 3-Weld connection 4- Proper material selection
Erosion Corrosion	Sand, liquid drop let, gas bubbles	Wear and erosion	Seals (1), Casing, impeller, screws Piping	1-Avoid sharp bend 2- Proper material selection 3-Surface hardening (Nitridation)
Intergranular corrosion	Sea water, Water containing H, H_2S+CO_2, O_2,	Material degradation	Impeller	1-Proper material selection 2- Proper heat treatment
Microbial Induced Corrosion (MIC)	Bacteria	Pitting, Crevice	Piping, process parts	1-Bacteria control by inhibitors. 2- Avoid stagnation points 3-O_2 Scavenger

(1) Not applicable if clean barriers fluid is supplied from topside.

CHAPTER 5

MATERIAL SELECTION

5.1 Introduction

For almost all subsea production units, the clients expect pumps to work non-stop for a long period, which requires a highly reliable system including electric motor, sealing parts, coupling, bearings and connections.

As explained before, the important parameters in pump type selection are DP, flow rate, GVF, fluid viscosity, sand content, Opex, Capex, reliability and asset life. One of the most critical aspects in pump design is material selection. Figure 33 shows a typical subsea pump and its main components:

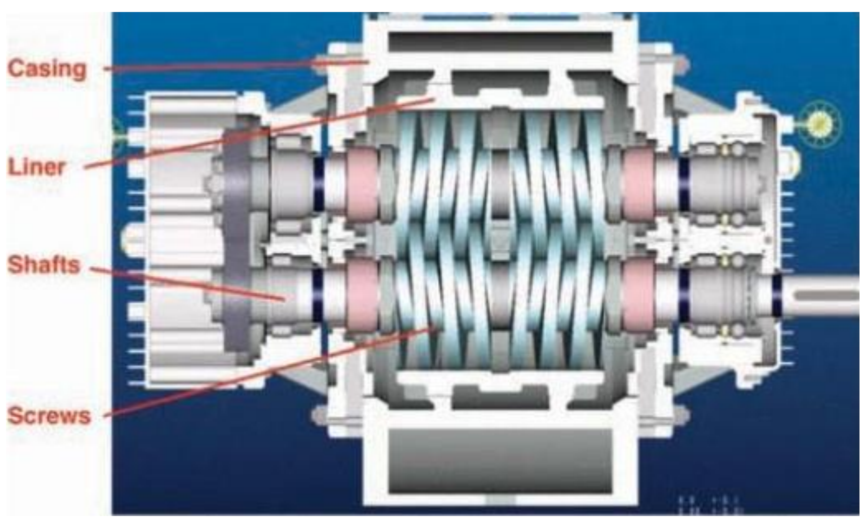

Figure 33: Typical Subsea Pump Components (Bornemann, 2013)

In this chapter, a classic approach for material selection based on the reference book (Ashby, 2011) is followed. The casing as shown in Figure 33 is the main part for which appropriate material is selected. For other parts, the same approach is applicable. The approach for this task is explained hereafter.

5.2 Strategy

The material selection is performed based on a classic procedure and without any bias to select the material. The main steps are listed here:

1- The problem statement, setting the sense.
2- The model, identifying function, constraints, objectives, and free variables, from which emerge the attribute limits and material indices.
3- The selection in which the full menu of materials is reduced by screening and ranking to a short-list of viable candidates.
4- The postscript, allowing a commentary on results and philosophy.

5.3 Problem Statement

It is clear in Figure 33 that the casing is in contact with high pressure process fluid from inside and sea water from outside. Although the casing is covered with an insulation, it must be able to tolerate severe conditions in case of insulation failure.

5.4 The Model

The applicable load and conditions on pump casing are listed below:

1- **Hydrostatic pressure** of water or liquid above pump due to weight of the medium

2- **Internal pressure** provided by pump impeller when working

The casing is supposed to work properly in the through life time (normally 3-5 years) under all working conditions. Since the casing can be assumed as a small pressure vessel, not a big one, the design criteria is “yield before break” not “ leak before break”.

3- Corrosion and erosion are main concerns as discussed before.

4- Almost all wells produce some percent of sands with specific shape and size. The pump should be able to handle that and must tolerate the erosive effect of sand particles.

5- In shut down condition, the well fluid flow stops and outside low temperature causes wax or hydrate formation. The seabed temperature (~ 40 °F) must be predicted in design and it is necessary to check Charpy V-Notch impact values at the low temperature to make sure that it is not a transient temperature for the material. For carbon and low alloy steel it might be an issue, but when stainless steel grade is selected, this temperature would not be a transient one.

6- High temperature fluids in some cases are the other concern. (In some cases it may exceed 750 °F (400 ℃), which is harmful to sealing elements.). But in case of low flow operation that will cause high temperature, it will be a contributing factor that may result in more severe conditions. This parameter can affect the elastomers and doesn’t have a severe effect on the metallic material and is a free variable.

7- Gas volume fraction (GVF): In high GVF, for example > 75%, the pump will work with minimum fluid for a while. In this period the temperature will increase due to friction losses. In case of long low flow operation, the temperature can reach ranges of 550-600 °F that can be harmful to the elastomers, but not the metallic parts. As explained before, the application of the recycle line in pumps is to ensure minimum flow to release this temperature. This parameter doesn't have a severe effect on material selection and is a free variable.

8- Equipment life: Since the pump is not accessible after installation or any physical access costs thousands of dollars, it is expected that the pump will work for a minimum 3-5 years without intervention. It shows the importance of correct material selection.

9- Opex and Capex: Although selection of high grade material will satisfy project technical requirements, the imposed cost would be a limiting factor which sometimes governs the selection. Therefore material selection should be done with consideration of cost issues.

10- Weight and Size: In subsea application, weight and size must be minimized as much as possible to reduce the installation and intervention costs.

Internal pressure , external pressure and corrosion are the most complicated and severe concerns in the material selection. Casing data is assumed as below to have an estimation of required yield / offset strength.

Casing Outside Diameter= 168 mm,

Casing Thickness= 14.1 mm

Water depth: 100 m

Sea water density: 1028 kg / m^3

g= 9.8 m / s^2

Internal design pressure= 23 Mpa

The casing is intended to work properly over the entire lifetime of the pump under all working conditions. According to chapter 6 of the reference book (Ashby, 2011), the casing can be assumed as a small pressure vessel; therefore, the design criteria is “yield before break” not “leak before break”.

5.5 Material Selection

External pressure is 1 Mpa and the internal pressure is 23 Mpa. Internal pressure is assumed to be the design pressure. The basis of design is “yield before break” method.

The hoop stress in the casing can be considered as:

$$\sigma_h = \frac{p.ID}{2t} \tag{5-1}$$

Where, p is working pressure, ID is diameter and t is wall thickness respectively.

t must satisfy 0.6 * Sy as the design criteria; therefore Sy should be more than 228 Mpa.

As per API 610 standard, radiographic test should be performed as per ASME Sec 5, Article 2 and 22. The acceptance criteria of radiographic test for both cases are available in ASME, Sec IV, Div 1 as per Figure 34.

Type of inspection	Methods	Acceptance criteria	
		For fabrications	For castings
Radiography	Section V, Articles 2 and 22 of the ASME Code	Section VIII , Division 1, UW-51 (for 100 % radiography) and UW-52 (for spot radiography) of the ASME Code	Section VIII, Division 1, Appendix 7 of the ASME Code
Ultrasonic inspection	Section V, Articles 5 and 23 of the ASME Code	Section VIII, Division 1, Appendix 12, of the ASME Code	Section VIII, Division 1, Appendix 7, of the ASME Code
Magnetic particle inspection	Section V, Articles 7 and 25 of the ASME Code	Section VIII, Division 1, Appendix 6 of the ASME Code	Section VIII, Division 1, Appendix 7, of the ASME Code
Liquid penetrant inspection	Section V, Articles 6 and 24 of the ASME Code	Section VIII, Division 1, Appendix 8 of the ASME Code	Section VIII, Division 1, Appendix 7, of the ASME Code

Figure 34: Material Inspection Standards (API 610, Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries, 2003)

Based on the radiographic test it can be established that the casing has no crack or flaw of diameter greater than $2a_c^*$ and the stress required to make the crack propagate is:

$$\sigma_h = \frac{CK_{1C}}{\sqrt{\pi a_c^*}} \tag{5-2}$$

Where, C is constant near unity and K_{1C} is plane strain fracture toughness.

On the other hand safety can be achieved by ensuring that the maximum working pressure is:

$$p < \frac{2t}{ID} \frac{K_{1C}}{\sqrt{\pi a_c^*}} \tag{5-3}$$

Therefore, the material with the greatest value of K_{1C} will carry the highest pressure as below:

$$M_1 = K_{1C} \tag{5-4}$$

This design is still not fail-safe because in case of incorrect inspection or a crack greater than a_c^*, the casing would fail. The safe criteria is achieved by defining that if the stress reaches 0.6 Sy called σ_f, the crack will not propagate.

For this condition σ equal to 0.6 Sy is set:

$$\pi a_c \leq C^2 [\frac{K_{1C}}{\sigma_f}]^2 \tag{5-5}$$

Now by choosing the material that maximizes $\frac{K_{1C}}{\sigma_f}$, the tolerable crack size and the integrity of the vessel will be maximized. This parameter may be defined as M_2:

$$M_2 = \frac{K_{1C}}{\sigma_f} \tag{5-6}$$

The working condition of the casing is critical, because it is under cyclic loading and a corrosive medium, therefore it would be necessary to do material examination periodically.

The crack should not be so large that it penetrates both inner and outer surfaces of the casing cylinder. Safety can be achieved by preventing the development of the crack. This will be obtained when the stress is limited to the value below;

$$\sigma = \frac{CK_{1C}}{\sqrt{\pi t / 2}} \tag{5-7}$$

The cylinder thickness is designed to withstand the differential pressure. It means:

$$t > \frac{p.ID}{2\sigma_f} \tag{5-8}$$

Inserting this equation into the previous one with $\sigma = \sigma_f$ gives:

$$p \leq \frac{4C^2}{\pi.ID}\left(\frac{K_{1C}^2}{\sigma_f}\right) \tag{5-9}$$

It means the maximum pressure is carried safely with largest value of M_3:

$$M_3 = \frac{K_{1C}^2}{\sigma_f} \tag{5-10}$$

Although the parameters M_2 and M_3 can be maximized when σ_f is minimum, it is not possible to select smaller σ_f, because the cylinder should be as thin as possible to decrease the weight.

K_{1C} (M_1) which is measured fracture toughness plotted against strength σ_f in Figure 35. K_{1C} for steel can be assumed equal to 50 ($MPa.m^{1/2}$), therefore M_1, M_2, M_3 and M_4 are 50, 0.22, 11 and 228 respectively.

5.5.1 Post Script

The selection process is explained hereafter based on the “yield before break concept”. Lines M_2 and M_3 lead us to a polygon at the top right side of the sketch. Minimum σ_f equal to 228 Mpa eliminates other materials. Hence, some kinds of stainless steel, nickel alloys and low alloy families satisfy the requirements. In this area K_{1C} has the highest rate which means the material is ductile.

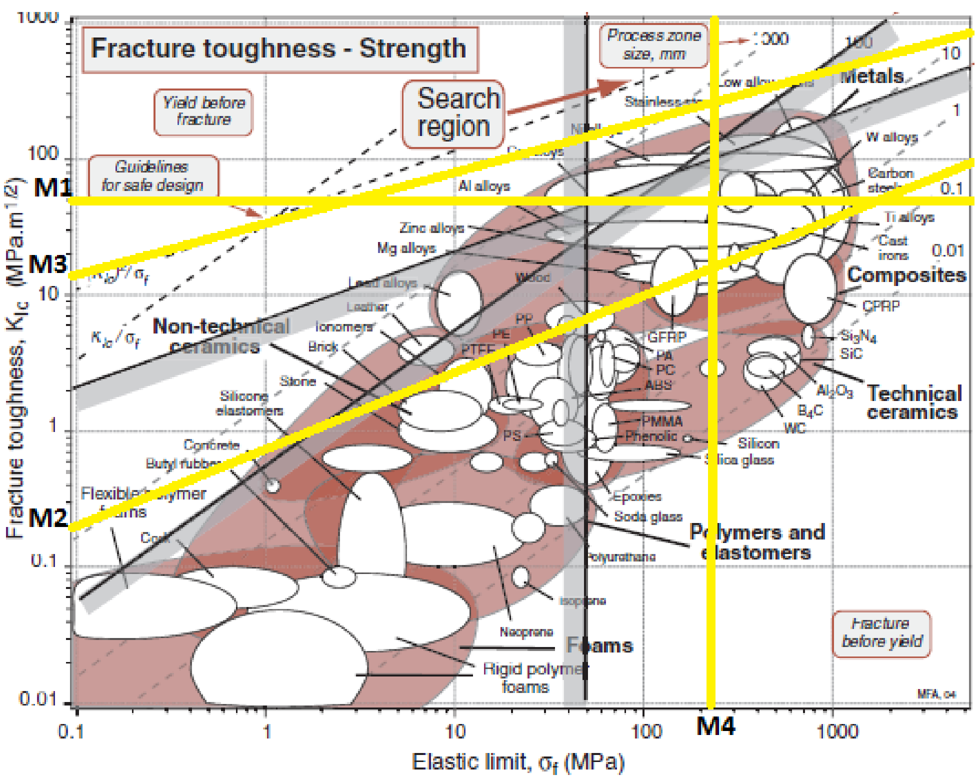

Figure 35: Fracture Toughness Versus Strength (Ashby, 2011)

Generally the stainless steels are resilient in a corrosive environment, but an accurate selection must be performed among the different grades of stainless steels.

5.7 Stainless Steel Grade Selection

Stainless steel as a composition of main elements such as Iron, Carbon, Chromium and Nickel is famous because of the resistance to corrosion. It is easier to shape and is more durable in the long term, which is extremely important in subsea application. Due to the expected life and severe working condition it would be better to select a stainless steel sub class rather than carbon steel. It will cost more but it is necessary to satisfy durability requirements.

Several types of stainless steel with different characteristics are available. Since the wetted part of the pump is in contact with almost all kinds of corrosions such as pitting, crevice, HIC and SSC, selection of material that covers all those issues is crucial. Let's look at the main types of stainless steels at first to find which one is more suitable:

Ferritic – The main elements are Iron, Chromium with small amounts of Carbon usually less than 0.10%. They are usually limited in use due to lack of toughness in welds. Lower chromium and nickel content result in reduced corrosion resistance which is not a good fit for subsea application.

Austenitic – The elements of microstructure are Iron, Chromium, Nickel, Manganese and Nitrogen. This structure gives these steels combination of weld ability and formability. Low strength is the negative factor of austenitic steels. They are also vulnerable to stress corrosion cracking and therefore are not a good category for subsea application.

Martensitic - These steels are similar to ferritic steels but have higher Carbon levels up as high as 1%. They are used where high strength is required. Although Martensitic stainless steels are the most reasonable type of stainless steels; they are not as corrosion-resistant as the other classes and cannot satisfy the subsea requirements.

Duplex - The microstructure of Duplex steel is approximately 50% ferritic and 50% austenitic. Figure 38 shows a sample of Duplex microstructure.

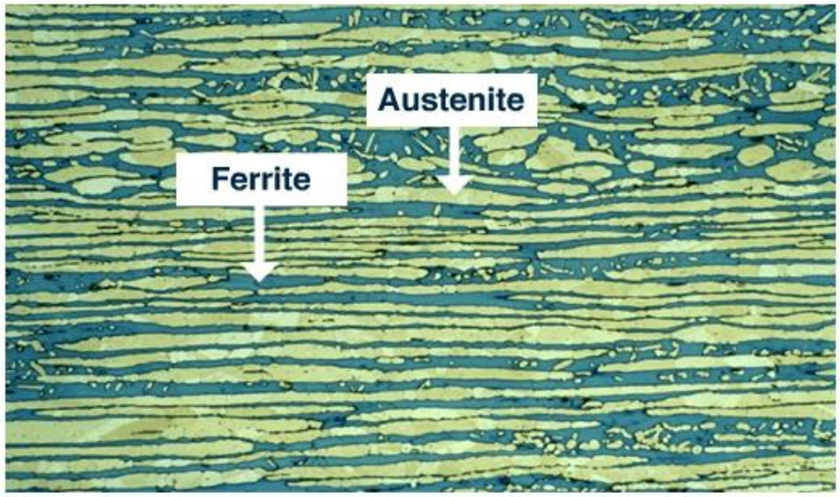

Figure 36: Duplex Stainless Steel Contains Ferrite and Austenite Microstructures (Imoa, 2013)]

Therefore, they have a higher strength than either ferritic or austenitic steels. They are resistant to stress corrosion cracking. Duplex steels are formulated to have resistance to stress corrosion cracking. “Super duplex” steels have enhanced strength and resistance to all forms of corrosion compared to standard austenitic steels. Duplex family has almost twice the strength compared to austenitic stainless steels and improved resistance to localized corrosion such as pitting, crevice corrosion and stress corrosion cracking.

Among all four stainless steel categories, Duplex and its sub category, Super Duplex, can satisfy all the corrosion categories as well as strength and hardness.

The material that can satisfy the subsea requirements is Super duplex. Looking at table H.2 of API 610 and table 3 of NACE MR0175, gives the appropriate material grade for the selected part. Figure 39 represents table 3 of NACE standard and different types of Duplex materials including wrought and Cast Duplex grades. The cast materials are not applicable for subsea pumps.

Ferritic	Martensitic	Precipitation-Hardening	Austenitic	Duplex (Austenitic/Ferritic)[D] (Wrought condition only)
AISI 405 430	AISI 410 501	ASTM A 453 Gr 660[A] A 638 Gr 660[A]	AISI 302 304 304L 305 308 309 310 316 316L 317 321 347	UNS S31260 UNS S31803 UNS S32404 UNS S32550 UNS S32760 UNS S39274 UNS S39277
ASTM A 268 TP 405, TP 430, TP XM 27, TP XM 33	ASTM A 217 Gr CA 15 A 268 Gr TP 410 A 743 Gr CA 15M A 487 Cl CA 15M A 487 Cl CA6NM UNS S42400	UNS S17400 UNS S45000 UNS S66286	ASTM A 182 A 193[B] Gr B8R, B8RA, B8, B8M, B8MA A 194[B] Gr 8R, 8RA, 8A, 8MA A 320[B] Gr B8, B8M A 351 Gr CF3, CF8, CF3M, CF8M, CN7M[C] A 743 Gr CN7M[D] A 744 Gr CN7M[D] B 463 B 473	Cast Duplex (Austenitic/Ferritic) stainless steel Z 6 CNDU 28.08 M, NF A 320-55 French National Standard UNS J93380 UNS J93404

Figure 37: NACE MR 0175, Table 3 (MR0175,Petroleum and Natural Gas Industries – Materials for use in H2S-containing Environments in Oil and Gas Priduction, 2003)

Table H-2 from API 610 standard in Figure 40 shows that A 182 Gr. F51 as a forged cylinder can be selected. The same materials are offered for other parts. For example A479/A276 for shaft and A 790 for piping system.

Material Class	Applications	USA ASTM	USA UNS [a]	International ISO	Europe EN [b]	Europe Grade	Japan Material No	Japan JIS
Duplex Stainless Steel	Pressure Castings	A 351 Gr CD4 MCu A 890 Gr 1 B	J93370 J 93372		EN 10213-4	GX2 CrNiMoCuN 25-6-3-3	1.4517	
		A 890 Gr 3A	J93371					G 5121, Gr. SCS 11
		A 890 Gr 4A	J92205		EN 10213-4	GX2 CrNiMoCuN 25-6-3-3	1.4517	
	Wrought / Forgings	A 182 Gr F 51	S 31803		EN 10250-4 EN 10222-5	X2 Cr Ni Mo N 22-5-3	1.4462	G 4319, Cl SUS 329
		A 479	S 32550		EN 10088-3	X2 Cr Ni Mo Cu N 25-6-3	1.4507	
	Bar Stock	A 276-S31803	S 31 803		EN 10088-3	X2 Cr Ni Mo N 22-5-3	1.4462	G 4303, Grl SUS 329 Gr SUS 329
	Plate	A 240-S31803	S 31 803		EN 10028-7	X2 Cr Ni Mo N 22-5-3	1.4462	G 4303, Gr SUS 329
	Pipe	A 790-S31803	S 31 803					G 3459, Gr. SUS 329J3LTP
	Fittings	A 182 Gr F 51	S 31803		EN 10250-4 EN 10222-5	X2 Cr Ni Mo N 22-5-3	1.4462	
	Bolts and Studs	A 276-S31803	S 31 803		EN 10088-3	X2 Cr Ni Mo N 22-5-3	1.4462	G 4303, Gr SUS 329
	Nuts	A 276-S31803	S 31 803		EN 10088-3	X2 Cr Ni Mo N 22-5-3	1.4462	G 4303, Gr SUS 329

Figure 38: Material Specification for Pump Parts (API 610, Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries, 2003)

The minimum yield strength of the selected material is 80 ksi (551 Mpa) which is twice compared to what is necessary.

Super duplex material satisfies the mechanical and corrosion requirements. The disadvantage of super duplex is the availability, machinability and weld ability. Sometimes it takes more than a year to prepare super duplex material. It could make a significant difference, if another option was available. This option is Austenitic Nickel-Chromium Super Alloys called Inconel that gives even better temperature, strength and corrosion resistance characteristics which means this material will last more in the harsh environment of the seabed. The yield strength of this grade is similar to super duplex (~ 550 Mpa).

Similarly the disadvantages of Inconel are the Capex (Capital Cost) and availability, because sometimes the delivery time is a more important concern for Inconel.

The material base for other components such as diffuser, rotor and sealing system is almost the same. According to Nace MR0175, for the sour service application, the Hardness Rockwell C number should be kept below the certain limit to reduce the risk of brittle fracture. Even for the parts such as wear-ring it must be followed.

CHAPTER 6

SELECTION CRITERIA FOR SUBSEA PUMPS

6.1 Procedure

The procedure for pump sizing and selection is explained in this chapter. The project data must be available for early, mid and late life. The limitations of each multiphase pump type are listed in table 9.

Table 9: Limits of Each Multiphase Pump

Limits of Each Multiphase Pump				
		Hydrodynamic (HAP)	**ESP**	**Positive Displacement (TSP)**
1	Maximum capacity (bpde)	> 150,000	30,000	< 150,000
2	Minimum capacity	Limited	Limited	Limited
3	Variation in Flow rate	No limit	Limited	Limited
4	Differential pressure (psi)	< 2,900	< 5,000	< 1,200
5	Series/ Parallel operation	No limit	No limit	Some limits in series operation
6	Erosion Tolerance	Acceptable with mitigation	Acceptable with mitigation	Acceptable with hardened screws
7	High GVF	< 70 %	< 30 %	< 98%
	High water oil ratio	No limit	No limit	No limit
8	High viscosity (cp)	Limited	Limited	No limit (especially>350)
9	Low suction pressure	Limited	Limited	No limit
10	Slug handling	Limited	Limited	No limit
11	High temperature	No limit	No limit	No limit
12	Installation (Ver./ Hor.)	Vertical	Hor. / Ver.	Ver. / Hor.
13	Reliability (MTBF)	Medium	Insufficient data	Insufficient data
14	Opex		High	
15	Capex	High		High

Preliminary evaluation steps are listed in Table 10. Some steps are the same for HAP and TSP; however, the separate steps for each type are also explained. Flow line sizing can be reviwed based on the flow assurance study and boosting requirements.

Table 10: Preliminary Evaluation Steps

Step	Action	
1	Gather the project architecture information	
2	Obtain flow rate data for the project life (gas, oil, water)	
3	Estimate the differential pressure required	
4	Take into consideration series and parallel operation	
5	Obtain viscosity data	
6	Estimate sand production	
7	Perform preliminary pump selection (HAP or TSP)	
	Hydrodynamic (HAP or Centrifugal)	**Positive Displacement (TSP)**
8	Estimate the efficiency	Estimate the axial velocity
9	Calculate the shaft power	Estimate the efficiency
10	Estimate the electric motor power	Estimate the slip
11	Calculate the Specific Speed ((applicable to centrifugal type))	Calculate the shaft power
12	Estimate the shape of the impeller (applicable to centrifugal type)	Estimate the electric motor power
13	Estimate the number of stages (applicable to centrifugal type)	Estimate minimum required sealing liquid
14	Estimate the shape of the performance curve (applicable to centrifugal type)	
15	Specify the sealing system requirements based on API 682	
16	Specify the coupling requirements based on applicable standard	
17	Specify the balancing requirements based on ISO 1940	
18	Specify bearing requirements	
19	Specify the materials	
20	Consult with the manufacturer regarding the items 8 to 19. Manufacturer data is required to perform some calculations.	

The steps that are mentioned in the Table 10 are explained in the following:

1- Gather project information including field architecture, pipe type and roughness, routing, length, depth, well head pressure, down hole pressure, flow rate, required topside pressure and temperature, viscosity and subsea architecture.

2- For higher flow rates (> 150,000 bpd), HAP is the better choice, while for the lower range (< 150,000 bpd) with limited changes, TSP works well.

4- Calculate the required differential pressure. Based on the records of installed subsea pumps, for the differential pressure < 2500 psi, HAP would be the better choice, while For DP<1200 psi, TSP would be better.

5- Study the series or parallel operation. If the flow rate, differential pressure or the reliability requirements (for example having two pumps in parallel) necessitate, TSP has some limitations in a series operation.

6- For the viscosity > 350 cp, TSP is the choice. HAP cannot handle high viscosities efficiently. Viscosity in start-up conditions may be higher.

7- In sand production, the size and expected amount of sand should be specified. HAP can tolerate sand to some extent and TSP can handle that only if the screws are hardened.

The above mentioned criteria is applicable to all types of seabed pumps whether HAP, TSP or ESP.

<u>To have a better picture of HAP or ESP characteristics, following steps should be considered:</u>

1- Estimate the efficiency based on the Jekat formula, (Jekat, 1976) or use the curve:

$$\eta hy = 1 - 0.071 / Q^{0.25} \quad \text{(SI Units…..Q is m3/s)} \tag{6.1}$$

or the following curve that may give a better estimation.

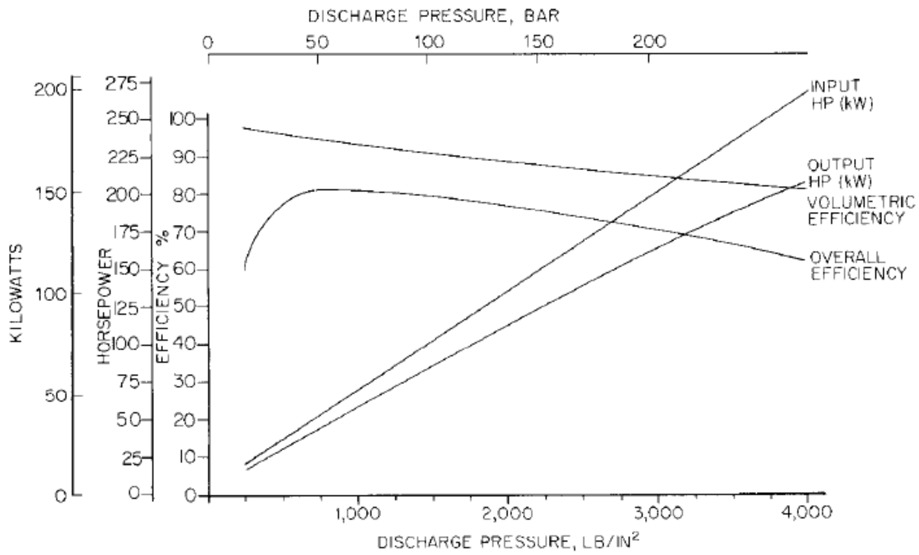

Figure 39: Efficiency Estimate (Karassik, 2008)

2- Convert differential pressure to head and calculate the shaft power:

$$BHP = [\frac{\rho_m ghQ}{3.6\times10^6}] / efficiency \quad (6.2)$$

Where, ρ_m is mixture density in multiphase flow (kg/m^3), g is equal to 9.81 m/s^2, h is defined as Head (m) and Q is the flow rate (m^3/hr).

3- Use Table 11 from API 610 to calculate the rated power. Since almost all subsea pumps are larger than 55 KW, normally 10% is added to shaft power to estimate the rated power.

Table 11: Power Ratings for Electric Motors (API 610, Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries, 2003)

Motor nameplate rating		Percentage of rated pump power
kW	(hp)	%
<22	(< 30)	125
22 to 55	(30 to 75)	115
>55	(> 75)	110

4- Calculate specific speed (more applicable to single phase flow):

$$n_q = \frac{nQ^{0.5}}{h^{0.75}} \tag{6.3}$$

Where, n, Q and h are rpm, flow rate and head respectively.

Standard ranges based on US standard units (Engineering Toolbox, 2013) are mentioned here. The range for multiphase pumps is more similar to that for axial flow and mixed flow.

- Radial flow - 500 < nq < 4,000
- Mixed flow – 2,000 < nq < 8,000
- Axial flow – 7,000 < nq < 20,000

5- Although it is possible to calculate head per stage, impeller diameter, and impeller shape based on the Specific Speed, for multiphase pumps, this information depends on the manufacturer. As shown in Figure 40, the shape of the impeller should be the same as the axial impellers.

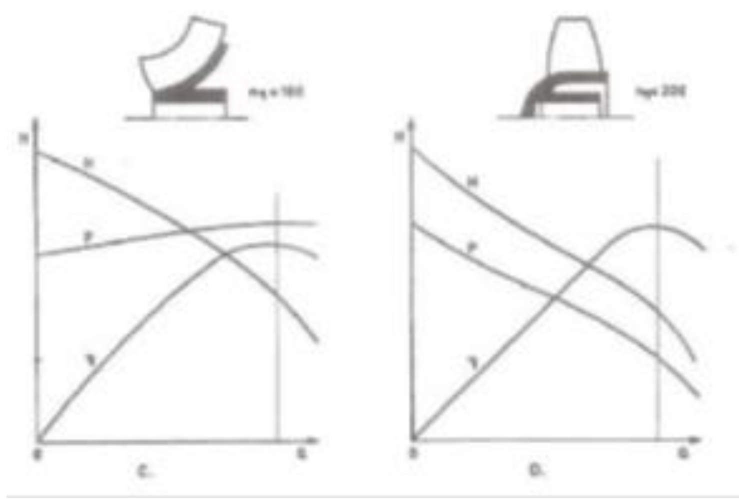

Figure 40: Impeller Shapes Based on the Specific Speed (Sulzer, 2003)

To have an idea about the detail characteristics of the TSP pump, the following steps can be followed:

Steps one to seven here are the same as mentioned before. It is assumed that due to high viscosity, low differential pressure or limited operating range, TSP is the desired type. Some basic calculations can be performed to have a better picture of the pump. This will be helpful for negotiations with vendors (Brennan, 2008).

1- Provide a range of suction lift, flow rate and viscosity of the pump for long term operation. Viscosity in start-up conditions may be higher.
2- Estimate the axial velocity based on the viscosity. Table 12 gives more information. The speed of the pump should be low when handling more viscous fluids to reduce losses. It is not necessarily applicable to subsea, but can be helpful for estimation.

Table 12: Internal Axial Velocity limits (Brennan, 2008)

Liquid	Viscosity, SSU	Velocity, ft/s (m/s)
Diesel oil	32	30 (9)
Lubricating oil	1,000	12 (3.7)

Since the TSP pump is selected because of the high viscosity of the fluid, it should be located as close as possible to the wellhead to reduce losses.

3- Estimate the pump efficiency.
4- Estimate the flow rate and slip based on the vendor catalogue:

$$S = k\left(\sqrt{\frac{\Delta P}{Viscosity}}\right) \tag{6.4}$$

Where, K is pump characteristic (from manufacturer: pitch= KD, see Fig. 43). S and ΔP are slip and pump differential pressure respectively.

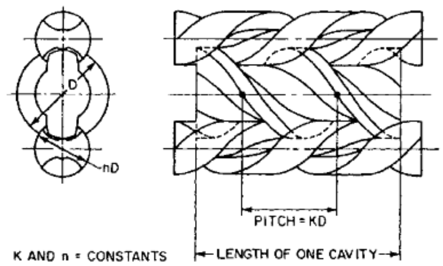

Figure 41: Screw Thread Proportion (Karassik, 2008)

5- Estimate the power using the following equations for SI or US units (Karassik, 2008):

$$Power_{hp} = \frac{Q.\Delta P}{1714} \tag{6.5}$$

$$Power_{kw} = \frac{Q.\Delta P}{36} \tag{6.6}$$

The power is independent of the viscosity.

6- The power imported to the liquid, called "Actual Power Output" can be simply computed by replacing Q_t with Q in the above formula. The value will always be less. The efficiency of the pump will be the ratio of power output to the Brake Horse Power.

7- Estimate minimum required sealing liquid to cool the screws as four to six percent of the total inlet volume flow rate. To ensure enough liquid recirculation in high GVF, a recycle chamber (tank) should be considered.

From this step to the end, the procedure is again the same for HAP and TSP:

8- Specify balancing grades as per ISO 1940:

ISO 1940, grade 6.3 for low speed non critical parts

ISO 1940, Grade 2.5 for high speed parts

ISO 1940, Grade 1 for very specific high speed parts

Grade 1 is not recommended, because of applicability of measuring a very small load accurately.

9- Specify coupling requirements and applicable standard. Since the uninterrupted operation of the pump is between 3-5 years, API 671 is the reference standard for that.

10- Specify the seal plan requirements. Although the sealing system is designed and selected based on API 682, it would not be exactly the same as that on an onshore pump, because for example it is not possible to use production fluid or sea water to control the seal temperature. The barrier fluid is supplied from the topside to the mechanical seal with required modifications for subsea applications.

11- Estimate the weight and size of the equipment. The pump module weight is preferred to be less than 80 tons, because it leads to reasonable handling and installation costs.

CHAPTER 7

DISCUSSION and CONCLUSIONS

Although the basis of a single or multiphase pump in subsea applications is similar to that of onshore and topside pumps, the subsea environment and high Capex and Opex are the factors that make it different. Subsea installation requires a high level of availability and reliability because the pump is intended to work non-stop for more than three years and any unplanned intervention has a significant cost impact. In this condition, a practical procedure that covers all applicable parameters and prioritizes them to optimize pump sizing and selection would be a contribution to the industry.

The available standards and recommended practices, such as API 610 and API 676, cover onshore and topside applications and the reference documents for subsea pump sizing and selection are limited to project reports (from the operators) and some manufacturers' papers. Since the operators are inherently more conservative than manufacturers, their records show a big gap compared to the manufacturers' latest capabilities and claims. An independent study that considers the operators' concerns and manufactures' latest achievements can work as a bridge between the two sides. Almost all available papers in this field give information regarding the influencing factors on pump sizing; however, a step by step procedure including all required calculations hasn't been offered yet.

In this research, all applicable subsea pumps including HAP, TSP and ESP were compared, and their advantages and disadvantages were discussed. The auxiliary equipment of each one such as recycle tank, conditioning tank, liquid management line and bypass line were explained. The characteristic curves and applicable operating ranges of each pump were analyzed.

The parameters that affect pump sizing and selection, including GVF, WOR, viscosity, differential pressure and water depth were listed and explained. The effect of each parameter and how it affects the pump operation were discussed. Some parameters such as GVF are addressed in different papers, and even operating ranges are offered just based on GVF, but it is not possible to make a correct pump sizing without taking into consideration other factors, such as viscosity, which were considered in this research.

Since one of the main issues in subsea applications is corrosion, this issue has been studied thoroughly. Sea water and production fluids contain corrosive agents such as acids, H_2S, CO_2 and hydrogen ions. In the presence of water and tensile stress these components can cause almost all types of corrosion including Uniform Corrosion, Hydrogen Damage, Environmental Induced Cracking, Galvanic Corrosion, Crevice and Pitting, Erosion, Intergranular Corrosion and Microbial Induced Corrosion. All types of applicable corrosion, corrosive agents, related effects and reactions were discussed. Mitigation methods were suggested for reducing the corrosion rate. Based on the corrosion study, the material selection process has been outlined. Super duplex and Nickel alloy materials were able to satisfy selection requirements. They are well proven materials for this application in the industry.

Chapter 6 explains the steps for subsea pump sizing and selection. It starts with some criteria that can help with selecting the main pump type; whether it is hydrodynamic or positive displacement type. These main criteria are capacity, differential pressure and viscosity as governing factors. Then for each type, specific sequences and calculations were mentioned. Applicable sections, calculations and requirements of API 610, API 17A, DNV A-203, ISO 1940 and API 676 standards have also been considered.

The procedure starts with the study of flow regime in order to specify if it is segregated, intermittent, distributed or transient. Then density and Reynolds' number for multiphase flow were calculated. Based these calculations, the gravitational and friction losses in the system can be calculated. The differential pressure of the pump can lead to calculating the BHP and shaft power. The other steps are dedicated to the specific pump type. Estimations for efficiency and installed power were offered for HAP. The calculations are more dependent on the vendor data for TSP; however, the calculations that can lead to proper sizing such as axial velocity, efficiency, slip and power were discussed. Balancing and coupling standards that are applicable to both types were explained at the end.

It should be noted that this procedure tries to help the pump user to size and select a pump for specific service. All the calculations and estimations should be negotiated with the pump manufacturer in the technical BID evaluation stage of the project.

The pump sizing and selection is a complicated process that must be conducted considering all technical aspects related to involved engineering disciplines including process, piping, electrical, instrumentation, metallurgy and safety as well as an approved manufacturer and third parties who test the equipment. It is outside of scope of this research to take the legal responsibility of any mal-function or failure of the equipment that may be selected or sized based on the present study.

7.2 Future Work

Subsea boosting is a new area of research and different aspects can be selected. Some of them are listed here:

1- To prepare a computer program that can consider all parameters and offer the applicable pump size and type.

2- To study the sealing system to reduce the risk of leakage

3- To study power transmission and control system

4- To do research in the reliability and availability of equipment for long term operation

5- To write a standard procedure or document that almost all operators, manufacturers and engineering companies can agree on. It can reduce several conflicts and provide a reference for all involved parties in a subsea processing project.

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