Proposal for new oil sampling pumps at Sture Terminal.

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Preface

During the spring of 2020, Equinor experienced operational issues with the sampling pumps mounted in the sampler cabinets for Grane and Oseberg located at the Sture terminal.

Equinor explained that the issue was due to cavitation and wax pollination and that they were actively seeking a solution. After a series of meetings, we - Rune Alcott, Daniel Egenes and David Tuntland, as graduating students of Mechanical Engineering, were invited to participate in this project as an implementation of our bachelor thesis.

The project was completed a year later in the spring of 2021, at the department of Mechanical and Marine Engineering at the Western University of Applied Sciences (WNUAS), campus Kronstad.

The experience has been both exciting and educationally rewarding. We have gained an overview and greater understanding of oil and gas piping systems, typically used in processing plants, and how crude oil is treated after it has arrived at the plant.

We would like to take this opportunity to thank our supervisor; Professor Boris Balakin, who has provided guidance and expertise every step of way. We would also like to thank Equinor, with a special thanks to our contact person; Tommy Blikberg and Senior Engineer; Iris Renate Tøkje Krokvik.

Abstract

The Sture terminal receives crude oil from both the Oseberg and Grane fields in the North Sea. When the oil arrives at the terminal, it travels through several hundred meters of pipes before the oil is treated and further distributed in the terminal. To be able to withdraw an oil sample, a sampling cabinet is installed for each system. Equinor are experiencing issues with the pumps installed in these cabinets.

Both pumps serve the same purpose; to extract oil samples from the main supply line, and to redirect the sample to a retrieval point. Here, the samples are withdrawn and taken to a laboratory where tests and analysis are performed to verify the quality and properties of the oil.

Both pumps experience circulation issues, but for different reasons. The "sampling pump" at Oseberg experiences "cavitation", which can give rise to mechanical damage or "pitting" of the impeller.

The pump mounted at Grane struggles to circulate the oil, most likely due to a high concentration of wax particles in the oil mixture.

The objective of this thesis is to correctly identify and provide evidence of the causes of these problems culminating in proposals for alternative pump models, which will better suit the criteria and parameters of the pump's functions and purposes.

To determine when and why the Oseberg pump cavitates, $NPSH_A$ calculations were made. The Roscoe-Brinkman equation was applied when calculating the wax percentage in the oil mixture at Grane.

Conclusion – Oseberg: The pump experiences cavitation and needs to be replaced with suitable alternative.

Conclusion – Grane: The volute channel becomes clogged with wax particles, consequently preventing the pump to circulate oil and eventually overheating. The pump needs replacing with a positive displacement pump, such as a screw pump or a vane pump.

Sammendrag

Stureterminalen mottar råolje fra både Oseberg- og Grane-feltene i Nordsjøen. Når oljen ankommer terminalen, beveger den seg gjennom flere hundre meter med rør før oljen blir behandlet og videre distribuert i terminalen. For å kunne ta et prøveuttak av oljen, er det installert uttakskabinett til hvert system. Equinor opplever problemer med pumpene som er installert i disse kabinettene.

Begge pumpene har samme formål; å hente et prøveuttak fra hovedlinjen, og videresende uttaket til et mottakspunkt. Her blir uttaket hentet og transportert til et laboratorium hvor det skal utføres tester og analyser.

Pumpene opplever sirkulasjonsproblemer, men av ulik årsak. «Sampler pumpen» ved Oseberg opplever «kavitasjon», som kan gi opphav til mekaniske skader, eller «pitting» på impelleren.

Pumpen som er installert ved Granelinjen sliter å sirkulere oljen, antageligvis grunnet høy konsentrasjon av vokspartikler i oljeblandingen.

Målet med denne oppgaven var å korrekt identifisere, samt fastslå årsakene til sirkulasjonsproblemene og til slutt, foreslå alternative pumpemodeller, som vil bedre passe formålet til pumpen.

For å avgjør når og hvorfor pumpen ved Oseberg kaviterer, ble det utført en $NPSH_A$ analyse. Roscoe-Brinkman formelen ble brukt for å regne ut vokskonsentrasjonen i oljeblandingen.

Konklusjon – Oseberg: Pumpen kaviterer. Utregningen foreslår at det er nødvendig å erstatte pumpen med et alternativ som bedre passer formålet.

Konklusjon – Grane: Pumpespalten tetter seg grunnet vokspartikler. Når dette skjer, sliter pumpen å sirkulere oljen, som følgelig resulterer med varmegang i pumpen. Voksanalysen foreslår at det er nødvendig å erstatte pumpen med en fortregningspumpe, for eksempel en skrupumpe eller en vingepumpe.

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Nomenclature

ρ	=	density [kg/m^3]
p	=	pressure [kg/m^2]
Q	=	flow [m^3/h]
V	=	velocity [m/s]
Re	=	Reynolds number
λ	=	Darcy friction factor
A	=	area [m^2]
ΔT	=	change in temperature [°C]
v	=	kinematic viscosity [m^2/s]
μ	=	dynamic viscosity [Pa·s]
l	=	length [m]
d	=	diameter [m]
ξ	=	local hydraulic resistance
φ	=	concentration [%]
H_a	=	The absolute pressure [bar]
H_s	=	The static head of the liquid over the pump [bar]
H_{vp}	=	The vapour pressure [bar]
H_f	=	The total friction loss for the suction side of the system [bar]
ϵ	=	Relative pipe roughness

1 Introduction

1.1 Our partners

With over 21,000 employees, Equinor are Norway's largest operator, one of the world's largest offshore operators and an ever-growing force in renewable energy.

Equinor complies with the Paris agreement - a net zero target for society and a low-carbon future. These are core values that the team share and are proud and grateful for the opportunity to participate in collaboration with Equinor.

In addition to working towards a greener future, Equinor are responsible for around 70 percent of the country's total oil and gas production - equivalent to around 2 million barrels of oil every day. (Equinor, 2021) - providing a significant contribution to the GDP of the country.

1.2 Sture Terminal

The Sture terminal is located northwest of Bergen, in the municipality of Øygarden in Vestland. The terminal opened in 1988 and was operated by Norsk Hydro Produksjon until 2007. The terminal receives crude oil and gas from both the Oseberg and Grane fields. The crude oil travels hundreds of kilometres through pipelines embedded on the sea floor, before arriving at the terminal and being temporarily stored in storage tanks in underground caverns, before being treated and further distributed throughout Europe. (Equinor, 2021)

Before the crude oil can continue its journey, several analyses and tests must be performed to ensure the quality of the oil. This is done by retrieving an oil sample from the main pipeline using a pump mounted in a sampling cabinet. Both the Oseberg and Grane pipeline systems has their own dedicated cabinet, where the samples are extracted and analysed daily.

Figure 1.1 - The Sture Terminal Photo: (Equinor, 2021)

1.3 Problem Statement

Both the Oseberg and Grane crude oil lines are experiencing circulation issues with the pumps mounted in the sampling cabinets.

Equinor have made an invitation to carry out research into the operation of the pumps to determine the cause of the problems in order to support investment in pump replacement or to apply technical modifications.

1.3.1 Oseberg

The first issue to be assessed is common to centrifugal pumps in general, and for the Oseberg sampling pump located at Sture.

The problem occurs when the pressure on the suction side of the sampling pump is too low, resulting in "flashing" - a phenomenon that occurs when the pressure drops below the vapour pressure of the fluid. The sudden drop in pressure, from the main pipeline to the suction side of the pump, causes the state of the fluid to alter from its liquid- to its gas phase, forming vapour bubbles and occupying a greater space in the fluid. When the bubbles pass the eye of the impeller, the pressure drastically increases, causing the bubbles to "implode". The implosion produces a supersonic liquid microjet. A truly undesirable condition known as "cavitation", producing extensive erosion of the rotating blades, additional noise from the resultant knocking and vibrations, as well as a significant reduction of efficiency because it distorts the flow pattern. (Tikkanen, 2010)

This phenomenon at Oseberg occurs especially when the oil temperature is around 50°C and higher, with a significant temperature increase of approximately 15°C being transmitted across the pump. Equinor have determined that the Oseberg pump in question is chosen improperly and has a nominal flow of 23 m^3/h and a minimum flow of 2,5 m^3/h. After they started to experience problems, the pumps have been operated as low as 0,75 m^3/h, the outlet-pressure is approximately 3 barg. This is not optimal, but necessary to prevent cavitation. According to the pump's datasheets, the flow should be 2 m^3/h.

1.3.2 Grane

The second issue is for the Grane sampling pump. The pump is overheating, likely because of wax particles in the oil. These particles are also solidifying on the impeller and the pipe walls. The current pump which is installed in the sampling cabinet is not suitable for oil contaminated with wax particles and struggles to circulate the oil, eventually overheats and stopes delivering flow all together.

When the crude oil is extracted from reservoirs, it emerges as a mixture, not pure oil. The mixture contains water, paraffin, in addition to several high viscosity substances. Due to the temperature- and pressure drop between the reservoir and the undersea pipes, the paraffin in the untreated crude oil solidifies on the pipe walls, when transported through the two-hundred-kilometre pipeline. Eventually the paraffin builds up a solid layer, reducing the inner diameter, consequently reducing the oil flow. When this happens, Equinor are forced to scrape the inside on the pipes by utilizing a pipeline inspection gauge (PIG). After the PIG arrives at the terminal, the excessive wax enters the pipeline system. Equinor invite research to calculate the wax concentration to use the resultant information to propose an alternative pump that better suits the purpose.

2 Method

This section introduces the theory, procedures and equations needed to solve the various problem related to the pumps.

2.1 Fluid flow

The flow in pipes is divided into two types of flows: laminar and turbulent. In this project both laminar and turbulent fluid flow occur in the pipelines. Laminar flow is characterized by having streamlines in parallel layers and often occurs at relatively low velocities, unlike turbulent flow, where the streamline of the flow behaves more erratic.

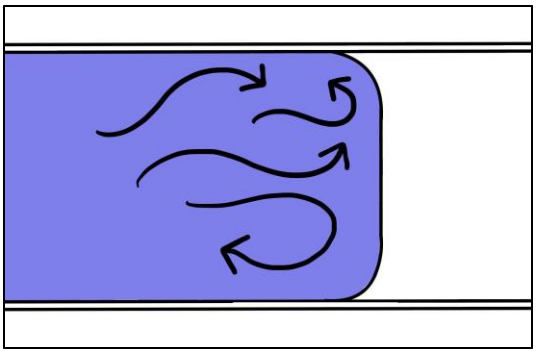


Figure 2.1 – Turbulent Flow

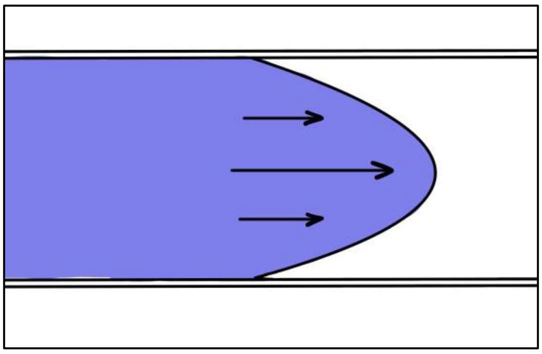


Figure 2.2 – Laminar Flow

The type of flow is determined by the Reynolds number, which is a correlation between velocity, viscosity, and the pipe diameter:

$$Re = \frac{V \cdot d}{v}$$

Equation 2.1 – Reynolds Number

Where V is the mean velocity of the flow [m/s], v is the kinematic viscosity of the fluid [m^2/s] and d is the inner pipe diameter [m].

If the Reynold number >> 2000, the flow becomes turbulent, however, when Reynolds number is between 2000-3000 (Moody Diagram) the flow is classified as “transitional flow”. Transitional flow is irregular, creating turbulence in the flow centre/axis and laminar flow at the inner pipe walls. – which gives rise to inconsistencies in applying formulae.

The mean velocity in the pipe can be determined by the relation between the flow $\left[\frac{m^3}{hr}\right]$ and cross section area A $[m^2]$ of the pipe.

$$Q = A \cdot V \rightarrow V = \frac{Q}{A}$$

Equation 2.2 - Velocity

2.2 Friction

When working with all types of hydraulic systems it is necessary to include the fluid resistance, or friction in pipes and fittings to achieve correct calculations. Friction in pipes occurs due to the effect of the viscosity of the fluid has against the inner pipe walls. Pressure loss in pipelines is often divided into three categories:

1. friction in straight pipes,
2. pressure loss due to pipe bends
3. pressure loss in fittings (valves, couplings etc.).

2.2.1 Friction in Straight Pipe

Friction in straight pipes is often the greatest friction loss in a system. The Darcy-Weisbach (2.3) Equation consider the friction losses in straight pipes i.e.:

> The frictional losses are due to the viscosity (molecular and of the fluids), which manifests itself during their motion and is a result of the exchange of momentum between molecules at laminar flow and between individual particles of adjacent fluid layers moving at different velocities, at turbulent flow. These losses take place along the entire length of the pipe. (Idel'chik, 1966, s. 29). The Darcy-Weisbach equation is presented below

$$\Delta p_{fr} = \frac{1}{2}\lambda\frac{l}{d_h}\rho V^2$$

Equation 2.3 - Darcy-Weisbach

Where d_h is the hydraulic diameter [m], l is the length of the pipe [m], ρ is the density of the fluid [kg/m^3], V is the velocity [m/s] and λ is the Darcy friction factor. Hydraulic diameter is defined as the "wetted diameter" in a pipe. For this project $d_h = d$.

The friction factor depends on the characteristics of the fluid and the roughness of the pipe. For a turbulent flow, the friction factor can be calculated using the following equation:

$$\lambda = \frac{0{,}316}{Re^{0{,}25}}$$

Equation 2.4 – Friction Factor for Turbulent flow

However, this equation does not take account for the roughness of the pipe. Therefore, the accuracy of this equation may decrease as the relative roughness of the pipe (ϵ) increases. Accordingly, it is more accurate to use "Moody Diagram" (see Attachment 6.3.1) when obtaining the friction factor, as this diagram includes the relative roughness of the pipe

Based on the Moody Diagram, it can be observed how the roughness of the pipe may impact the friction factor as the roughness increases. It is also noticeable from the diagram that the pipe roughness becomes irrelevant when working with laminar flows. It is therefore necessary to use Equation 2.5 independently of the pipe roughness for laminar flows.

$$\lambda = \frac{64}{Re}$$ Equation 2.5 - Friction Factor for Laminar Flow

2.2.2 Pressure Loss in Elbows

When the flow direction is altered in curved conduits such as pipe bends, an increase in pressure at the outer wall and a decrease at the inner wall will occur. These zones are referred to as “eddy zones”.

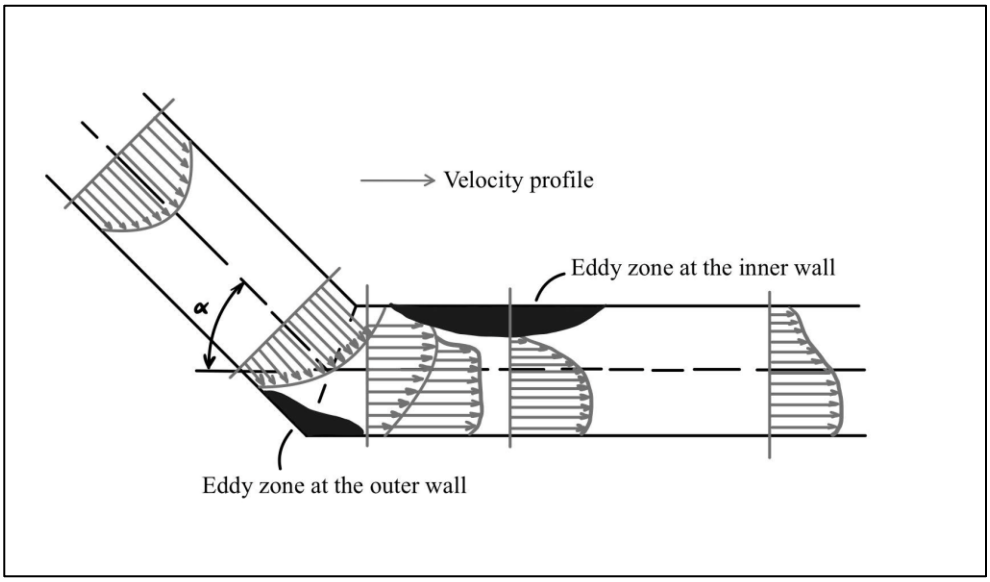


Figure 2.3 – Eddy Zones (Reproduced from (Idel'chik, 1966, s. 190))

The flow in the eddy zones circulate, causing the zones to act as a reduction in the stream section, which leads to pressure loss. The magnitude of this loss in pressure is correlated to the angle and bend radius. Pressure loss in pipe elbows can be found using Equation 2.6.

$$\Delta p_{loc} = \frac{1}{2}\xi\rho V^2$$ Equation 2.6 – Local Losses

Where ($\xi = \xi_{\mathrm{bend}}$) is the coefficient of local resistance in the pipe bend and is based on the extent of the flow disturbance caused by the bends. There are several methods for finding this resistance coefficient, where the shape, angle and the radius of the pipe bend plays a large factor in calculating the coefficient. The coefficient can be obtained by using a diagram for pipe bends at 90°.

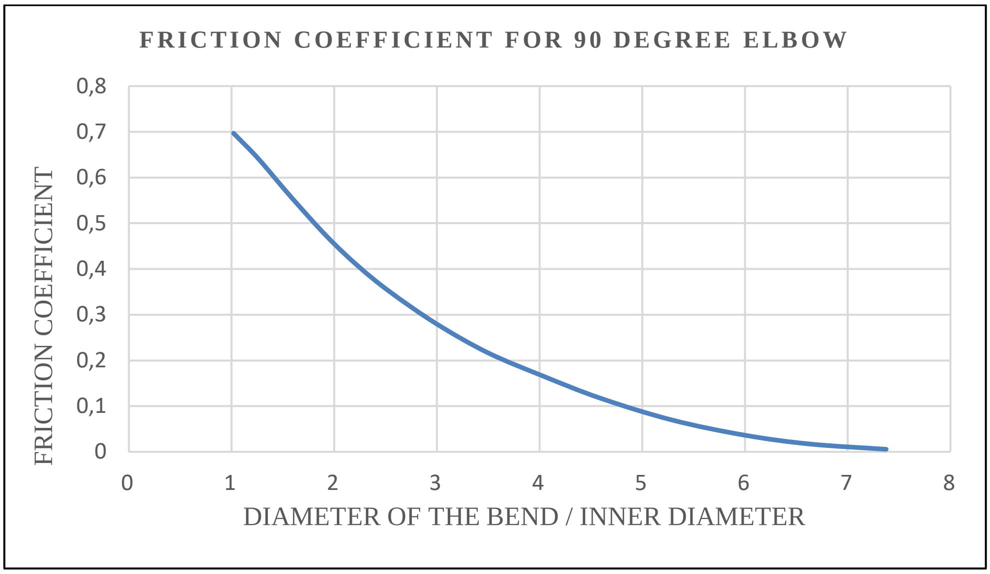


Figure 2.4 - Friction Coefficient (Reproduced from (Brautaset, Innføring i oljehydraulikk, 1987, s. 64)

Since the coefficient is highly dependent on the shape and angle of the pipe, sharp bends will cause more friction loss than smoother bends.

2.2.3 Pressure Loss in Valves

There are often multiple components in a hydraulic system that can cause friction loss, such as valves and couplings. Different valves cause different amount of friction loss, depending on the type and shape of the valve. For example, globe valves cause a lot more friction loss than ball valves. The amount of friction loss from a fluid moving through a valve is dependent on the friction factor $\xi = \xi_{valve}$ of the valve and can be calculated using Equation 2.6. The friction factor for many valves can be obtained from Handbook of Hydraulic Resistance (Idel'chik, 1966). Equinor uses predominantly ball valves and gate valves in the pipelines relevant to this project. These types of valves cause very low friction losses when opened to full extent.

2.3 Viscosity

Viscosity is defined as a fluid's resistance to flow. In fluids, an increase in the viscosity value is resultant in an increase in the resistance forces between each liquid particle. This is called the fluids inner friction. Newtonian liquids such as water, alcohol and different types of oil only vary their viscosity when there are changes in temperature or pressure. (Brautaset, Innføring i oljehydraulikk, 1987, s. 25)

Viscosity can be in two forms: dynamic or kinematic.

The dynamic (also known as absolute) viscosity μ is defined as the ratio between the shear stress and the velocity gradient. It is different from the kinematic viscosity v which is defined as the ratio between dynamic viscosity and its density. (Brautaset, Innføring i oljehydraulikk, 1987, s. 28)

For typical mineral oils it is possible to find their viscosity by using the graph: Figure 2.5 which illustrates the relationship between the kinematic viscosity and the temperature of crude oil.

The function is logarithmic, but the coordinate system is linear. There is a recreated version in (Brautaset, Innføring i oljehydraulikk, 1987, s. 31), where the coordinate system has a logarithmic scale, and the function is linear. By utilizing the recreated version, the graph can easily be modified to obtain the viscosity for various mineral oils such as crude oil. A new linear function can be inserted into this graph by inserting two arbitrary viscosities at different temperatures. This new linear graph can be used to estimate the viscosity at most temperatures. (Brautaset, Innføring i oljehydraulikk, 1987, s. 31).

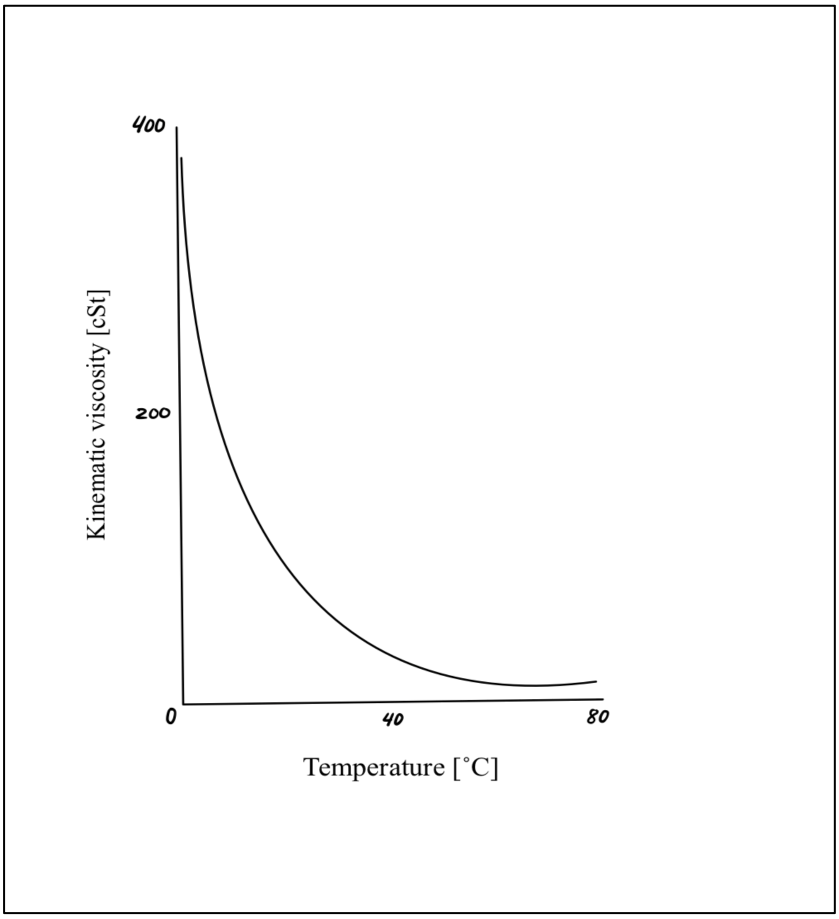


Figure 2.5 - Change in kinematic viscosity with the temperature of a typical mineral oil (Reproduced from (Brautaset, Innføring i oljehydraulikk, 1987, s. 30))

The viscosity of crude oil varies, not only as a factor of temperature and pressure, but also if the composition changes. An example of composition change is the percentage of wax in the oil. Because wax has a higher viscosity than oil, a higher amount of wax in the oil will increase its viscosity. To calculate the dynamic viscosity of the oil as a factor to the wax percentage, Equation 2.7 can be applied (K.Qin, 2003).

$$\mu_{mixtrue} = \mu_{oil} \cdot (1 - \varphi_{VFP})^{-2.5}$$

Equation 2.7 – Viscosity & wax concentration

2.4 Wax

2.4.1 Wax in pipelines

Wax in the transit of oil is a major consideration in the petroleum industry.

The subsea pipes that carry the oil from the North Sea platforms to the Sture terminal are hundreds of kilometres long. The temperature of the oil during transit is cooled down by the surrounding seawater giving rise to undesirable wax solidification.

Wax precipitates from the oil when the temperature reduces to approximately 30-40°C. When this occurs, the wax solidifies on the pipe walls and reduces the flowrate. (Gudmundsson, 2018)

Various research has been conducted to eliminate or reduce this phenomenon.

One possible solution could be to increase the temperature of the oil by installing heating cables around the pipes, but this is not deemed to be realistic due to exorbitant costs and the impracticalities of execution.

A more realistic and cost-effective solution is to mechanically scrape the walls of the pipe at appropriate time intervals. This technique is achieved by use of a purpose made Pipeline Integrity Gauge (PIG). This operation is accomplished by inserting the PIG in a PIG entry point. This allows the PIG to be launched and received without disturbing the flow. The PIG is forced through the pipeline by the force of the oil flow as illustrated in Figure 2.6.

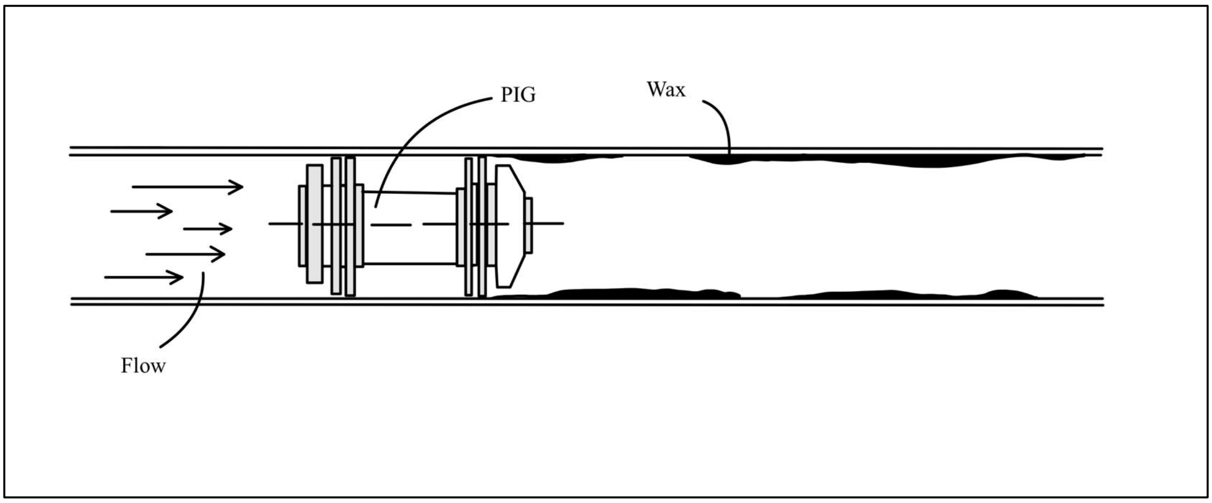


Figure 2.6 - Mechanically scraping of the pipeline walls by using a PIG.

2.4.2 Wax in centrifugal pumps

As with pipelines described above, wax particles are a major problem in centrifugal pumps.

The reason wax is a major problem in centrifugal pumps is due to an increase in viscosity and the introduction of solid wax particles into the fluid. This increase in viscosity can cause the pump to cavitate. The wax particles can also stick to the volute of the pump. This would gradually decrease the volute area and eventually stop the flow completely.

2.5 Net Positive Suction Head

Net Positive Suction Head (NPSH) is a measure of the pressure experienced by a fluid on the suction side of a pump. The standard unit of measurement is usually given as "Head" (in metres) but can also be quoted in actual pressure (Pa). This is a measurement of the pump's ability to lift different fluids to the same vertical height irrespective of density.

This thesis uses actual pressure (Pa) as opposed to metres, while discussing NPSH. This is also the unit of measurement Equinor use in their Superior Technical Information and Document system (STID).
The purpose of NPSH is to prevent flashing and consequently cavitation. To prevent cavitation, it is important to first understand exactly when and how flashing occurs. To calculate this, the Net Positive Suction Head Available ($\mathrm{NPSH_A}$) must be compared to the Net Positive Suction Head Required ($\mathrm{NPSH_R}$). To theoretically prevent cavitation, the $\mathrm{NPSH_A}$ value must be equal to, or greater than, the $\mathrm{NPSH_R}$ value. In reality, a safety margin must be implemented. As an industry standard the $\mathrm{NPSH_A}$ value should be 0,5 to 1 meter (0.05-0.01 bar) greater than the $\mathrm{NPSH_R}$ value.

2.5.1 Net Positive Suction Head Required

The $\mathrm{NPSH_R}$ value is quoted by the pumps manufacturer and provides the operator with an indication of the minimum suction pressure that must be exceeded for the pump to operate without experiencing flashing.

2.5.2 Net Positive Suction Head Available

The Available value differs from Required value as it is not a fixed value, but rather an independent variable determined by four factors:

1. The absolute pressure (H_a)
2. The static head of the liquid over the pump (H_s)
3. The vapour pressure (H_{vp})
4. The total friction loss up to the suction side of the system (H_f)

The equation used to determine the $\mathrm{NPSH_A}$ value is presented below.

$$NPSH_A = H_a + H_s - H_{vp} - H_f$$

Equation 2.8 - Net Positive Suction Head Available

2.5.2.1 Absolute Pressure (H_a)

The absolute pressure is normally utilized when discussing NPSH. The absolute pressure p_{abs} is measured relative to the absolute zero pressure - the pressure that would occur at an absolute vacuum.

Equinor document the system pressure as "gauge pressure", not absolute pressure. The only distinction between the two units of measurements being that gauge pressure reads the value relative to the surrounding pressure. It is possible to convert from absolute pressure to gauge pressure by subtracting the atmospheric value from the absolute pressure value (1 ATM at sea level or approximately 1 Bar or $1 \cdot 10^5\ Pa$).

$$p_g = p_{abs} - p_{atm}$$ Equation 2.9 - Gauge Pressure

Where:

$p_g = gauge\ pressure\ [Pa]$

$p_{abs} = absolute\ pressure\ [Pa]$

$p_{atm} = atmospheric\ pressure\ [Pa]$

The absolute pressure used in the $NPSH_A$ calculation is the pressure within the system.

2.5.2.2 Static Head (H_s)

Static head is the change in elevation between the point where H_a is measured and the pump inlet. The vertical height can be converted to pressure by multiplying the height by the density and gravitational force.

$$H_S = \rho \cdot g \cdot h$$ Equation 2.10 - Hydrostatic Pressure

Where:

$\rho = fluid\ density\ \left[\frac{kg}{m^3}\right]$

$g = gravity\ \left[\frac{m}{s^2}\right]$

$h = vertical\ heigh\ [m]$

The static head will either be positive or negative depending on whether H_a is measured above or below the pump.

2.5.2.3 Vapour Pressure (H_{vp})

"Vapour pressure is a measure of the tendency of a material to change into the gaseous or vapour state, as it increases with temperature. The temperature at which the vapour pressure at the surface of a liquid becomes equal to the pressure exerted by the surroundings is called the boiling point of the liquid." (Encyclopaedia Britannica, 1998)

Figure 2.7 represents a fluid travelling through a pump. The graph illustrates how the pressure drops as the fluid approaches the impeller. If the pressure drops below the vapour pressure, vapour bubbles form and flashing occur. When the bubbles pass through the impeller the pressure increases, and the bubbles implode.

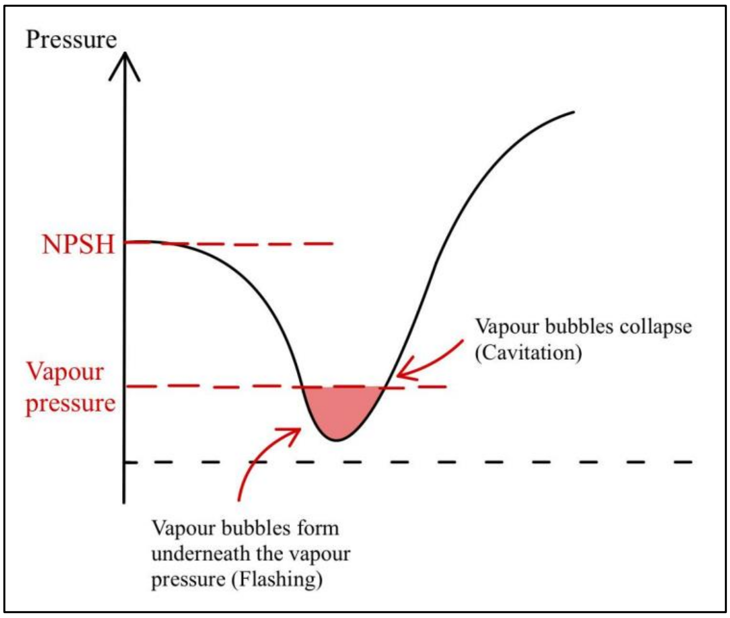


Figure 2.7 - Flashing and Cavitation

If the fluid substance is unknown, it is possible to estimate the pressure-temperature properties of pure substances by studying the Cox Chart. The chart serves as a method for estimating vapor pressure for common petroleum hydrocarbons and is defined as "A straight-line graph of the logarithm of vapor pressure against a special nonuniform temperature scale; vapor pressure-temperature lines for many substances intersect at a common point on the Cox chart" (Parker, 2003).

To further estimate the vapor pressure value for various temperatures, it is possible to use Antoine equation. The Antoine equation is the empirical relationship between temperature and vapor pressure of liquids (Parker, 2003).

$$\log P = B - \frac{A}{C + T}$$

Equation 2.11 – Antoine Equation

Where A, B and C are experimental constants retrieved from a summary of results of correlation of the experimental data and T is temperature.

2.5.2.4 Friction Loss (H_{fr})

There are two different types of fluid losses related to the movement of fluids: the friction losses (Δp_{fr}) and the local losses (Δp_{loc}). They can be expressed by the following equations:

$$\sum \Delta p_{loss} = \sum \Delta p_{fr} + \sum \Delta p_{loc}$$

Equation 2.12 – Pressure Drop

Where Δp_{fr} is the Darcy-Weisbach equation (2.3) and Δp_{loc} is the local losses equation (2.6).

The local losses appear as a result of a disturbance of the normal flow. Equation 2.6 is used to calculate the friction loss due to bends, transitions, valves, inlets, and outlets etc.

The coefficient of friction loss ξ, is based on the extent of the flow disturbance caused by the obstacle.

In certain situations, it is possible to calculate or utilize tables to find the coefficient of friction loss, for instance 90° bends (Figure 2.4), but in most instances, this value is specified by the manufacture of the component or obtained in a handbook of coefficients.

2.5.2.5 Preventing Cavitation Due to Vaporization

There are several ways to prevent cavitation due to vaporization:

1. Lower the temperature of the fluid
2. Raise the level of the storage tank
3. Use a booster pump to feed the principal pump
4. Reduce the revolutions per minutes (RPM) of the motor
5. Increase the diameter of the impeller

If none of the methods above are feasible, the pump needs replacing with a better suited alternative.

3 Results

3.1 Schematics

Sture terminal is a large facility with a complex layout of pipelines and machinery.
This study has invested significant time and resources studying the schematics for the Grane and Oseberg pipelines in order to achieve a thorough understanding of the layout of the terminal.
Equinor use Piping and Instrumentation Diagrams (P&ID) for their schematics. It is important for this project to understand the journey of the fluids from the arrival at the terminal through to the sampling pumps. When working with NPSH and cavitation, it is necessary to decide the starting point (where H_a is measured) of the fluid, as this is important when calculating $NPSH_A$. From experience, this starting point is usually set at a tank or reservoir where the fluid is stationary. In this case, there is no tank or reservoir to use as a starting point that would benefit further calculations. At both the Grane and Oseberg pipelines, the starting point has been set to the "main pipe", *0600-PL-21-16002-CA2-00-A* at the Grane pipeline and *0600-PL-20-30836-CA2-06-A* at the Oseberg pipeline (see Attachments 6.1.1 and 6.2.1). These pipes are the nearest "main pipes" to the sampler cabinets (see Attachments 6.1 and 6.2 for more information about the pipelines).

Another important piece of data to obtain from the schematics is the total length and height from H_a to the pump inlet. The length is important for friction calculations and the height is important when calculating the static head loss. The length and height of every pipeline at Sture terminal can be found on STID. Using Attachment 6.1 and 6.2, the length from H_a to the pump inlet at the Oseberg pipeline is 62.32 m and 3.53 m at the Grane pipeline. The height at the Oseberg pipeline is 3.67 m and 0,39 m at the Grane pipeline. These values will be useful in further calculations. The same schematics are used to locate pipe bends and valves in the pipelines that have relevance in further calculations.

3.2 Oseberg (Results)

To calculate the NPSH value, it is first necessary to determine the viscosity of the fluid. The viscosity is an important variable when determining Reynolds number, which also affects the friction factor.

3.2.1 Viscosity

The kinematic viscosity of the crude oil at 20°C and 40°C is found using data provided by Equinor and illustrated in attachment 6.5. It is important to determine the viscosity at 50°C, as this is the temperature of the oil at the inlet of the pump. To obtain this information, Figure 2.5 (on page 31 in 'Innføing I oljehydraulikk' can be used (see Section 2.3)). This figure shows the kinematic viscosity as a factor of temperature. By plotting in the viscosity values supplied by Equinor (at 20 and 40°C), it is possible to draw a new linear graph between the points and further utilize this graph to find the viscosity at various temperatures. The modified graph indicates that the viscosity of the oil at 50℃ is equal to 2,7 cSt. (Brautaset, Innføring i oljehydraulikk, 1987, s. 31)

The viscosity also varies as the wax percentage varies. Equation 2.7 is used to calculate the dynamic viscosity of the oil mixture at different concentrations of wax particles. The wax percentage (φ_{VPF}) is set to be between 0-60%. The viscosity of the oil (0% wax particles) is set to 2,7 cSt as estimated above. The results of the calculations are shown in Figure 3.1, The percentage of wax particles in the oil mixture is unknown and therefore not further considered in the calculation.

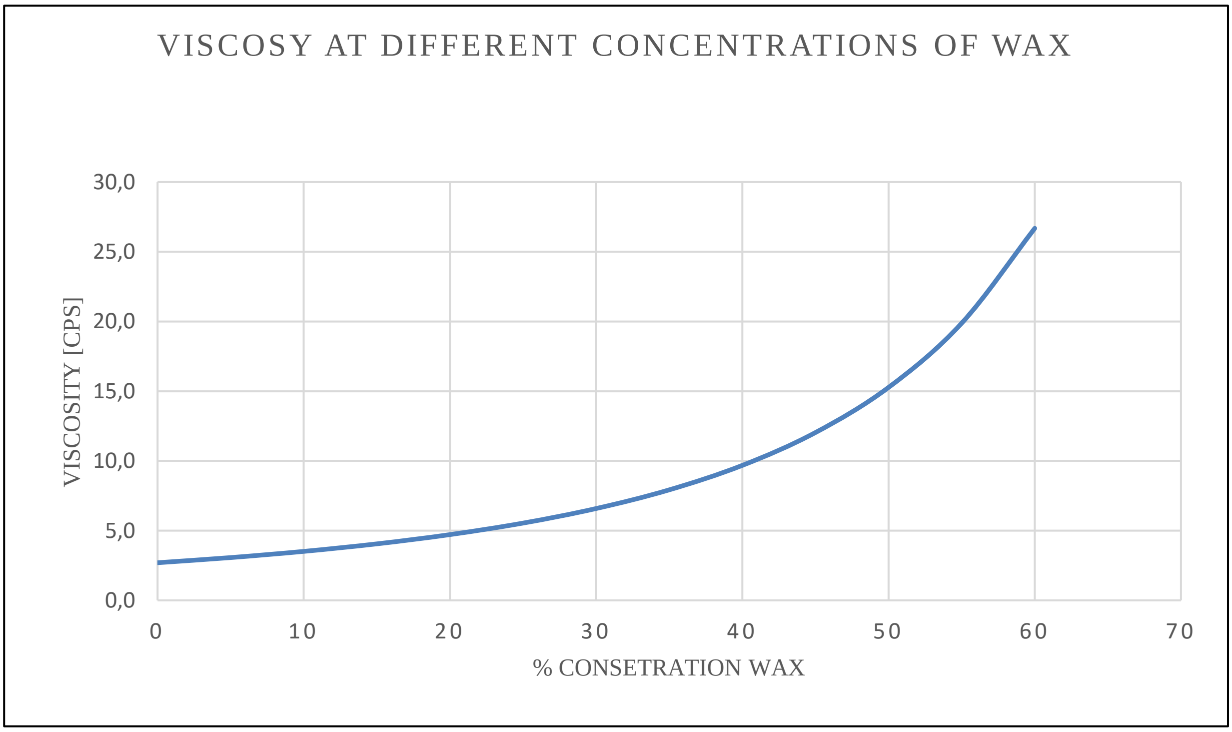


Figure 3.1 - Viscosity with Wax Particles (Oseberg)

3.2.2 Current Pump on the Oseberg Line

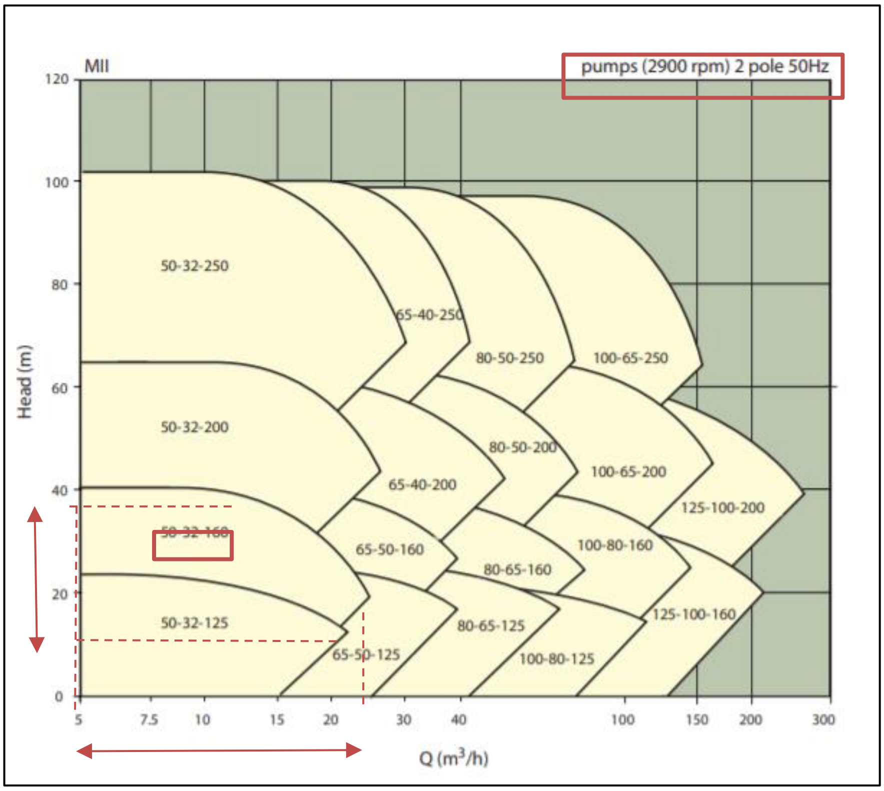
The pump currently installed in the sampling cabinet connected to the Oseberg line is a centrifugal pump and is manufactured by Verdermag Global. Extracted from STID, the relevant specifications are presented in Table 3.1

Table 3.1 - Pump Specifications (VerderMag Global)

Manufacturer	Verdermag Global
Series	MII
Model	50-32-160
Distributor	TuroTeknikk
RPM	2900
$NPSH_R$ [Bar]	1.28
Minimum Flow $\left[\frac{m^3}{hr}\right]$	2.5
Dimensions in mm (not including motor) (VergerLiquids)	190 x 407.5 x 252

In the problem statement, Equinor express concern regarding the dimensioning of the current pump. It is believed that the pump is over dimensioned for its purpose. By obtaining the datasheet provided by the manufacturer it is possible to determine the recommended head at different RPMs. This information is presented in Table 3.2.

Table 3.2 – Head/Flow Diagram (VerderMag Global)



The diagram suggests that the 50-32-160 model, while operated at 2900 RPM should have a flow rate between 5 and 25 m^3/h, resulting in a head between 14 and 40 metres . Thus, confirming that the pump is over dimensioned. As a temporary measure to prevent cavitation, Equinor have reduced the flow rate to 0,75 m^3/h. This is not a long-term alternative as operating the pump below the minimum flow rate may cause suction recirculation, thermal instability or discharge recirculation that may result in vibration or inefficiency.

Of the four factors that form the NPSH equation, the surface pressure is simplest obtained. In this study no calculation is needed, as Equinor state the pressure in the main line is 1,25 bar directly after the quill. The pressure loss over the quill is stated by Equinor to be 0,2 bar.

3.2.3 Net Positive Suction Head Results (Oseberg)

The following section presents the $\mathrm{NPSH_A}$ calculations and compares the result against the $\mathrm{NSPH_R}$value.

3.2.3.1 Calculating the Static Head

To calculate the static head, the vertical distance between the pump and the main supply line is required. This value is obtained by studying the schematics of the system.

As stated in section 3.1, the vertical length of the relevant pipes is equal to 3,759 metres and the density of the oil is stated as 827 $\frac{\mathrm{kg}}{\mathrm{m}^3}$ in Attachment 6.5.

$$H_s = \rho \cdot h \cdot g$$

Equation 3.1 – Hydrostatic Pressure

$$H_s = 827 \cdot 3.759 \cdot 9.81 = 30496\ Pa$$

$$H_s = 0.305\ bar$$

3.2.3.2 Vapor Pressure – Cox Chart & Antione Equation

The available vapor pressure is an approximation, as the value Equinor state in their Crude Summary Report in attachment 6.5 is the raid vapor pressure (RVP), which is always given at 37,8°C, *not* 50°C. This is related to samples taken during ship loading.

The vapor pressure of liquids increase when the temperature rises. This increase in pressure can often be substantial, so it is crucial to obtain the correct vapor pressure value. If the value is unavailable, it is possible to obtain a calculated estimate. If the molecules of the mixture, as well as the vapor pressure at an arbitrary temperature are known, one possible method to achieve this is by using the Cox-Chart.

In the same report where the vapor pressure is stated, Equinor also list the molecules in the oil mixture. The molecules are listed in Table 3.4.

Table 3.3 - Molecules in oil mixture

Molecules (% wt on crude)	
Methane + Ethane	0,02
Propane	0,61
Isopentane	0,38
n-Butane	1,18
Isopentane	1,03
n-Pentane	1,54
Cyclopentane	0,17
C_6 paraffins	2,92
C_6 naphthene	1,93
Benzene	0,48
C_7 Paraffins	2,80
C_7 Naphthene	2,90
Toluene	1,17

By studying the Cox-Chart, it can be observed that $C_6 - paraffin$ has the nearest vapour pressure value, compared to the oil mixture (8 psi, when the temperature is 100°F/0,55 bar when the temperature is 37,8°C)

By plotting in the specific constants for C_6, it is possible to calculate an estimated vapor pressure value.

$$\log_{10} p = A - \frac{B}{C + T}$$

Equation 3.3.2 - Antoine Equation

According to "Summary of the results of the correlation of the experimental data with Antoine equation for vapor pressure" published by Charles B. Willingham in "Research of the National Bureau of Standards" (Willingham, 1945), the hydrocarbon with the nearest vapor pressure value, to the oil mixture is 2.3 Dimethylbutan $- \mathrm{C_6H_{14}}$. The component-specific constants are presented in Table 3.5.

Table 3.4 - Experimental Constants of Hydrocarbons (Willingham, 1945, s. 239)

Compound	Formula	A	B	C
2.3-Dimethylbutan	C_6H_{14}	6,80983	1127,187	228,900

$$\log_{10} p^{sat} = A - \frac{B}{C+T}$$

$$p^{sat} = 10^{\left(A - \frac{B}{C+T}\right)}$$

$$p^{sat}_{37,8°C} = 10^{\left(6.80983 - \frac{1127.187}{228.900+37.8°C}\right)} = 383.2\, mmHg \approx \underline{0.51\, bar}$$

The temperature at the inlet of the pump is 50°C. In the problem statement it is reported that the rise across the pump is 15°C. By dividing the rise in temperature by two and adding this value to the inlet temperature, it is estimated that the average temperature at the eye on the pump is 57°C.

$$p^{sat}_{57°C} = 10^{\left(6.80983 - \frac{1127.187}{228.900+57°C}\right)} = 735.06\, mmHg \approx \underline{0.98\, bar}$$

Although to some extent inaccurate, the estimated vapor pressure is set to 0.98 bar at 57°C.

3.2.3.3 Calculating the Friction Loss

The first step to calculating the pressure drop due to friction, is solving the Darcy-Weisbach Equation (2.3). The friction loss is calculated in four pipes with an inner diameter of 50mm, 25mm, and two pipes with a diameter of 40mm. To calculate the friction loss in these pipes, the velocity is required. Equinor state a flow rate of 0,75 m^3/h in these pipes. By utilizing Equation 2.2, it is possible of calculate the velocity of the fluid.

$$\sum \Delta p_{fr} = \frac{1}{2}\lambda \frac{l}{d_h}\rho V^2$$

Where:

$$\lambda = \frac{64}{Re}\ (Laminar)\ or\ \lambda = \frac{0.316}{Re^{0.25}}\ (Turbulent)$$

The friction factor equation is better suited for laminar flows. If the Reynolds number indicates that the flow is turbulent, a better option for calculating the friction factor is the "Moody Diagram", as this diagram also considers the roughness of the pipe. (Attachment 6.3.1)

$$Re = \frac{V \cdot d}{v}$$

$$Re_{25mm} = \frac{0{,}424 \cdot 25 \cdot 10^{-3}}{2{,}7 \cdot 10^{-6}} = 3925.93$$

$$3925.926 \gg 2300 \rightarrow Turbulent$$

By studying the Moody Diagram (assuming the pipe is smooth structural steel), the friction factor for the 25mm pipe, prior to the pump is equal to 0,04.
(Compared to calculating the friction factor (Equation 2.4): $\frac{0.316}{3925.93^{0.25}} = 0.0399$)

It is now possible to apply the Darcy-Weisbach Equation (2.3) to the 25mm pipe.

$$\Delta p_{fr_{25mm}} = \frac{1}{2} \cdot 0.04 \cdot \frac{4.932}{25 \cdot 10^{-3}} \cdot 827 \cdot (0.424)^2$$

$$\Delta p_{fr_{25mm}} = 586.45\ Pa$$

The Reynolds number for the 40mm and the 50mm pipes:

$$Re_{40mm} = \frac{0.166 \cdot 40 \cdot 10^{-3}}{2.7 \cdot 10^{-6}} = 2457.17 \rightarrow Transitional\ flow$$

$$Re_{50mm} = \frac{0.106 \cdot 50 \cdot 10^{-3}}{2.7 \cdot 10^{-6}} = 1964.94 \rightarrow Laminar\ flow$$

According to the Moody Diagram, the friction factor for the 40mm pipe is 0.05.

For the 50mm pipes, the friction factor equation is utilized:

$$\lambda = \frac{64}{1964.94} = 0.033$$

Darcy-Weisbach is used to calculate the remaining friction losses.

$$\Delta p_{fr_{40mm_1}} = \frac{1}{2} \cdot 0.05 \cdot \frac{1.280}{40 \cdot 10^{-3}} \cdot 827 \cdot 0.166^2 \cdot 827 \cdot (0.166)^2$$

$$\Delta p_{fr_{40mm_1}} = 18.23\ Pa$$

$$\Delta p_{fr_{40mm_2}} = \frac{1}{2} \cdot 0.05 \cdot \frac{1.149}{40 \cdot 10^{-3}} \cdot 827 \cdot (0.166)^2$$

$$\Delta p_{fr_{40mm_2}} = 16.36\ Pa$$

$$\Delta p_{fr_{50_{mm}}} = \frac{1}{2} \cdot 0.033 \cdot \frac{54.96}{50 \cdot 10^{-3}} \cdot 827 \cdot (0.106)^2$$

$$\Delta p_{fr_{50mm}} = 168.53\ Pa$$

$$\sum \Delta p_{fr} = \underline{789.57\ Pa}$$

The next step is to calculate the local friction losses in the pipe bends. In the schematics of the Oseberg pipeline from STID, the pipe bends are marked with the initials "LR" (long radius) (see Attachment 6.1.5), which indicates that the radius of the bend is equal to the diameter of the pipe. Using Figure 2.4, the local friction coefficient for the pipe bends is set to 0,45. Using Equation 2.6, the friction in the pipe bends can be found.

$$\Delta p_{loc} = \left(\frac{1}{2}\xi\rho V^2\right)\cdot pipe\ bends$$

$$\Delta p_{loc_{25mm}} = \frac{1}{2}\cdot 0.45\cdot 827\cdot 0.424^2\cdot 2 = 66.9\ Pa$$

$$\Delta p_{loc_{40mm_1}} = \frac{1}{2}\cdot 0.45\cdot 827\cdot 0.166^2\ \cdot 2 = 10{,}25\ Pa$$

$$p_{loc_{40mm_2}} = \frac{1}{2}\cdot 0.45\cdot 827\ \cdot 0.166^2\cdot 2 = 10{,}25\ Pa$$

$$p_{loc_{50mm}} = \frac{1}{2}\cdot 0.45\cdot 827\ \cdot 0.106^2\cdot 14 = 29.17\ Pa$$

$$\rightarrow \sum \Delta p_{loc} = 66.9 + 10.25 + 10.25 + 29.17 = 116.57\ Pa = 0.00165\ bar$$

The friction loss from valves in the Oseberg pipeline turned out to be negligible because the friction factor ξ_{valve} for fully open ball valves are 0,05 and 0,15 for fully open gate valves. (Idel'chik, 1966, Ch. 9.) As there is just two gate valves and two ball valves, the friction losses from these components would not make a difference in the bigger picture.

The pressure loss in the pipe reducers (from 50mm to 40mm and 40mm to 25mm) is also negligible.

The total pressure loss in the pipes, prior to the pump inlet is summarised below:

$$\sum \Delta p_{loss} = \sum \Delta p_{fr} + \sum \Delta p_{loc} = 0.0080 + 0.00165 \approx 0.01\ bar$$

Equinor desire a flowrate between 0,75 m^3/h - 2 m^3/h in the sampling cabinet. Therefore, it is also necessary to calculate the $NPSH_A$ value when the flow is set to 2 m^3/h. The only difference between the $NPSH_A$ value at 0,75 m^3/h compared to 2 m^3/h is the friction loss H_{fr}, as the velocity varies.

H_{fr} for a flowrate of 2 m^3/h is calculated using the same approach as for a flowrate of 0.75 m^3/h

$$\sum \Delta p_{loss} = \sum \Delta p_{fr} + \sum \Delta p_{loc} = 5642.61\ Pa \approx 0.05\ bar$$

3.2.3.4 NPSH$_A$ & NPSH$_R$

By combining the four factors and comparing the result against the given NPSH$_R$ value, it is possible to determine if the pump should theoretically experience cavitation.

$$NPSH_A = H_a + H_s - H_{vp} - H_f$$

$$NPSH_{A_{0,75}} = 1.25 + 0.305 - 0.982 - 0.01 = 0.562$$

$$NPSH_{A_{0,75}} = 0.562\ bar$$

$$NPSH_{A_{2.0}} = 0.519\ bar$$

$$NPSH_R = 1.28\ bar$$

$$NPSH_A < NPSH_R \rightarrow \underline{Cavitation}$$

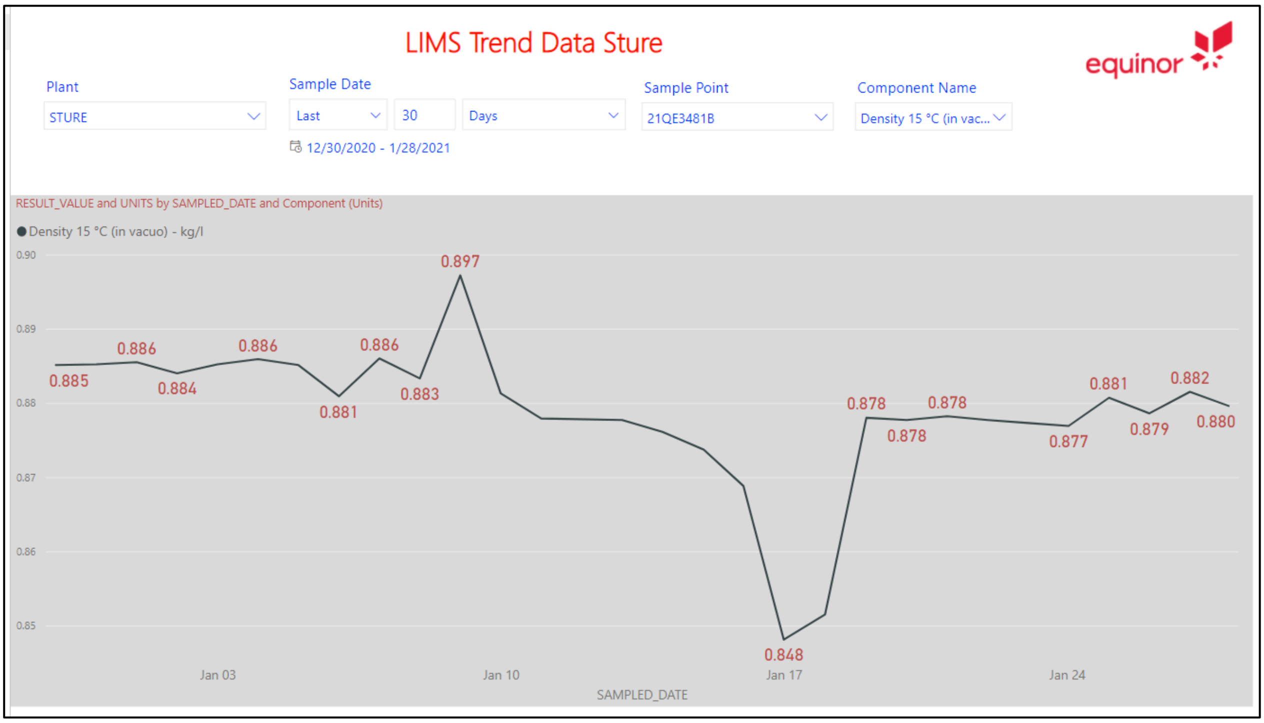
3.3 Grane (Results)

The pump at Grane has a different problem than the pump at Oseberg. Instead of having problems with cavitation, the sampler pump at the Grane system has problems handling wax particles in the crude oil that comes from Grane platform. The pump becomes clogged due to a substantial amount of wax particles in the oil, especially after pigging of the pipeline. The reason for this is most likely that the temperature at the pump inlet is 12.5°C and paraffin wax has a melting temperature approximately 30-40°C. The pump has a low $NPSH_R$ value of 0.19 bar, or 2m head (according to the manufacturer; GREENPUMPS) and can operate between -100°C and up to 315°C. P_{atm} at the Grane pipeline is 3 bar (according to STID).

3.3.1 Density

The density of the oil is given by Equinor (Table 3.7). It illustrates the density over a 30-day period. The density varies between 0.848-0.897 kg/l during the period due to change in water and wax content. The highest density is used when calculating the friction loss, whereas the lowest density is used when calculating the static head in section 3.3.3.1, The reason for this, is to achieve a $NPSH_A$ value that is appropriate for most scenarios.

Table 3.5 - Density of the oil during a 30-day period (Grane)



3.3.2 Viscosity

The viscosity at 20°C and 40°C is given by Equinor in Attachment 6.4. By utilizing Figure 2.5 (on page 31 in ‘Innføing I oljehydraulikk' (see section 2.3)), it is possible to find the viscosity of the oil mixture at different temperatures. This figure shows the kinematic viscosity as a factor of temperature.

By insertion of a new line using the viscosity at 20°C and 40°C. this is then used to obtain the viscosity at 12.5°C which is 27 cSt. (Brautaset, Innføring i oljehydraulikk, 1987, s. 31).

As the wax percentage varies the viscosity also varies. This is calculated using the Roscoe-Brinckmans equation (Equation 2.7) and presented in (Table 3.6). The percentage of wax φ is set between 0-60%. The correct amount of wax is found by matching the viscosities from the graph compared to the max viscosity of the pump. The max viscosity of the pump is stated as 250-300 cPs (supplied by the manufacturer), which converted to centistokes is between 285.7-342.9 cSt. For the oil to reach such high viscosity the oil mixture would have to exceed 50% of wax. One possible theory is that the wax particles get stuck into the volute of the pump, which will result in solid wax particles accumulating and eventually clogging the pump (even if the viscosity of the oil is below 250-300 cPs). If this theory is correct, it will not be possible to calculate the concentration of wax in the oil mixture. The viscosity given at 50% wax particles will be used in further calculations. The reason for this is because this is the highest realistic level of wax after pigging.

Figure 3.2 illustrates how the viscosity rises when the wax concentration increases and how the temperature of the fluid is a substantial factor when it comes to viscosity.

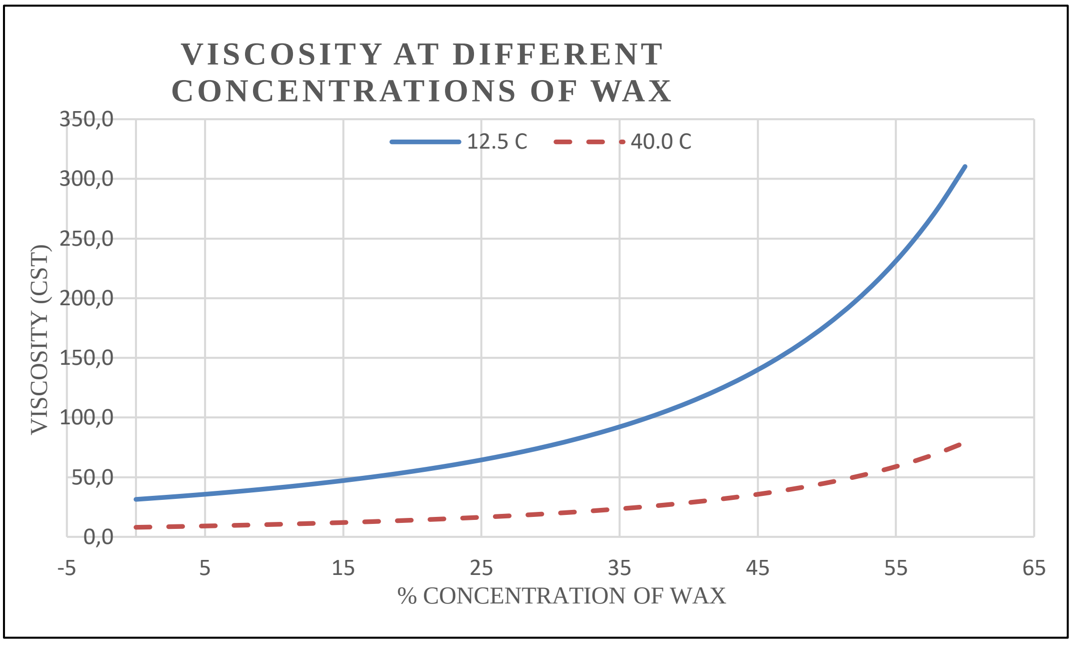


Figure 3.2 - Viscosity at Different Concentrations of Wax

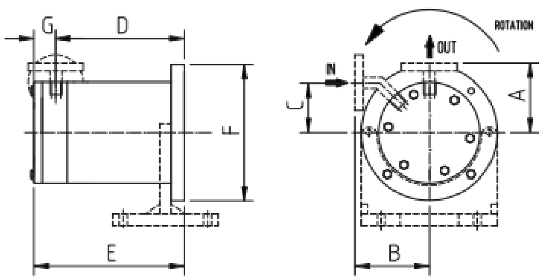
3.3.3 Current Pump on the Grane line

The current pump at the sampling cabinet at the Grane pipeline, is a CASTER MTA Regenerative Turbine Pump, model; MTA 49 (according to datasheet on STID).

Table 3.5 – Pump specifications

Manufacturer	CASTER
Series	MTA
Model	MTA 49
Distributor	Fuglesangs
RPM	2950
NPSH$_R$ [Bar]	0.19
Viscosity range	250-300 cPs
Dimensions in mm (not including motor)	196 x 207 x 200

The dimension of the pump is found by using a datasheet of the MTA 49 model provided by pump supplier AxFlow (Axflow). This supplier does not include dimensions in their datasheet, however, the RPM for the MTA 49 (STM 2.5x8 in the datasheet provided by AxFlow AS) is included, which is listed as 2950 rpm (50Hz). To find the dimension of the pump, a datasheet from GREENPUMPS is used to further find the correct GREENPUMPS model. Using the performance curves for 50Hz, a vertical head of 0.39m and a flowrate of 2.5 m^3/h, the GREENPUMPS model GPTA 30 is found. The dimensions of the pump model GPTA 30 is stated below:



PUMP TIPE	OVERALL DIMENSION							PORTS		mm / inches
	A	B	C	D	E	F	G	TRD	FLG	WEIGHT
GPTA20	88 / 3"1/2	88 / 3"1/2	54 / 2"1/8	150 / 6"5/8	198 / 8"3/8	160 / 6"5/8	23 / 7/8"	1/2"FPT / 1/2"NPT	DN 15 / DN 1/2"	5 Kg / lbs11
GPTA30	100 / 4"	98 / 3"7/8	61 / 2"3/8	155 / 6"3/4	207 / 8"5/8	200 / 6"5/8	25 / 1"	3/4"FPT / 3/4"NPT	DN 20 / DN 3/4"	6Kg / lbs14

Figure 3.3 – GPTA 30 dimensions from datasheet (Greenpumps)

3.3.4 Net positive Suction Head Results (Grane)

The following section presents the $NPSH_A$ calculations and compares the result against the $NSPH_R$value. Even though this pump does not have a problem with cavitation it is necessary to calculate the $NPSH_A$ value, as this is an important factor if the conclusion is to replace the existing pump. The $NPSH_A$ is expected to exceed 0.19 bar. According to the Equinor, the flow rate through the pump approximately $2 \pm 0.5\, m^3/h$ and the pressure is 3 barg. Since the flow rate has a relatively high uncertainty factor, calculations at the minimum and the maximum acceptable flowrate is necessary.

3.3.4.1 Calculating Static Head

The static head is calculated using Equation 2.10 The static head requires the height from P_{atm} to the pump inlet, which from section 3.1 is stated as 0.394 m.

$$H_s = \rho \cdot h \cdot g = 848 \cdot 0.394 \cdot 9.81 = 3277.63Pa \approx 0{,}033\ bar$$

3.3.4.2 Vapour Pressure

Since the Reid vapor pressure of the oil at the Grane-line is so close to the one at the Oseberg-line, the constants for C6H14 can be applied using Antoine equation (Sub Chapter 3.2.4). The constants are shown in Table 3.9.

Table 3.6 - Experimental Constants of Hydrocarbons (Willingham, 1945, s. 239)

Compound	Formula	A	B	C
2.3-Dimethylbutan	C_6H_{14}	6.80983	1127.187	228.900

$$\log_{10} p^{sat} = A - \frac{B}{C + T}$$

$$p^{sat} = 10^{\left(A - \frac{B}{C+T}\right)}$$

$$p^{sat}_{37{,}8°C} = 10^{\left(6.80983 - \frac{1127.187}{228.900 + 37.8°C}\right)} = 383.18\ mmHg \underline{\approx 0.51\ bar}$$

$$p^{sat}_{12{,}5°C} = 10^{\left(6.80983 - \frac{1127.187}{228.900 + 12.5°C}\right)} = 138.18\ mmHg \underline{\approx 0.18\ bar}$$

Although to some extent inaccurate, the estimated vapor pressure is set to 0.18 bar at 12.5°C.

3.3.4.3 Calculating Friction loss

Equation 2.1, 2.3 and 2.5 are used to calculate the friction losses at the Grane pipeline. Firstly, the friction losses in straight pipes are calculated.

$$\sum \Delta p_{fr} = \frac{1}{2}\lambda \frac{l}{d_h}\rho V^2$$

To calculate λ, Reynolds number and the type of fluid flow must be established. The velocity of the fluid is calculated from a flowrate of 2 m^3/h, which results in a velocity of 1,132 m/s in the pipe with diameter of 25 mm. The length of the pipe is stated in 3.1 and the viscosity of the fluid is set to 155.5 m^2/s.

$$Re = \frac{V \cdot d}{v} = \frac{1.132 \cdot 0.025}{155.5 \cdot 10^{-6}} = 182 \rightarrow laminar\ flow$$

$$\rightarrow \lambda = \frac{64}{Re} = 0.35$$

$$\Delta p_{fr_{25mm}} = \frac{1}{2} \cdot 0.35 \cdot \frac{3.533}{0.025} \cdot 897 \cdot 1.132^2 = 29238.9Pa \approx 0.3bar$$

The local pressure losses from pipe bends and valves are negligible as there is only one ball valve and two smooth pipe bends (“LR”) that causes very low amounts of local friction.

The fluid passes through a quill shortly after P_{atm}, which (according to Equinor) is very similar to the quill at the Oseberg pipeline. For this reason, a pressure drop of 0,2 bar is also included in calculating the friction losses at the Grane pipeline, even though this is somewhat inaccurate.

$$\sum \Delta p_{fr} = 0.2bar + 0.3bar = \underline{0.5bar}$$

3.3.4.4 Net Positive Suction Head Results (Grane)

To calculate the $NPSH_A$ value at the Grane pipeline, Equation 2.8 is used.

$$NPSH_A = H_a + H_s - H_{vp} - H_{fr} = 3 + 0.033 - 0.18 - 0.5 = 2.353 bar$$

$$\rightarrow NPSH_A \gg NPSH_R$$

The calculation confirms that the $NPSH_A$ value is much greater than the $NPSH_R$ value for the pump, as expected.

The amount of wax in the crude oil varies daily, and it is difficult to find out the exact concentration of wax in the fluid. The concentration of wax affects the $NPSH_A$ value, where it will decrease as the wax content increases. This is because the viscosity of the fluid increases as the wax concentration increases, resulting in a lower $NPSH_A$ value. To get an impression of how the $NPSH_A$ value changes as the viscosity changes due to wax, Figure 3.4 is plotted using wax concentrations up to 60%. The graph also considers the $2 \pm 0.5\, m^3/h$ flowrate of the fluid.

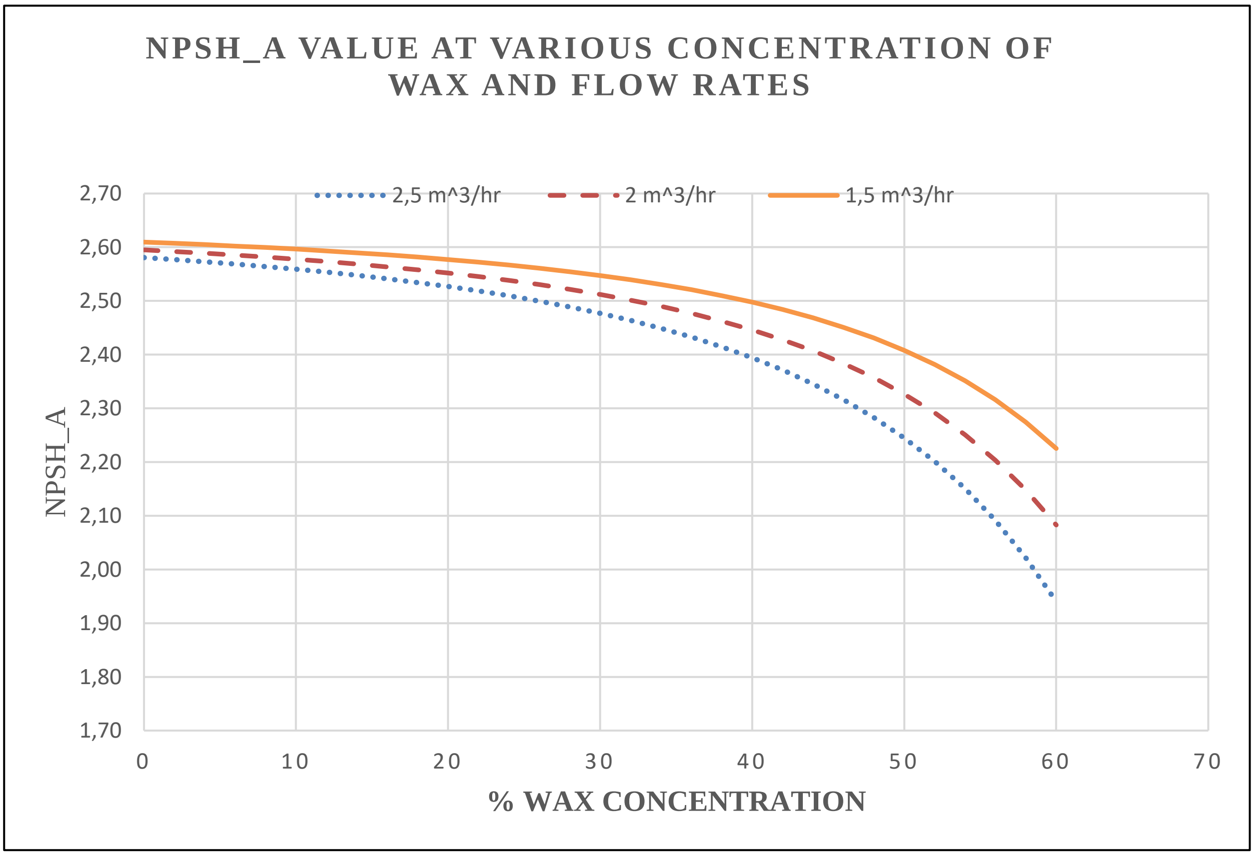


Figure 3.4 - NPSH_A Value at Various Concentrations of Wax and Flow Rates

As presented in the graph, the $NPSH_A$ value decreases a considerable amount when the wax concentration increases. This is because the friction loss in the pipe is largely affected by the change in viscosity, as an increase in viscosity increases the Darcy friction factor, λ.

3.4 Sources of error

There are several sources of error when working with mechanical instruments and manual readings. This section will discuss possible errors and inaccurate values.

3.4.1 Parallax Error

Parallax is a displacement or difference in the apparent position of an object viewed along two different lines of sight (Oxford English Dictionart, 2005). When manually reading the value from a pressure gauge, one possible source of error is parallax error – An error in reading an instrument as a result of the pressure scale and the indicator (pointer) not being precisely coincident.

To avoid parallax error, it is important that the user's line of sight is perpendicular to the scale.

3.4.2 Pressure in "Main Pipe"

While discussing pressure in various pipes, it was discovered that there is a shortage of pressure indicators leading to the sampling pump connected to the Oseberg line. According to Equinor's document system, STID, the pressure is set to 2 bar in the "main pipe", but due to bends, transitions, valves, and various instruments, the pressure is not stable though the whole system and that the value is in fact taken from a pressure gauge installed at a point significantly further from the "main pipe" than first anticipated. It was also discovered that 2 bar is an old value.

It is therefore important to mention that the atmospheric pressure used in the $NPSH_A$ calculations at Oseberg is not completely accurate. Equinor estimate that the true value is closer to 1,25 bar.

The same uncertainty applies to the main supply pipe at Grane, where the pressure is set to 3 bar.

3.4.3 Change in Pressure Across the Quill

There is a pressure drop between the main pipe and the quill that extracts an oil from the main flow.

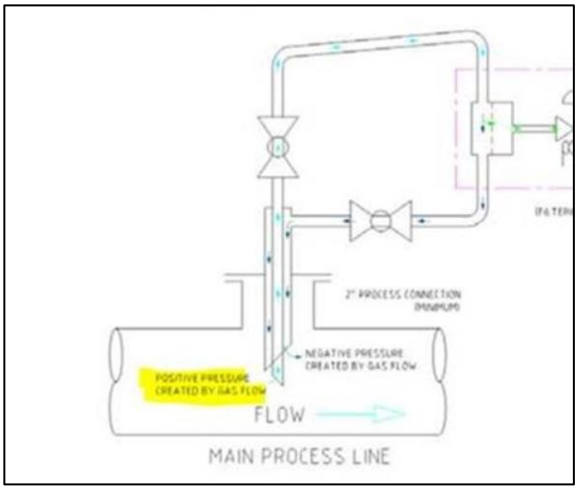


Figure 3.5 - Quill

It is possible to reduce the pressure drop by adjusting the angle of the quill, such that the opening of the quill faces the direction of the flow.

There are no pressure indicators installed directly after the quill, only at the inlet on the pump, mounted 62 meters after the quill. This causes uncertainties when calculating the pressure drop due friction (H_{fr}).

3.4.4 Vapor Pressure Estimate

The vapor pressure value Equinor publish in their annual Crude Summary Report is the vapor pressure given at 37,8°C, not the operational temperature. To calculate the vapor pressure at various temperatures, Antoine equation (Equation 2.11) is used.

The Antoine equation describes the relation between vapor pressure and temperatures for pure substances. Unfortunately, oil is not a pure substance, but rather a mixture of hydrocarbons. The calculations are based on Dimethylbutan (C_6H_{14}) – An organic compound with a vapor pressure close to the vapor pressure of the oil mixture.

According to Equinor's report, the vapor pressure of the oil mixture (at 37,8°C) is equal to 7,7 psi or 0,53 bar. Antoine equation shows that the vapor pressure of Dimethylbutan is 386 mmHg or 0,51 bar.

As the vapor pressure is estimated for Dimethylbutane, *not* the actual oil mixture, the value entails an uncertainty to some extent.

3.4.5 $NPSH_R$ value (meters liquid column)

In the original problem statement supplied by Equinor, and included in the documentation package, the $NPSH_R$ value for the pump at Oseberg was stated as 3,5 barg. The team perceived this value as unusually high, but the value was not questioned at that time on the assumption that all project data and criteria had been assessed prior to study inception.

Further along in the research on the currently installed pump, it was discovered that the $NPSH_R$ value is not 3.5 barg, but rather 3.5 meters liquid column (mlc). After notifying Equinor, the error was corrected, and the value was reset to 0.28 barg (even though this value is also a calculated estimate by Equinor). This new information did not become evident until a few days before the (final) report was due. Accordingly, the study team wish to inform future readers that certain variables in the $NPSH_A$ calculation may be inaccurate.

4 Conclusion

This section is a summarisation of the most important results, in addition to a proposal for the way forward for the project.

4.1 Oseberg

On the Oseberg-line, the main result is the $NPSH_A$ value. Although Equinor has reduced the flow below the recommended minimum flow, calculations suggests that the pump still cavitates.

Conclusion: The pump needs to be replaced with suitable alternative.

$$NPSH_{A_{0.75}} = 0.564\ bar$$

$$NPSH_{A_{2.0}} = 0.519\ bar$$

$$NPSH_R = 1.28\ bar$$

$$NPSH_A < NPSH_R \rightarrow \underline{Cavitation}$$

Equinor state that they need a pump that can operate with a flow between 0,75 m^3/h - 2 m^3/h. Any replacement pump should be chosen accordingly. A safety margin of 1 meter (0,09 bar) is applied (see section 2.5). It is also important that the dimensions of a replacement pump are suitable for the sampling cabinet. However, drawings with sufficient annotation of the sampling cabinet are not available on STID, only drawings that are partly dimensioned (see Attachment 6.1.7). For this reason, the replacement pump should not me bigger than the current pump.

Important requirements for the replacement pump:

1. $NPSH_R < 0.5$bar
2. Flowrate between 0.75 m^3/h - 2 m^3/h
3. Operational temperature > 60℃
4. Maximum operational viscosity > 25 cSt
5. Minimum vertical head > 3.67 m (same as static head)
6. Dimensions of the pump in mm ≤ 190 x 407 x 252 (see Table 3.1)

4.2 Grane

The problem on the Grane-line is caused by a high amount of wax particles in the oil.

The study concluded with two theories.

Theory 1

The first theory was that the wax particles increase the viscosity of the oil to go beyond the max viscosity set by the pump manufacturer. This would cause problems to the pump and eventually cause it to stop circulating. Calculations were made to find the amount of wax particles that needs to be present in the oil for the viscosity to go beyond the max viscosity of the pump. The results suggest that the wax percentage is greater than 50%, which is considered to be unrealistic. Therefor another theory was considered.

Theory 2

Since the first theory did not seem possible, a conclusion was made that the wax particles stuck to the volute of the pump. As this would gradually decrease the volute area and eventually stop the flow completely, this seems like a possible and logical reason to the problem Equinor are experiencing.

Due to project limitations, it is not possible to test these theories which must therefore remain as just that – theories – albeit well researched and documented.

Conclusion: The best option for the sampler system at the Grane pipeline is to replace the existing pump with a pump that can handle particles of wax better. A positive displacement pump, such as a screw pump or a vane pump, is a possible candidate for a new pump. Screw pumps are able to break down the wax particles with the screw gear mechanism.

Important requirements for any replacement pump:

1. Must be able to handle wax particles
2. Maximum operational viscosity > 310 cSt
3. $NPSH_R$ < 2.343 bar
4. Must operate optimal with a fluid temperature of 15℃
5. Minimum vertical head > 0.39 m (same as static head)
6. Dimensions of the pump in mm ≤ 196 x 207 x 200 (see Table 3.5)

4.3 Project Proposal

Since the vapor pressure estimate (0.98 bar) at the Oseberg pipeline is greater than the $NPSH_A$ value (0.5 bar), it is recommended to lower the temperature of the crude oil blend to $< 35℃$, as this will make sure the vapor pressure is below the $NPSH_R$ value, which is important to avoid cavitation. However, lowering the temperature at the Oseberg pipeline has the potential to increase the amount of solid wax particles in the crude oil, because paraffin wax has a melting point between 30-40 ℃. This, however, should not be a problem using a screw pump or (some) vane pumps.

4.4 Pump Suggestions

The team has been in contact with both national and international pump suppliers but are yet to receive a proposal from the suppliers. Therefore, one possible pump suggestion for the Grane and Oseberg sampling cabinet is presented below, however, a potentially better suited pump from one of the pump suppliers will be presented at the presentation for this bachelor thesis in June. So far PSG Dover is the only pump suppliers that have proposed a suggestion.

4.4.1 Suggestion for Oseberg

The $NPSH_A$ value is below the of $NPSH_R$ value, meaning that the only suitable option is to replace the current pump at the sampling cabinet with a new pump that fit the requirements listed in Section 4.1. Since there are no problems due to wax particles in the centrifugal pump at the Oseberg pipeline, a centrifugal pump, or a vane pump, with a lower $NPSH_R$ value would be suitable as a replacement pump. The pump suggested for Grane in section 4.3.2 is also suitable for the sampling system at Oseberg.

"PSG Dover" has recently suggested a pump for both the Grane and Oseberg sampling cabinets. They have suggested Blackmer XL1.25 (PSG Dover, 2021), a sliding vane pump that can operate under low flowrates and have a substantially low $NPSH_R$ value (according to the manufacturer). Sliding vane pumps are more appropriate for systems with a low $NPSH_A$ value compared to many centrifugal pumps (VanLeeuwen, 2020).

This pump is also suitable on the Grane sampling cabinet, as the wax particles in the crude oil are not substantially large ($< 40\mu m$) (D. Eskin, 2011). The manufacturer states that the XL1.25 model will be able to handle these wax particles.

The Blackmer XL1.25 pump must be further evaluated before a final decision can be made. The team will reach a decision and present the final pump suggestion(s) at the bachelor thesis presentation in June.

4.4.2 Suggestion for Grane

There are several methods of separating wax particles from oil. Effective methods that were considered, include installing either a cyclone pump or a heat exchanger before the pump.

A cyclone is a cone-shaped separating device that uses centrifugal force to remove solid particles from a liquid. This is a possible solution, but not a space-efficient solution.

Wax particles melt and become homogeneous with the oil at 30 - 40°C. (Gudmundsson, 2018) By installing a heat exchanger directly prior to the pump, it is possible to melt the particles and avoid the pump clogging up. This method was considered but disregarded as it is not space-efficient, nor cost-efficient.

The best solution is to replace the current centrifugal pump with a positive displacement pump. One of the main advantages with displacement pumps over centrifugal pumps is that they can easily handle highly viscous fluids, containing solid particles. The reason for this is that positive displacement pumps, such as screw pumps and gear pumps grind the particles as they move through the pump. This prevents the particles from solidifying on the volute lip and clogging the pump.

Blackmer AS manufacture a twin-screw pump with timing gear, available in a wide range of configurations, including horizontal, vertical, and suitable for high viscosity fluids. The pumps provide flow rates ranging from 0,3 – 150 m^3/hr at pressures up to 60 bar. The pumps can also handle viscosities ranging from 0,5 – 200 000 cSt and temperature up to 350°C.

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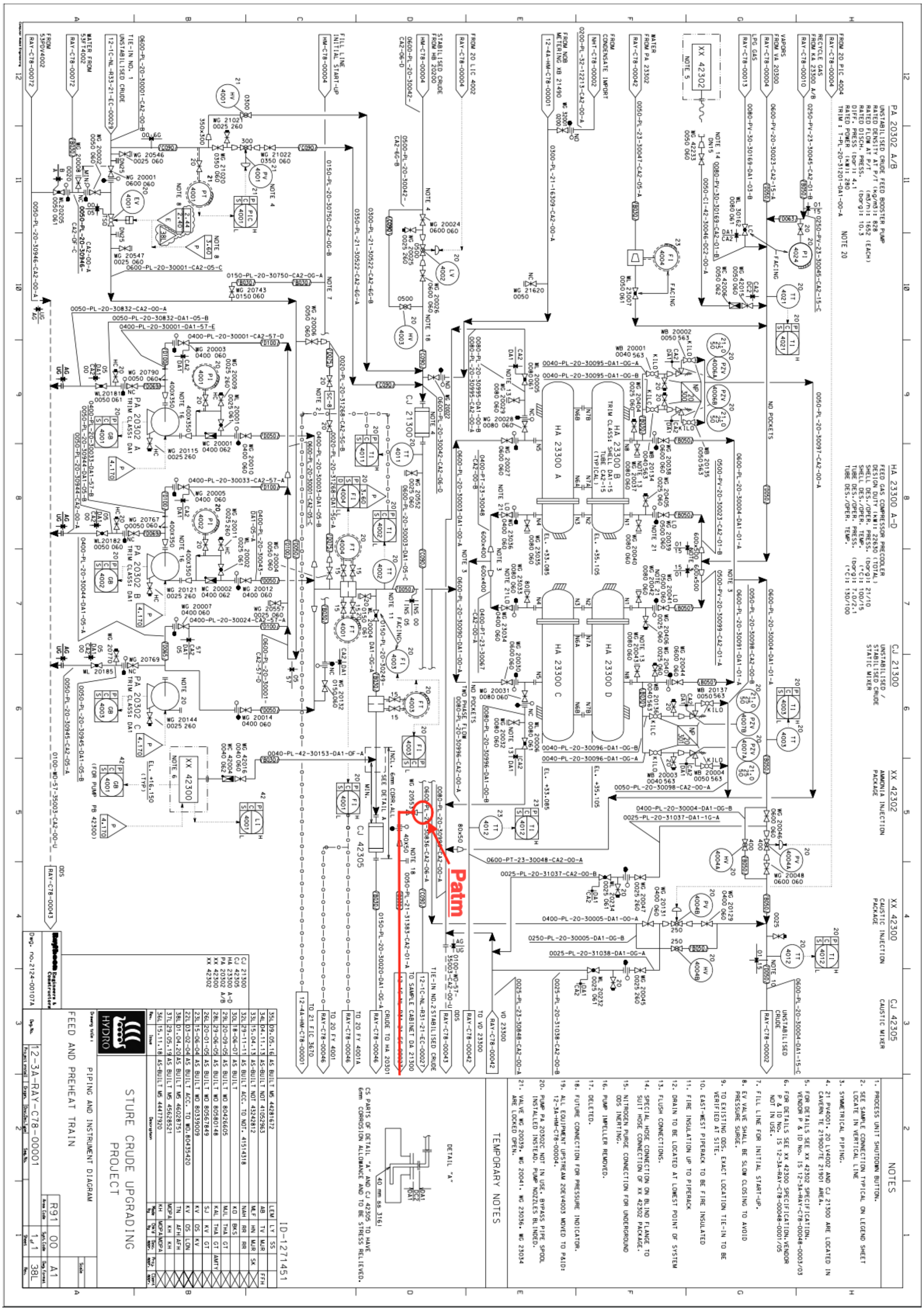
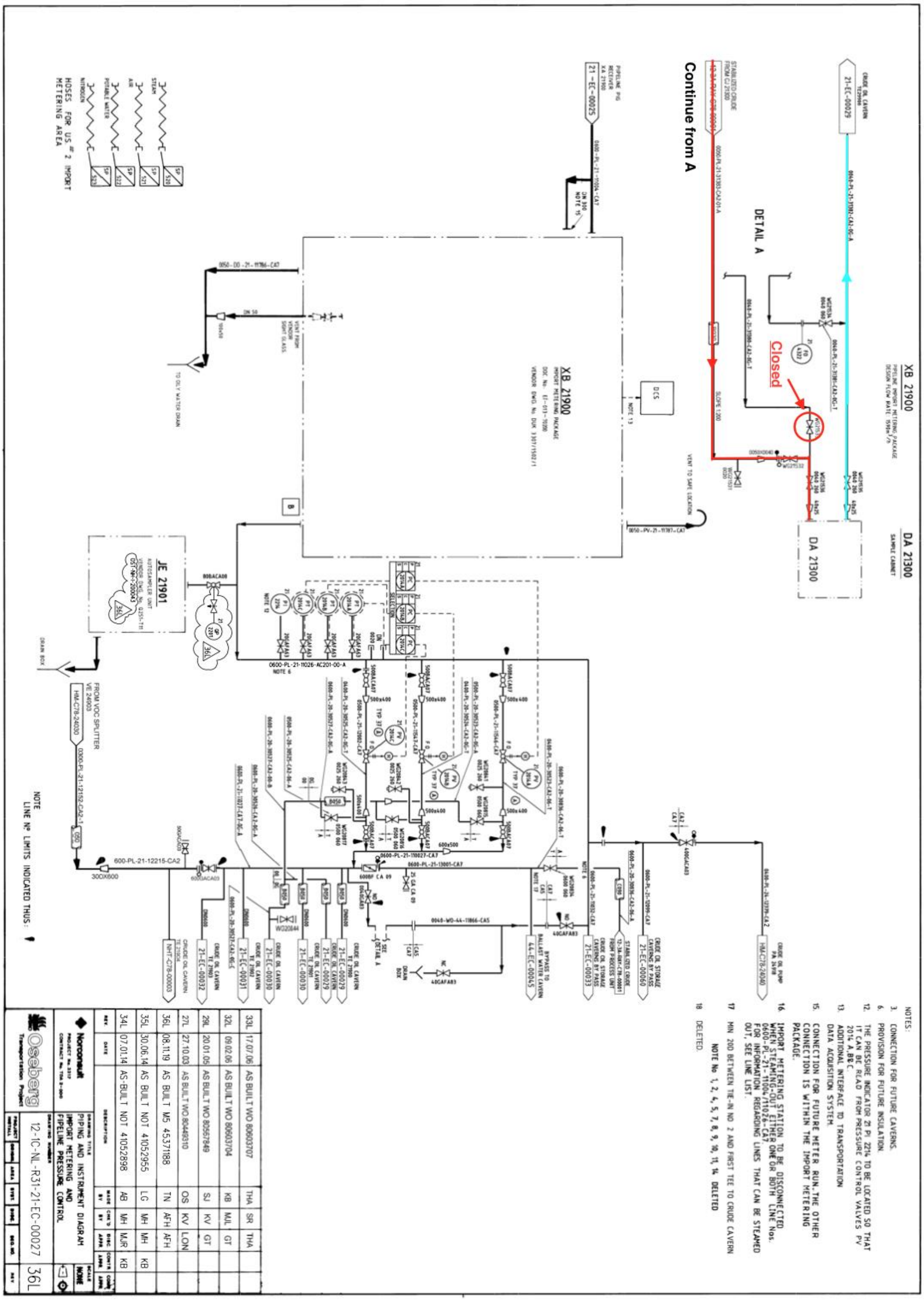
6 Attachments

6.1 Oseberg Pipeline

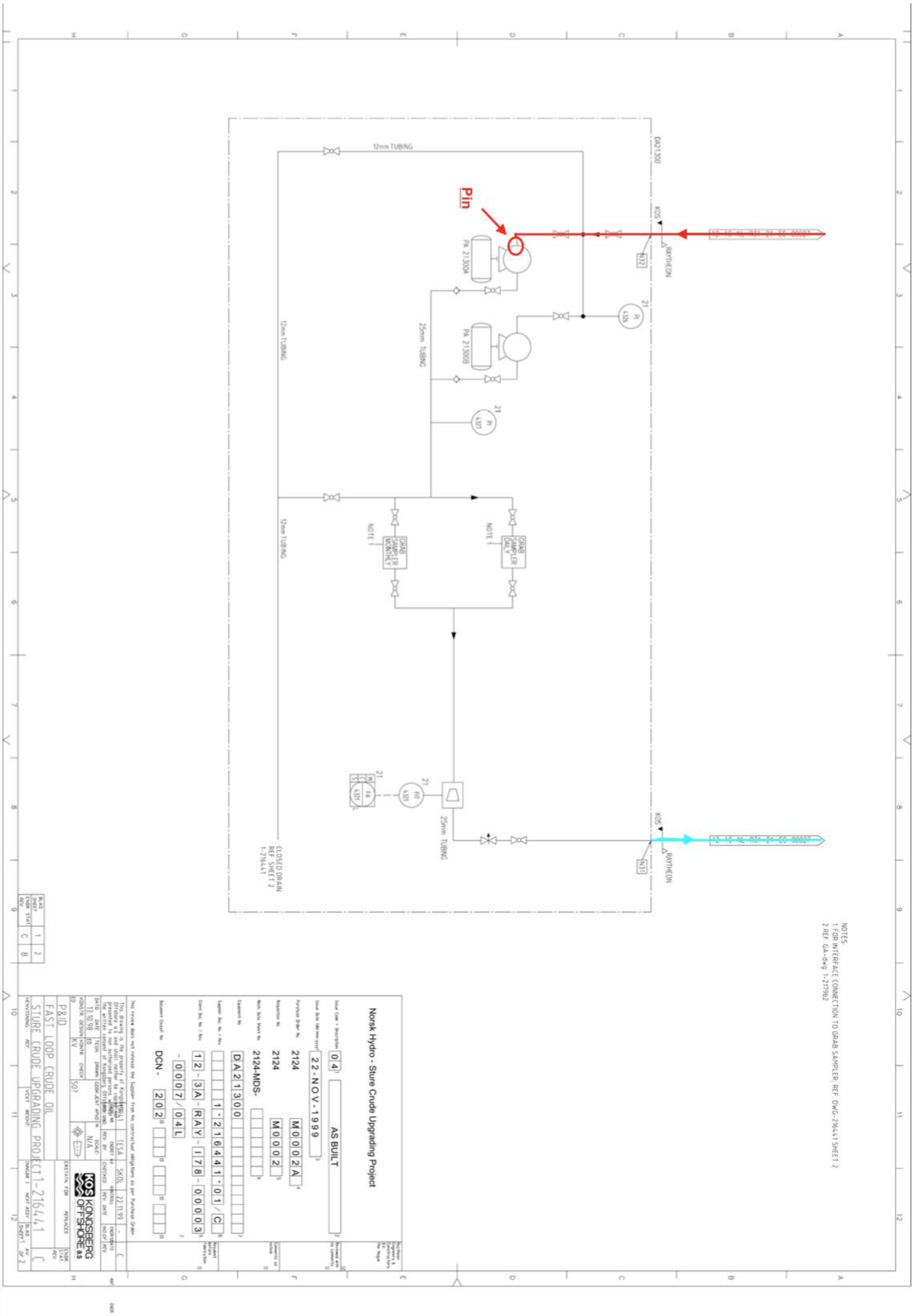
Attachment 6.1.1: A



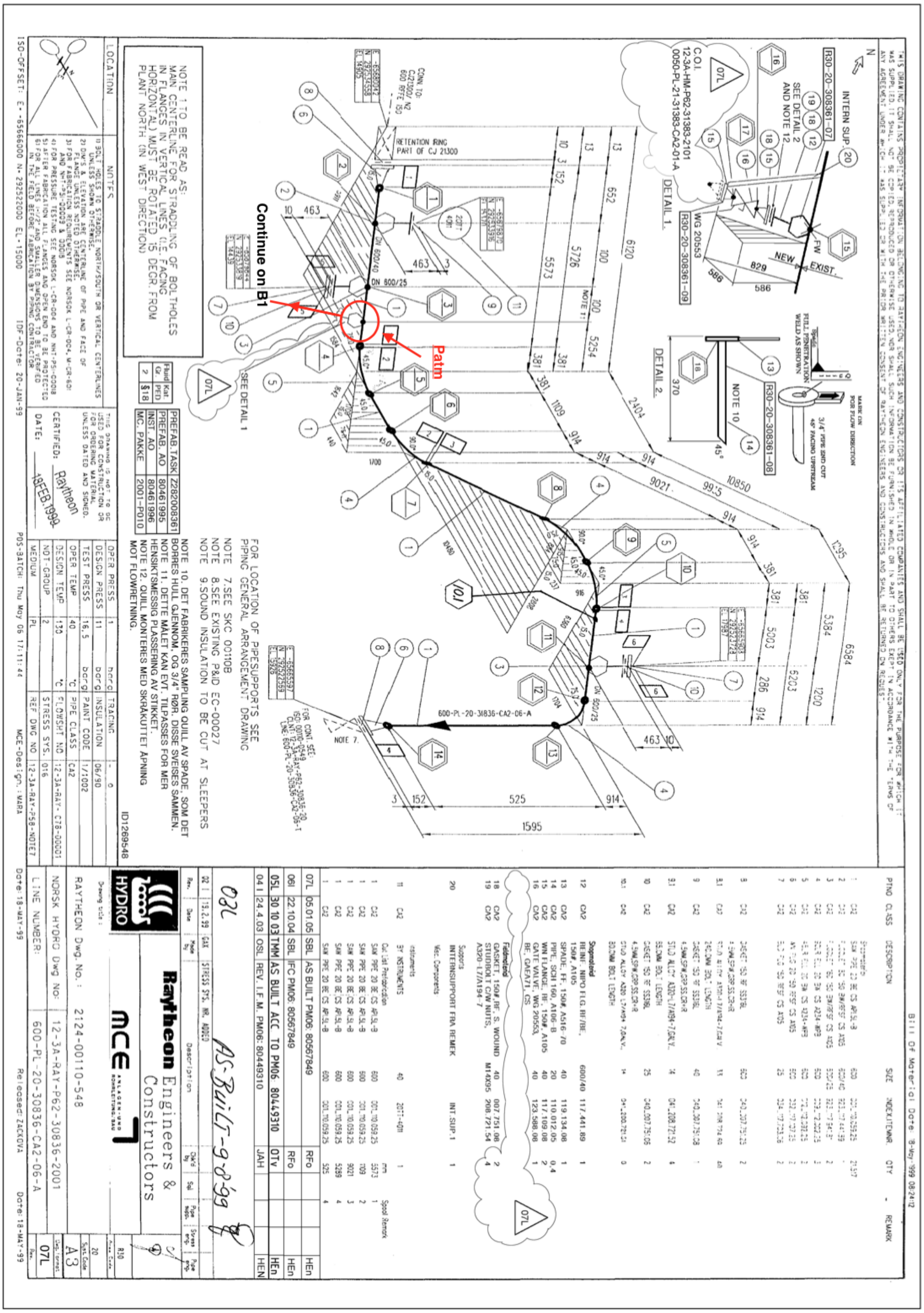
Attachment 6.1.2: B



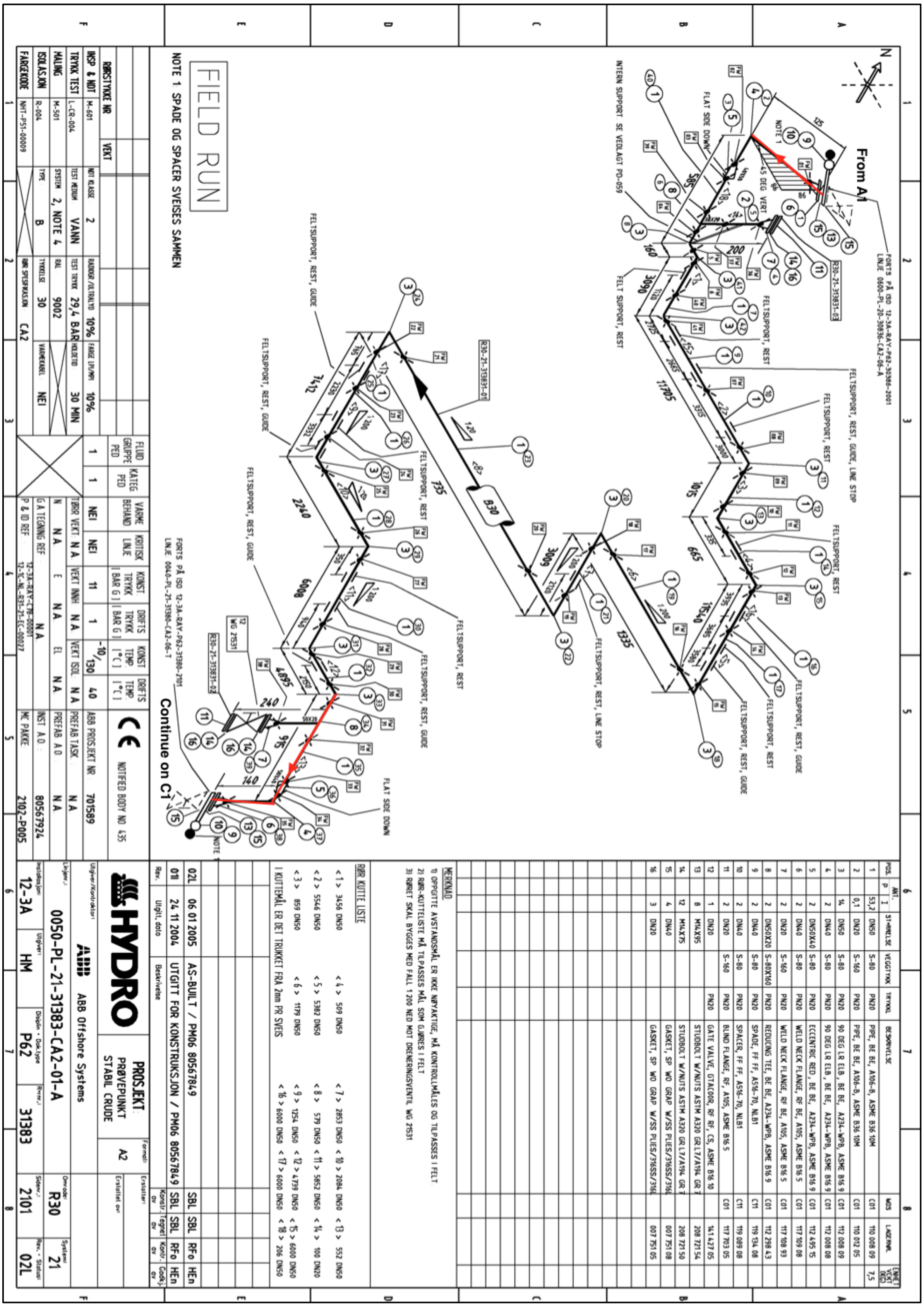
Attachment 6.1.3: DA 21300



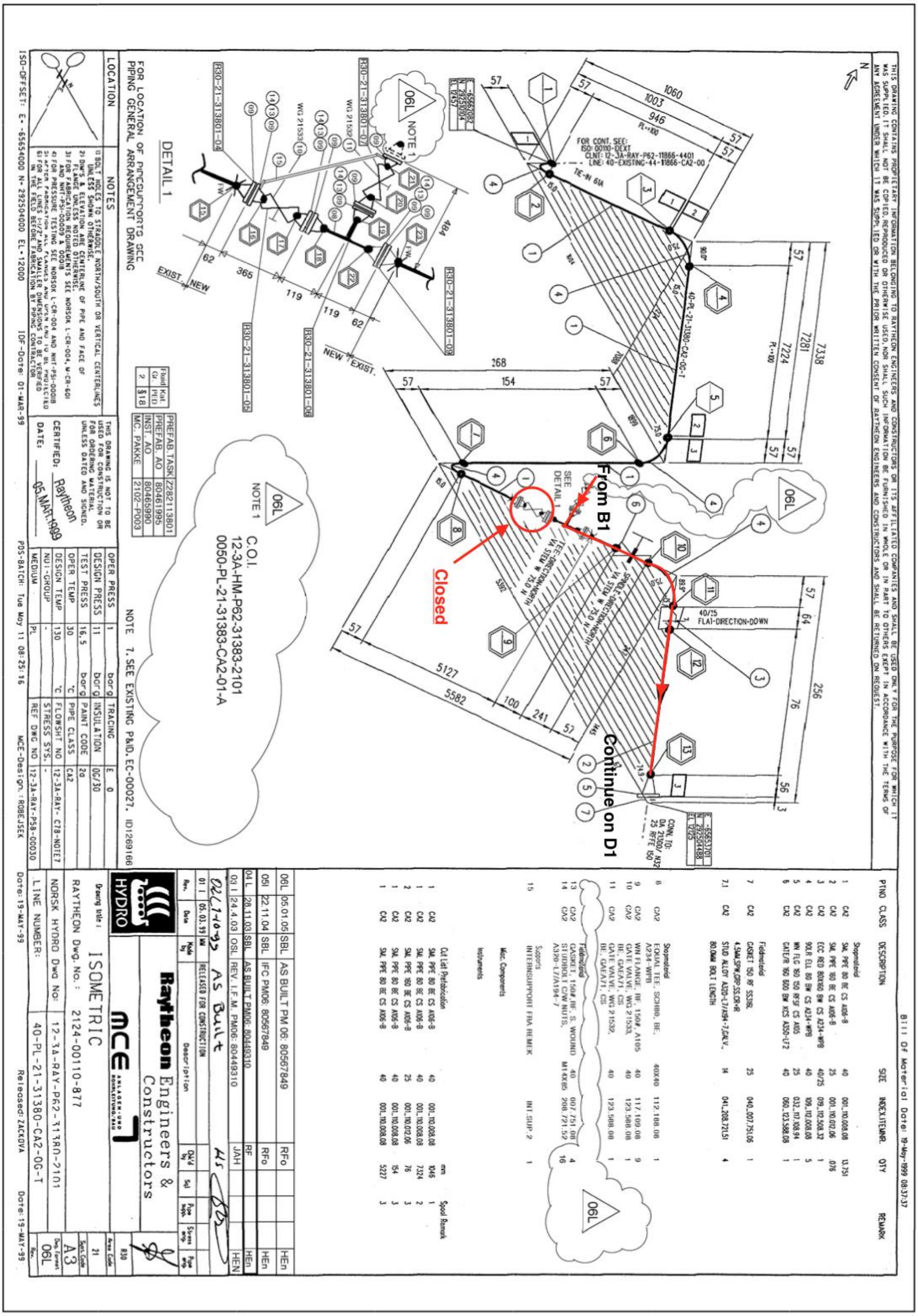
Attachment 6.1.4: A1



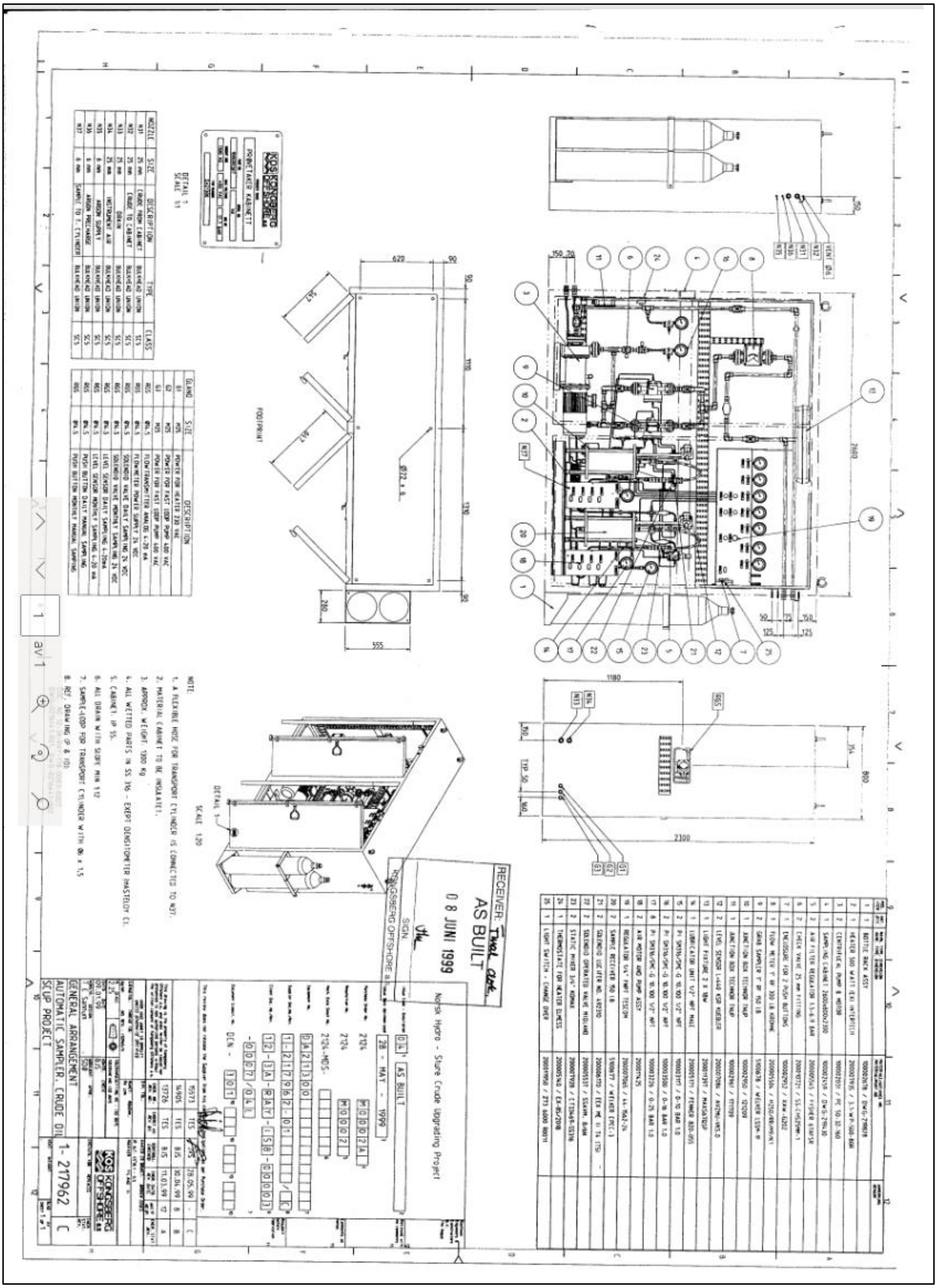
Attachment 6.1.5: B1



Attachment 6.1.6: C1

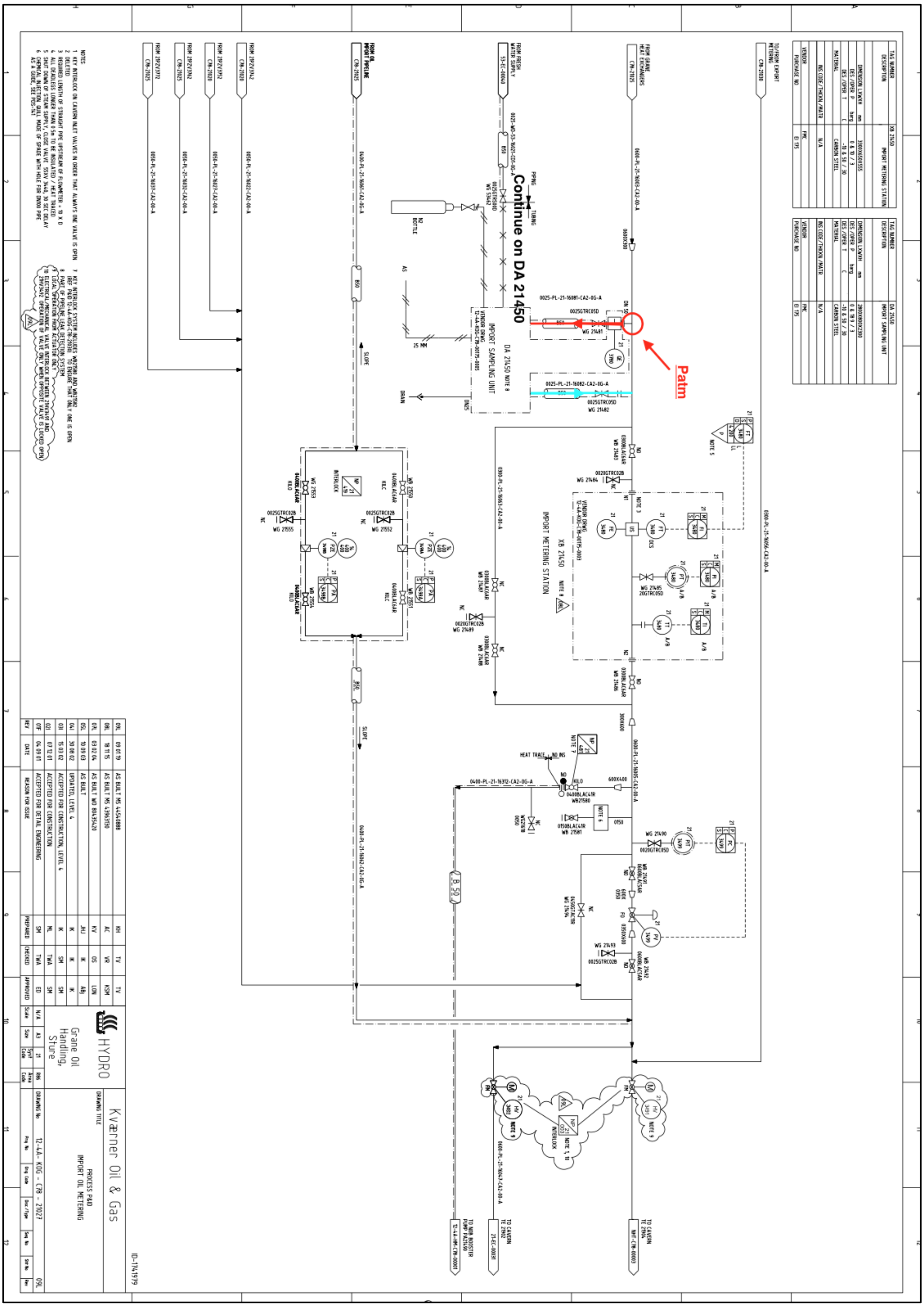


Attachment 6.1.7: D1 (DA 21300 - dimensions of sampling cabinet)

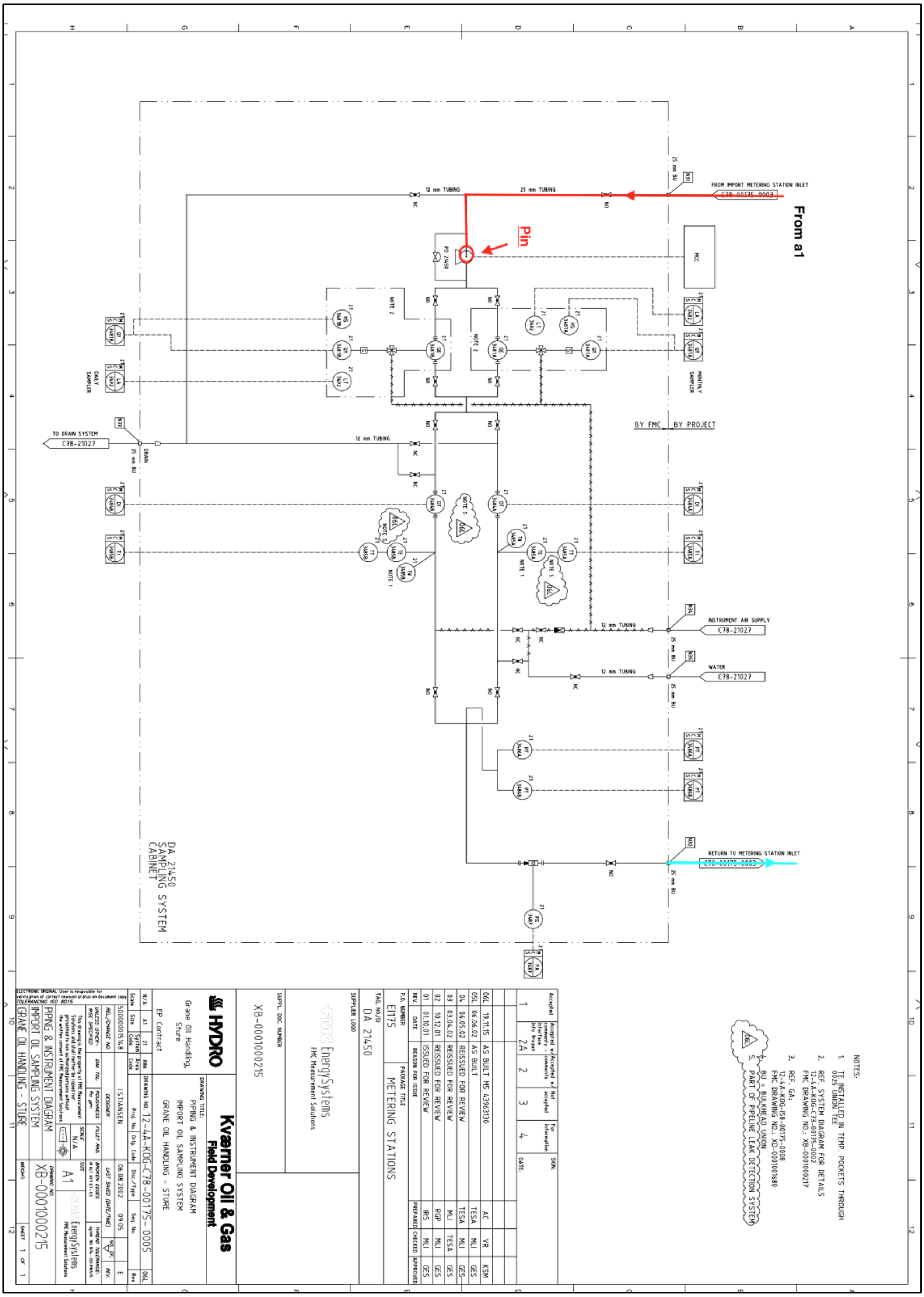


6.2 Grane Pipeline

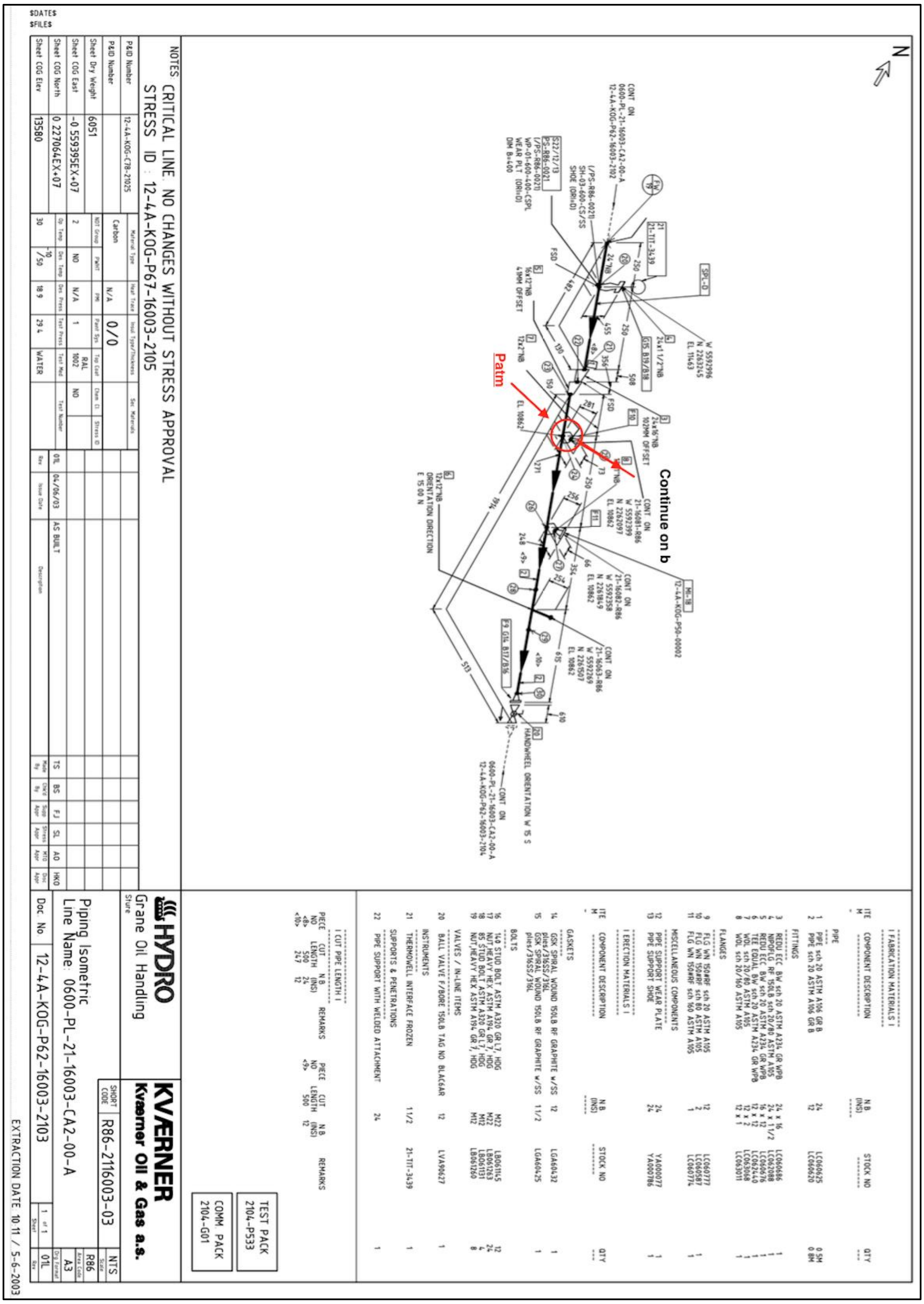
Attachment 6.2.1: a1



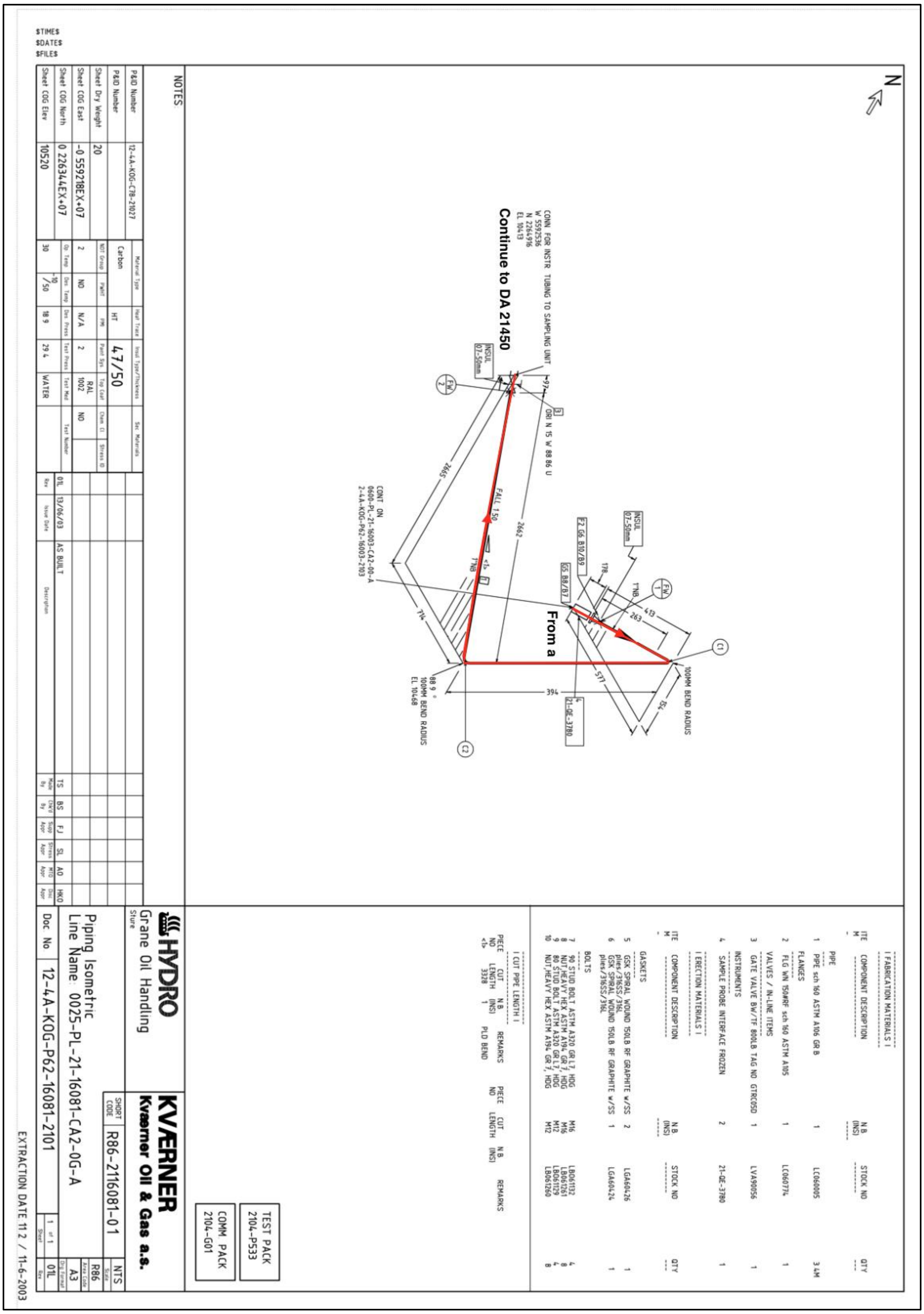
Attachment 6.2.2: b1



Attachment 6.2.3: a

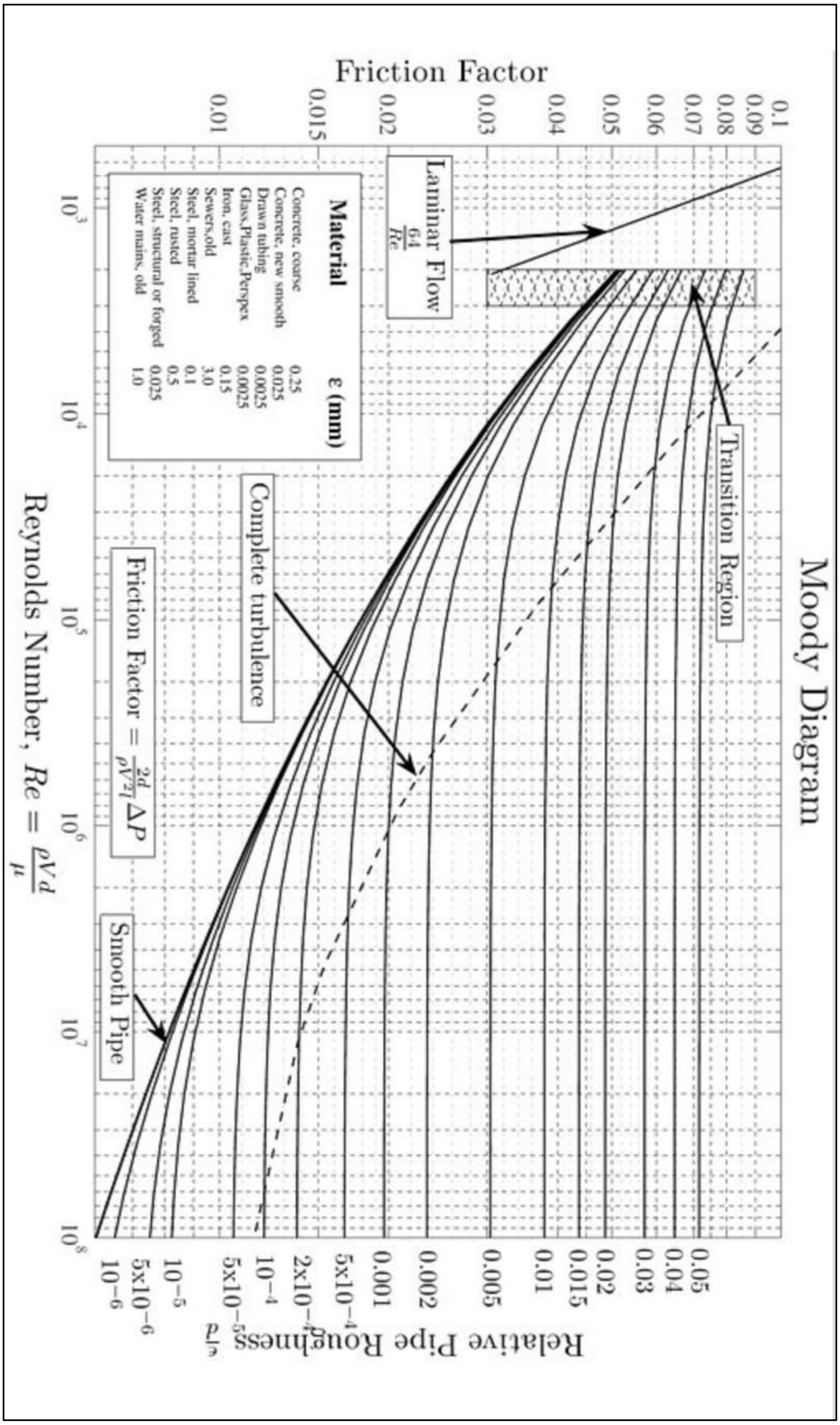


Attachment 6.2.3: b



6.3 Other

Attachment 6.3.1: Moody Diagram



6.4 Grane Blend

Crude Summary Report

General Information		Molecules (% wt on crude)		Whole Crude Properties	
Name:	GRANE BLEND 2020 10	methane + ethane	0,04	Density @ 15°C (g/cc)	0,875
Reference:	GRANEBLEND202010	propane	0,49	API Gravity	30,2
Traded Crude:	Grane	isobutane	0,27	Total Sulphur (% wt)	0,57
Origin:	Norway	n-butane	0,90	Pour Point (°C)	0
Sample Date:	30 oktober 2020	isopentane	0,61	Viscosity @ 20°C (cSt)	17
Assay Date:	04 januar 2021	n-pentane	0,88	Viscosity @ 40°C (cSt)	8
Issue Date:	04 januar 2021	cyclopentane	0,11	Nickel (ppm)	2,6
Comments:		C_6 paraffins	1,55	Vanadium (ppm)	8,5
		C_6 naphthenes	1,10	Total Nitrogen (ppm)	1502
		benzene	0,23	Total Acid Number (mgKOH/g	0,87
		C_7 paraffins	1,32	Mercaptan Sulphur (ppm)	1
		C_7 naphthenes	1,70	Hydrogen Sulphide (ppm)	0,0
		toluene	0,78	Reid Vapour Pressure (psi)	7,7

Taken from: [Crude oil assays - Crude oil assays - equinor.com]

6.5 Oseberg Blend

Crude Summary Report

General Information		Molecules (% wt on crude)		Whole Crude Properties	
Name:	OSEBERG 2016 04	methane + ethane	0,02	Density @ 15°C (g/cc)	0,827
Reference:	OSEBERG201604	propane	0,61	API Gravity	39,6
Traded Crude:	Oseberg	isobutane	0,38	Total Sulphur (% wt)	0,20
Origin:	Norway	n-butane	1,18	Pour Point (°C)	-15
Sample Date:	16 April 2016	isopentane	1,03	Viscosity @ 20°C (cSt)	4
Assay Date:	20 May 2016	n-pentane	1,54	Viscosity @ 40°C (cSt)	3
Issue Date:	24 May 2016	cyclopentane	0,17	Nickel (ppm)	1,0
Comments:	Sampled at the Sture terminal.	C_6 paraffins	2,92	Vanadium (ppm)	1,0
		C_6 naphthenes	1,93	Total Nitrogen (ppm)	739
		benzene	0,48	Total Acid Number (mgKOH/g	0,12
		C_7 paraffins	2,80	Mercaptan Sulphur (ppm)	1
		C_7 naphthenes	2,90	Hydrogen Sulphide (ppm)	-
		toluene	1,17	Reid Vapour Pressure (psi)	8,0

Taken from: [Crude oil assays - Crude oil assays - equinor.com]