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TKP4170 PROCESS DESIGN PROJECT

Title: Åsgard A Top Side Compressor Train studies for Equinor	**Keyword (3-4):** Compressors, energy savings, anti-surge
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Summary:
In this project the Equinor topside facility of Åsgard A is studied for potential energy savings around the compressor train. In the current operation, the compressor train is identified to operate with partially open recycle valves, to optimise the process for a high efficiency. However, by closing the recycle valves further until the control surge points are reached, energy can be saved. The potential energy savings are calculated to be 9.54 MW. An option for replacing these compressors is evaluated assuming a constant production profile, which shows that the project has a positive NPV when the oil price is normal (498.40 NOK) to high (747.45 NOK). With ageing wells, the pressure is depleted and bottlenecks in the plant are identified to be the export compressors. The export compressors in this case would be unable to meet the landing pressure specifications of the export gas (being a constraint given by Equinor).

Conclusions and recommendations:
Focus should be placed on trying to close the recycle valves as much as possible and operating closer to the surge control line to realise the energy savings. Equinor should consider the benefits of CO_2 reduction as well as the improvements in production that this project offers in comparison to similar opportunities that are available before making an investment decision. Possible modifications to the export compressors need to be analysed if it is required to maintain export gas while well pressures deplete in the future.

Date and signatures:
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Abstract

In this project the Equinor topside facility of Åsgard A is studied for potential energy savings around the compressor train. In the current operation, the compressor train is identified to operate with partially open recycle valves, to optimise the process for a high efficiency. However, by closing the recycle valves further until the control surge points are reached, energy can be saved. The potential energy savings are calculated to be 9.54 MW. An option for replacing these compressors is evaluated assuming a constant production profile, which shows that the project has a positive NPV when the oil price is normal (498.40 NOK) to high (747.45 NOK). With ageing wells, the pressure is depleted and bottlenecks in the plant are identified to be the export compressors. The export compressors in this case would be unable to meet the landing pressure specifications of the export gas (being a constraint given by Equinor).

Focus should be placed on trying to close the recycle valves as much as possible and operating closer to the surge control line to realise the energy savings. Equinor should consider the benefits of CO_2 reduction as well as the improvements in production that this project offers in comparison to similar opportunities that are available before making an investment decision. Possible modifications to the export compressors need to be analysed if replacement of these is required to maintain export of gas while well pressures deplete in the future.

Contents

1 Introduction

The Åsgard oil and gas facility is operated by Equinor Energy AS in the Åsgard field located in the Norwegian sea. The Åsgard field was discovered in 1981, and includes deposits Smørbukk, Smørbukk Sør and Midgard from which the facility started exporting gas in 2000. Production of the before mentioned deposits is based on pressure depletion with in most cases pressure support from gas injection. This is mainly because of the low amount of resources present in the Åsgard field (initial: 425.3 mill Sm^3 oil equivalents; current: 79.7 mill Sm^3 oil equivalents) [1].

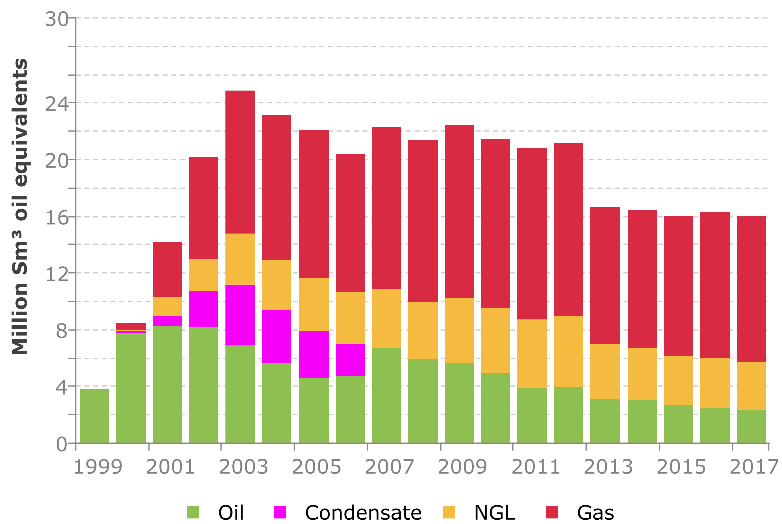


Figure 1.1: Production profile of Åsgard field [1]

The oil and gas facility consists of three main parts [1]:

- Åsgard A - Production, storage and offloading vessel (FPSO) + subsea;
- Åsgard B - Semi-submersible facility for gas and condensate processing;
- Åsgard C - Storage vessel for condensate.

The oil and condensate are temporarily stored at the cargo vessel of Åsgard A, and then shipped to land by tankers. The gas from Åsgard A is first refined at Åsgard B, before it is exported to Kårstø terminal in Nord-Rogaland [1].

In this project, the operating condition of the Åsgard A topside facility is studied, with a focus on the compressors section. Currently, the facility is inefficient due to pressure depletion of the wells compared to the initial production in that started in 2000. This causes a lower flow rate of gas through the compressors making them operate closer to their surge points. To avoid surge the compressors are operating with open anti-surge valves.

The goal of this project is to perform a techno-economical optimisation of the oil- and gas facility without exceeding given constraints of the process and product. This is done by reducing the total energy required using the available handles: compressor speed and anti-surge valve opening. Optimisation of both, the existing equipment and newer, better sized recompression compressors have been considered. In Addition, a case study has been performed to give recommendations about future pressure depletion of the well.

A `UniSim` model has been provided by Equinor to evaluate the operation.

The report consists of four main parts, being:

1. Model fitting: Compare the received `UniSim` model with current operational data and implement battery limit into design basis.
2. Optimisation: Optimise operation based on the fitted `UniSim` model.
3. New equipment: Consider buying new compressors and give a comparison with the optimised `UniSim` model.
4. Future operation: Apply a pressure depletion of the well into the optimised `UniSim` model to give recommendations for future operation.

2 Design basis

In this section, the battery limit, feedstock specifications, constraints, and handles are defined.

2.1 Battery limit

Defining the battery limit and simplifying the `UniSim` model is the first step before optimisation and other implementations can be examined. The battery limit of the project was taken as the topside facility of the oil and gas facility only, excluding the subsea part of the plant. Also, the production water from the topside facility is outside the battery limit.

Implementation of the battery limit into the received `UniSim` model is done by removing unnecessary process equipment. For the optimisation part of this project, the focus is the compressor trains. Therefore, the streams into the first stage gas-liquid separation vessel are considered to be constant (the mean is taken from the incoming streams).

2.2 Feedstock specifications

Table 2.1 shows the mean composition of the inlet streams, extracted from the received `UniSim` model. The three inlet streams are Manifold A, Manifold B and Test Manifold, going into the first stage separator unit. The composition for each stream is given in table B.1, B.2 and B.3 in appendix B.i. ”Others” include all the heavier hydrocarbons.

Table 2.1: Mean feed compositions for Manifolds A, B and Test.

Compound	Mole fraction	Vapour phase	Liquid phase
H_2O	0.1013	0.0034	0.0015
Nitrogen	0.0050	0.0058	0.0004
CO_2	0.0386	0.0437	0.0172
Methane	0.6753	0.7688	0.1478
Ethane	0.0878	0.0983	0.0709
Propane	0.0472	0.0509	0.0939
i-Butane	0.0067	0.0069	0.0243
n-Butane	0.0128	0.0127	0.0596
Others	0.0252	0.0095	0.5845

Table 2.2 shows the mean properties of the inlet streams into Manifolds A, B and Test in the first stage separator. The properties of each of the inlet streams are given in table B.4 in appendix B.i.

Table 2.2: Mean feed properties for Manifolds A, B and Test.

Properties	Mean
Molecular weight M [g/mol]	23.69
Standard volumetric flow rate ϕ_v (STD) [Sm^3/h]	296561.08
Average liquid density ρ_{liq} [$kmol/m^3$]	17.25
RVP at 37.8 °C [kPa]	1001.85
TVP at 37.8 °C [kPa]	4432.47
Pressure [kPa]	4901.00
Temperature [°C]	52.28

2.3 Product specifications

Table 2.3 shows the composition of the gas export stream going to Åsgard B, extracted from the received `UniSim` model. ”Others” include all the heavier hydrocarbons.

Table 2.3: Composition of exported gas to Åsgard B.

Compound	**Mole fraction**	**Vapour phase**	**Liquid phase**
H_2O	0.0009	0.0005	0.0004
Nitrogen	0.0056	0.0056	0.0009
CO_2	0.0442	0.0443	0.0308
Methane	0.7566	0.7577	0.2848
Ethane	0.1028	0.1028	0.1298
Propane	0.0585	0.0583	0.1768
i-Butane	0.0083	0.0082	0.0459
n-Butane	0.0151	0.0149	0.1080
Others	0.0080	0.0076	0.2226

Table 2.4 shows the properties of the gas export stream going to Åsgard B.

Table 2.4: Properties of exported gas to Åsgard B.

Properties	
Molecular weight M [g/mol]	21.9070
Standard volumetric flow rate ϕ_v (STD) [Sm^3/h]	234470.5090
Average liquid density ρ_{liq} [kmol/m^3]	16.6091
Pressure [kPa]	6798.5786
Temperature [°C]	22.2798

Table 2.5 shows the composition of the oil product from Åsgard A. ”Others” include all the heavier hydrocarbons.

Table 2.5: Oil product composition.

Compound	**Mole fraction**	**Vapour phase**	**Liquid phase**
H_2O	0.0007	0.0006	1.0000
Nitrogen	0.0000	0.0000	0.0000
CO_2	0.0001	0.0001	0.0000
Methane	0.0003	0.0003	0.0000
Ethane	0.0017	0.0017	0.0000
Propane	0.0209	0.0209	0.0000
i-Butane	0.0209	0.0209	0.0000
n-Butane	0.0704	0.0704	0.0000
Others	0.8848	0.8849	0.0000

Table 2.6 shows the properties of the oil product from Åsgard A.

Table 2.6: Oil product composition.

Properties	
Molecular weight M [g/mol]	125.7154
Standard volumetric flow rate ϕ_v (STD) [Sm^3/h]	49370.4049
Average liquid density ρ_{liq} [$kmol/m^3$]	6.1205
RVP at 37.8 °C [kPa]	78.4105
TVP at 37.8 °C [kPa]	101.2025
Pressure [kPa]	106.3250
Temperature [°C]	33.2447

2.4 Constraints

All thermodynamic properties in the `UniSim` model provided were held constant for all analysis. The battery limit conditions that needed to be satisfied for converging all the simulation cases were agreed upon.

Feeds going into the 1^{st} stage separator (Streams Manifold A, Manifold B and Test Manifold) were taken as inputs for the study. Any change in these input flow conditions were treated as a separate case for the production profile expected from the ageing wells (pressure depletion case study).

The export gas pressure from compressor unit 27KA500 to Åsgard B has to be maintained constant at 67.9 bar(a). The gas sent for reinjection has to be maintained at 402.9 bar(a). This was done by implementing adjusters to the `UniSim` model. There was no constraint on the molar flow on these lines and hence they have been used as handles to optimise the respective compressors.

The pressure in the condensate tank is maintained at 1.063 bar(a), the same as in the received model. The RVPE of the condensate in tank has an upper limit of 11.5 psi (0.793 bar(a)), and the RVPE is also a sales product specification of the condensate.

The heat exchangers in the plant used cooling water with 35% TEG as the cooling medium. Since it was known that these exchangers are oversized for the current operating conditions, the exit temperature controllers dictate the outlet temperatures and were kept constant and equal to the received model.

The compressors have to operate within the available compressor speeds and to the right of the surge control line (defined as 10 % from the first point in the performance test curves). Compressors that are modelled individually in the model, but which share the same shaft in the plant have the same operating speeds in the model.

The received specifications and constraints can be found in table 2.7.

Table 2.7: Åsgard A specifications and constraints, received from Equinor on 30.09

Variable	**Specifications**	**Notes**
Oil product		
RVPE	11.5 psi	RVP Equivalent
Compressor shafts		
23KA500, 23KA501		Same shaft and motor
23KA502		Individual shaft and motor
26KA500		Individual shaft and motor
27KA601, 27KA602		Back-to-back, same shaft
27KA701, 27KA702		Back-to-back, same shaft
Turbines		
26KA601 26KA602	29.5 MW	Max. power, Direct operation
26KA701 26KA702	29.5 MW	Max. power, Direct operation

23KA500A/B 23KA500 23KA501 23KA502 27KA500	41 + 39 MW	Generator turbines
Cooling		
Cooling medium	TEG [i] 65 Wt%	Freshwater 35 Wt%
23KA500, cooler	25 °C	Max. cooling before compressor
23KA501, cooler, low flow	51 °C,	Max. cooling before compressor
23KA501, cooler, high flow	61 °C	Max. cooling before compressor
23KA502, cooler	44 °C	Max. cooling before compressor
26KA500, cooler	24 °C	Max. cooling before compressor
27KA601/701, cooler	24 °C	Max. cooling before compressor
27KA602/702, cooler	44 °C	Max. cooling before compressor

2.5 Handles

To optimise the process, it is to important to know what operating settings can be changed, and which must be kept constant. In the plant all compressor are run by variable speed motors, so the motor speeds of the compressors can be considered as handles. Since compressor 23KA500 and 23KA501 are on the same shaft, the speed for these compressors will only be one handle. The same is the case for the reinjection units as the compressor are on a common shaft there as well. One of the major source of power usage is the anti-surge system. This consumption can be reduced as the anti-surge valves can be adjusted, and can be considered as handles.

In the `UniSim` model the speeds of the compressors are calculated from the compressor curves, and cannot be manually adjusted. This means some other parameters must be used as handles in the `UniSim` model, which affect the speeds. From a compressor curve diagram it is apparent that the speed of a compressor is determined by the volumetric flow rate and the polytropic head. The inflow to the compressors are partially determined by much of is recycled through the anti-surge valve. The anti-surge valves are modelled as splits with split factors, instead of valves with a valve opening percentages. For the case studies these split factors will be used as handles. From equation 4.4, polytropic head is a function of suction pressure and

[i] Triethylene glycol

discharge pressure. The discharge and suction pressures of the compressors can be used as handles, as long as all constraints are met.

For the reinjection units the suction pressure for 26KA601/701 is specified by the pressure of the first stage separator. This pressure will henceforth be referred to as the first stage pressure, and will therefore not be changed in the optimisation cases of this project. The discharge pressures of compressors 26KA602/702 in the reinjection units are also constraints that must be met. Since compressors 26KA601/701 and 26KA602/702 are on the same shaft, the intermediate pressure must be met and is not an independent handle. For compressor 27KA500 the suction pressure is the first stage pressure which is not a handle, and the discharge pressure is set by the outlet conditions, so it is a constraint which must be met.

For compressor 23KA502 the discharge pressure is set by the first stage pressure and is not a handle. The suction pressure is set by the gas pressure from the second stage separator, or second stage pressure as it will be referred to henceforth. This pressure is not defined and can be adjusted as long as the constraints of the system are met, so the second stage pressure will be used as a handle for the case studies. The discharge pressure of compressor 23KA501 is also set by the second stage pressure. The suction pressure of compressor 23KA500 is set by the gas pressure from the third stage separator, or third stage pressure as it will be referred to henceforth. This pressure is also not defined, and will be used as a handle. The intermediate pressure between compressors 23KA500 and 23KA501 is not a handle as these two compressors are on the same shaft.

To summarise, the handles in the plant are the anti-surge valves for the compressors, and speeds of the compressors. In the `UniSim` model the handles which will be manipulated are the split factors splitting the outlet stream of each compressor, and the third and second stage pressures.

3 Process description

The task of this project is to find a more energy efficient operating condition for the topside facility of Åsgard A. There seems to be an energy efficiency improvement opportunity in operations for Equinor Åsgard A, since the flow rate from the wells have decreased over the years compared to the initial start of the plant. To optimise the topside facility, a `UniSim` model of the entire facility was provided by Equinor. This model contains both the subsea part and topside part of the facility. As described in the battery limit, section 2.1, the subsea part, wells, and production water are out of the scope of this project. The first task was to become familiar with the model and start simplification of the `UniSim` model considering the scope of the project. To describe the process, a process flow diagram (PFD) of the topside oil and gas facility for the battery limit is demonstrated and explained in this chapter.

3.1 Main process equipment

The main equipment in the Equinor topside facility, with symbols from PFD:

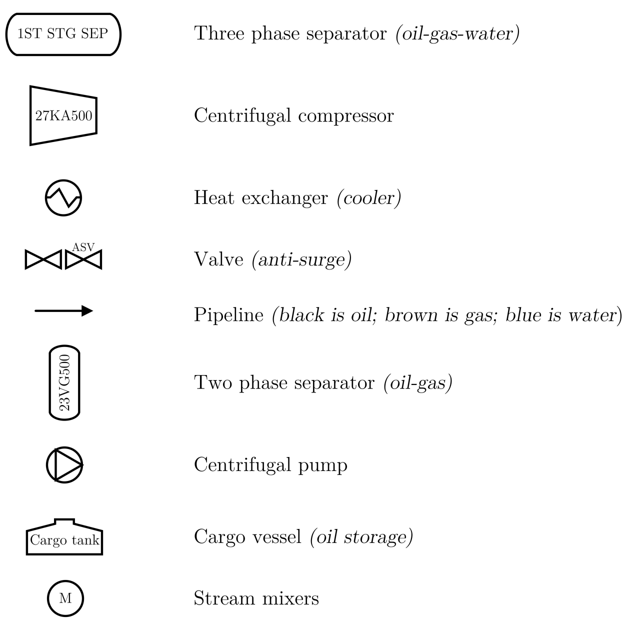


Figure 3.1: Symbol list for the main equipment of the facility, made with MS Visio.

3.2 Process flow diagram

The process flow diagram of the Equinor topside facility, modified for this project scope, is demonstrated in figure 3.2 below:

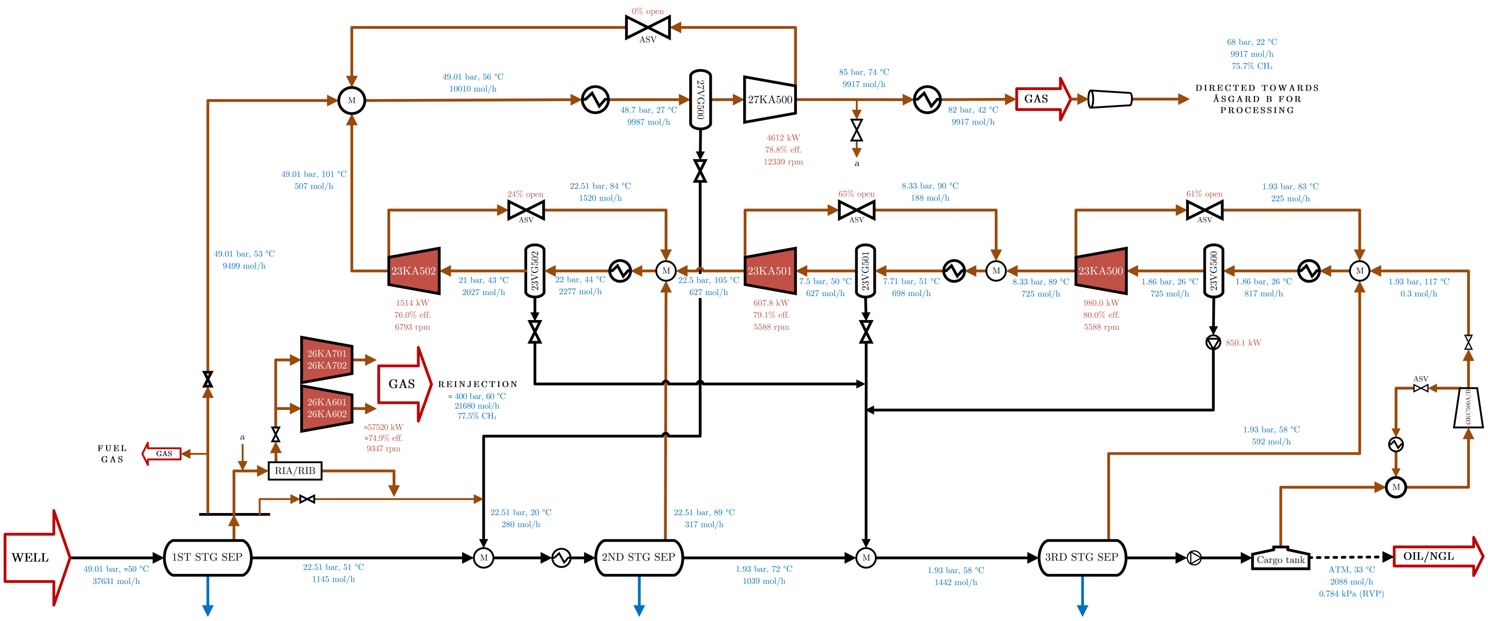


Figure 3.2: Process flow diagram of the Equinor oil and gas facility, adjusted for the battery limit of the project, and made with MS Visio.

From the Åsgard A field there are several wells that are recovering a blend of heavy-, light- and intermediate hydrocarbons, including water. For the scope of the project, the properties of the well stream into the 1st stage separator are constant, and based on the plant operation on 06.01.17 (as featured in the `UniSim` file). Averages are given in table 2.2.

The well streams enter the first stage separator unit, which contains three stages of oil-gas-water separators. Most of the gas, brown pipeline going up, goes to the reinjection train (RIA and RIB) where the gas is compressed, using two identical compressor trains, to around 400 bar(a). This gas is reinjected into the well to improve the production by a so called 'lift' phenomenon, where the density of the liquids is decreased. Most of the remaining gas from the first stage separator is sent to the 27KA500 compressor (export compressor). The remaining gas is sent to the second stage separator unit. The oil from the first stage separator is expanded and sent to the second stage separator. The process contains three separator stages which are labelled as 1st, 2nd and 3rd stage separators. In each separator the inflow is separated into a gas, liquid and water flow. The water streams from each separator is removed from the process (outside of project scope).

The recompression stage, consisting of compressors 23KA500/501/502, increases the pressure of the gas coming from the 2nd and 3rd stage separators to match the gas pressure of the 1st stage separator gas going to the export compressor. Each recompression stage consists of a cooler, two phase separator, centrifugal compressor and an anti-surge valve. Before compression, the gas needs to be cooled in a heat exchanger, and remaining liquids have to be removed to prevent damaging of the compressor unit. After compression the gas is split into two streams, one going back to the same compressor to prevent surge (see section 4.1) and another one going to the next recompression unit. The recompression compressors are driven by electrical motors. Compressors 23KA500 and 23KA501 are driven by one motor and shaft, and therefore, the speed of these compressors are equal. Compressor 23KA502 is driven by its own motor and shaft.

The final liquid stream out of the third stage separator unit is the oil product, which is stored in the cargo tank until unloaded from the platform. The oil in the cargo tank has a specific vapour pressure (RVPE measurement), and the so called 'boil-off' gas is removed from the tank to prevent pressure build up. This 'boil-off' gas is mixed into the recompression stage.

In the topside facility there are a total of nine compressors. All compressors in this process are centrifugal compressors. Three compressors recompressing the gas streams from the second- and third stage separator units, four compressors for reinjection into the well, a small compressor for the 'boil-off' from the cargo tank, and one compressor for export of the gas to Åsgard B for further refining of the gas. The export compressor ensures that the final gas pressure specification from Equinor is maintained. Hence, it is not a handle for optimisation of the plant. However, in the first operational years of the facility the flow rate from the wells was higher than the current ones. Therefore, the compressors have become oversized and surge has to be prevented.

4 Theoretical background

In this section, a theoretical background is given on the most important aspects of this project. The subjects that are discussed are: compressors, anti-surge control, and Reid vapour pressure.

4.1 Compressor theory

A compressor is a device that increases the pressure of a gas stream by applying mechanical work ($\dot{W}_s$), as illustrated in figure 4.1 [2]. The Equinor facility uses centrifugal compressors, whose shafts are driven by gas turbine(s). In a gas turbine, fuel gas is converted to mechanical work.

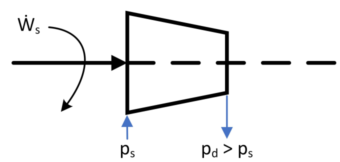
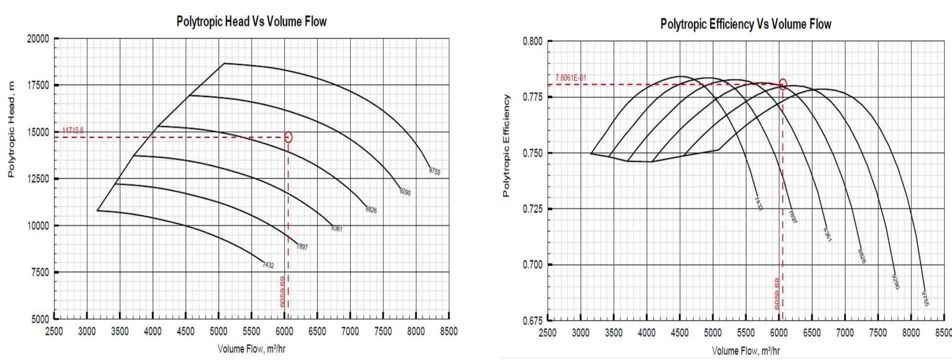


Figure 4.1: Compressor[2]

4.1.1 Compressor performance curves

Compressor performance curves illustrates relations between the volumetric flow rate of the compressor (x-axis), and the polytropic efficiency or polytropic head (y-axis) for different operational speeds of the compressor [3].

An example of the compressor performance curves is given in figure 4.2.



(a) Polytropic head vs Flow.

(b) Efficiency vs Flow.

Figure 4.2: Compressor 23KA502; Polytropic head and efficiency curves, from Equinor.

The compressor performance curves example in figure 4.2 consist of a polytropic head versus volumetric flow rate (figure 4.2a) plot, and a polytropic efficiency versus volumetric flow rate plot, which both are collected from the Dresser-Rand file for compressor 26KA601.

The compressor performance curves give the operating range of the compressors for given compressor characteristics. Looking at figure 4.2, curves are given for six different compressor speeds [rpm]: 7432, 7897, 8361, 8826, 9290, 9755. Here, the volumetric flow rate ($x-$axis) is in cubic meters per hour, polytropic efficiency ($y-$axis) is in fraction, and polytropic head ($y-$axis) is in meters.

From figure 4.2, the operational point can be found as:

- Polytropic head; 14715.8 m
- Polytropic efficiency; 78.1 %
- Volumetric flow rate; 6059.7 m^3/h

Compressor head is a way of expressing the mechanical energy needed to do the compression per unit weight of the fluid [4]. Therefore, the h_s, shaft head of a compressor can be expressed as in equation 4.1.

$$h_s = \frac{\dot{W}_s}{\dot{m}g} \tag{4.1}$$

Here, $\dot{W}_s$ [W] is the shaft work, $\dot{m}$ [kg/s] is the mass flow rate through the system, and g is the gravitational acceleration.

In general, the efficiency of a compressor can be expressed as [2]:

$$\dot{W}_s = \frac{\dot{W}_s^{\text{rev}}}{\eta} \tag{4.2}$$

W_s is the real compressor work, $\dot{W}_s^{\text{rev}}$ is the useful work done while compressing the fluid, and η is the efficiency. The energy corresponding to $\dot{W}_s - \dot{W}_s^{\text{rev}}$ is lost work and goes into heating of the discharge stream [2].

For a compressor with variable-speed drive technology, the operational range of the compressor can be at any point as long as it falls between the surge- and stonewall line and doesn't go past the operational speed limit of the compressor (design characteristic of the compressor).

Surge happens as a consequence when the flow rate is too low at the inlet of the compressor. This results in a higher pressure at the discharge of the compressor compared to the inlet, leading to a momentary flow reversal. As a consequence, the discharge pressure decreases and the compressor continues to deliver gas, reversing the flow direction. This causes pulses in the outlet pressure and flow rate making the compressor aerodynamically unstable, which is know as surge [5]. Surge takes place when the combination of head and/or efficiency causes the operational point to go past the left side of the compressor curves (vertical full line in figure 4.2a represents the surge line). In industry, a safety margin of 10% from the original surge line is taken as the maximum operational point for optimisation. This safety margin is therefore taken into account in the `UniSim` optimisation model.

Stonewall happens as a consequence from the flow rate being too high at the inlet of the compressor, and therefore, the facility becomes unstable. Since the well pressure is depleting, the flow rate will never reach this state for the current compressor layout. The Stonewall line is present at the right side of figure 4.2a.

4.1.2 Polytropic processes

A polytropic process is a process where equation 4.3 is true.[2, 6]

$$pV^n = C, \qquad 0 \leq n \leq \infty \tag{4.3}$$

p is pressure and V is volume. The exponent n may take any value between 0 and infinity, and the value depends on the process of interest [6]. Certain values of n yields the general cases as showed in table 4.1.

n	**Process**	**Characteristics**
0	Isobaric	Constant pressure
1	Isothermal	Constant temperature
$+\infty$	Isochoric	Constant volume
γ	Adiabatic and isentropic[7]	No heat flow, and constant Entropy (assumed ideal gas).

Table 4.1: Special cases for certain values of the polytropic exponent n from equation 4.3. γ is the heat capacity ratio $\gamma = \frac{C_p}{C_v}$, C_p being the heat capacity at constant pressure and C_v being the heat capacity at constant volume.

Most real world compression and expansion processes can be described as processes where exchange of both heat and work takes place, which is a combination of an isothermal and an adiabatic process. This is a polytropic process [8], where the polytropic exponent n takes other values than the values corresponding to an *iso*-state showed in table 4.1.

The real polytropic head is shown in equation 4.4 (Derivation in appendix D).

$$h_{s,\text{poly}} = \left(\frac{n}{n-1}\right)\frac{T_s Z_s R}{M g \eta_{\text{poly}}}\left[\left(\frac{p_d}{p_s}\right)^{\frac{n-1}{n}} - 1\right] \tag{4.4}$$

Equation 4.4 gives head in units of meters. Since the variable head h may have many different units, some conversions may me made. Conversion formulas are given in appendix C.

4.2 Anti-surge control

The anti-surge valves represent major energy losses, as they take the stream that was just compressed and decompresses it to the suction pressure of the compressor. The ASVs should generally be closed for more efficient production, and the current operation mode can benefit greatly from optimisation of the ASV openings.

The anti-surge control is an important feature for compressor units. When the flow into the compressor decreases too much, the discharge pressure becomes too large, forcing the flow to change directions back into the compressor. This is known as surge. An anti-surge valve is therefore included after the compressor, to ensure circulation of the gas in the desired direction.

The gas export facility at Åsgard has been in production since 2000, and the pressure in the wells is not as large as it was when the compressor train was installed [1]. The plant uses pressure depletion and partly injection of gas fro production. The decreasing well pressure causes the flow rate into the separation and compression facility to also decrease over time. The compressors installed in Åsgard A today are built for higher flow rates, making it a necessity to operate with open anti-surge valves to avoid surge. A set of new, smaller compressors may be a good solution for the plant, as the energy saved from being able to operate with open anti-surge valves to a larger extent will be great.

4.3 Reid vapour pressure

Reid vapour pressure, or RVP, is a common measurement for the volatility of hydrocarbons in liquid phase. The RVP is defined as the pressure exerted from a liquid mixture, in absolute units, at 100 °F = 37.8 °C and with a ratio of 4 between vapour and liquid. True vapour pressure, TVP, is given at a specific temperature, and is defined as the vapour pressure of a mixture when the vapour and condensate are in equilibrium. Because RVP can be used to find an estimated TVP at any temperature, RVP is a common specification criterion for blends of petroleum products [9].

The RVPE, Reid vapour pressure equivalent is an alternative measuring technique of the RVP and takes into account the temperature difference that is present in the oil storage vessel [10].

5 Model fitting

In this section, the `UniSim` model and operational data, received from Equinor, is fitted. The result is a model that can be used for case studies.

5.1 Received data

A `UniSim` model of the entire topside and subsea facility was received from Equinor. This model is the foundation for the simulations performed in this study. Operational constraints are given by Equinor, as can be seen in table 2.7. These values must be satisfied when case studies are performed. Equinor provided compressor performance curves and a screenshot from the current operation of the facility, which are compared and if needed implemented.

5.2 `UniSim` modification

Improvements are made to convert the received `UniSim` model in order to satisfy the battery limit that is specified in section 2.1. Furthermore, current operational data, consisting out of compressor performance curves and a screenshot of the stable operation were compared.

The compressor performance curves are obtained in pdf-format, and are digitised and used in an excel sheet. The data between the compressor performance curves of the `UniSim` model and operational data fit quite well. The operational compressor performance curves have more sampling points than the `UniSim` ones, but since the curves in `UniSim` were already correct, no new points have been added to the simulation file.

The inlet- and outlet temperatures of the heat exchangers in the received `UniSim` model from Equinor, originally specified as temperature differences (ΔT), are adjusted to satisfy the outlet temperature constraints (table 2.7).

Several recycle blocks in the `UniSim` model are removed to eliminate mass balance errors. The sensitivities of the remaining recycle blocks are tightened.

5.3 Received UniSim model versus operational data

Figure 5.1 illustrates a screenshot of the plant under stable operation conditions. The UniSim model is mostly based from data represented here.

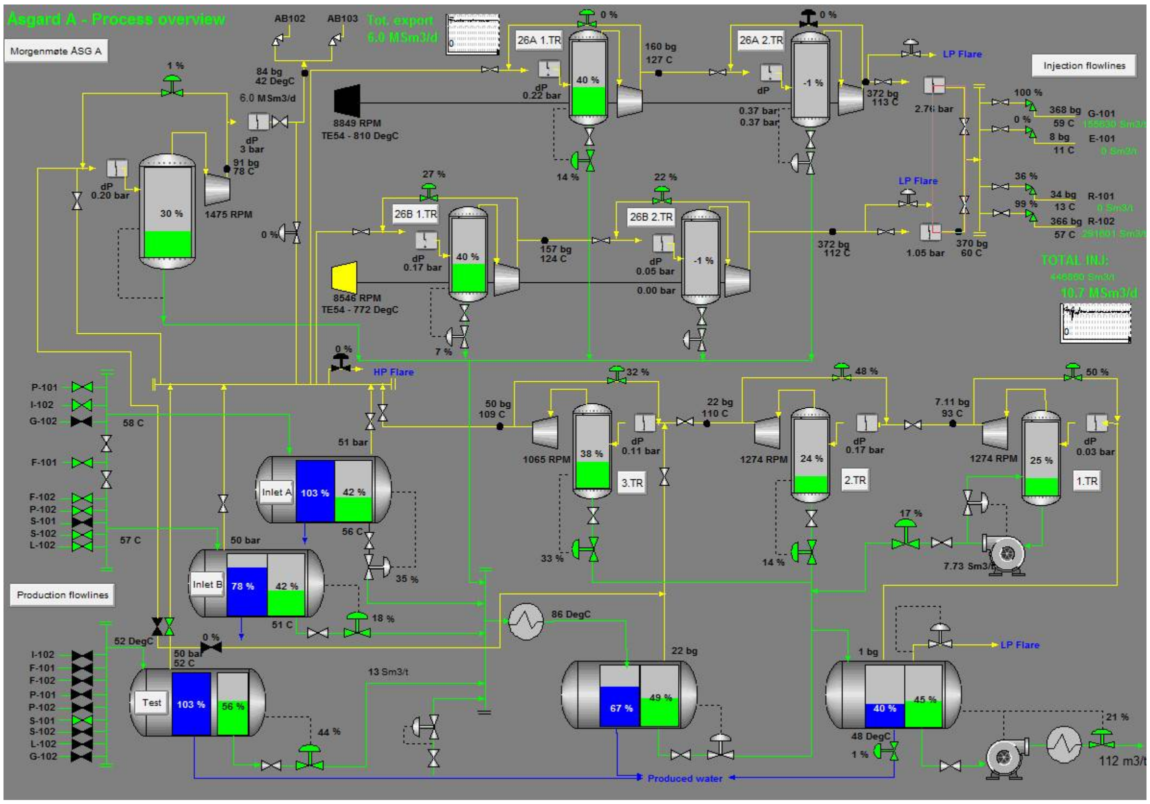


Figure 5.1: Screenshot of plant operation, received September 21, 2018.

The operational data in figure 5.1 is compared to the base model. The comparison between both is shown in table 5.1. Most of the data points between the operational data and base model data are similar. Small errors in the operating points are within the measurement error tolerances. The major exception is the compressor speeds of the recompression train which deviate significantly, this is assumed to be caused by missing gearboxes in the plant. A check is done to confirm this, using the compressor speeds from the UniSim base model in power calculations yields. Observed is a similar power consumption as the operating data. This gives more confidence about the true compressors speeds.

Comparison between the operational data and received model:

Table 5.1: Given operation data compared to the `UniSim` base model received from Equinor. | * Mismatch due to the fact that rpm is measured in plant before the gearbox.

	Operation point	**Base model**
Separator discharge temperature		
1^{st} stage (vapour, mean)	52 °C	52.28 °C
Separator discharge pressures		
1^{st} stage (vapour)	51 bar(a)	49.01 bar(a)
2^{nd} stage (vapour)	23 bar(a)	22.51 bar(a)
3^{rd} stage (vapour)	2 bar(a)	1.933 bar(a)
Compressor speeds		
23KA500*	1274 rpm	5588 rpm
23KA501*	1274 rpm	5588 rpm
23KA502*	1065 rpm	6793 rpm
26KA601, 26KA602 (RIA)	8849 rpm	9347 rpm
26KA701, 26KA702 (RIB)	8546 rpm	9347 rpm
27KA500*	1475 rpm	12339 rpm
Compressor discharge pressures		
23KA500	8.1 bar(a)	8.33 bar(a)
23KA501	23 bar(a)	22.51 bar(a)
23KA502	51 bar(a)	49.01 bar(a)
Inlet conditions		
Inlet A, temperature	58 °C	50.44 °C
Inlet B, temperature	57 °C	56.25 °C
Test, temperature	52 °C	50.16 °C
Inlet A, pressure	51 bar(a)	49.01 bar(a)
Inlet B, pressure	50 bar(a)	49.01 bar(a)
Test, pressure	50 bar(a)	49.01 bar(a)
Injection conditions		
RIA (26A), int. temperature	127 °C	138.5 °C
RIA (26A), dis. temperature	113 °C	113 °C
RIB (26B), int. temperature	124 °C	138.5 °C
RIB (26B), dis. temperature	112 °C	113 °C
RIA, int. pressure	158 bar(a)	173.4 bar(a)
RIA, dis. pressure	373 bar(a)	402.9 bar(a)
RIB, int. pressure	158 bar(a)	173.5 bar(a)
RIB, dis. pressure	373 bar(a)	406 bar(a)

Figure 5.2 illustrates the data of the oil- and gas production from July 2017 to July 2018 of the Åsgard facility.

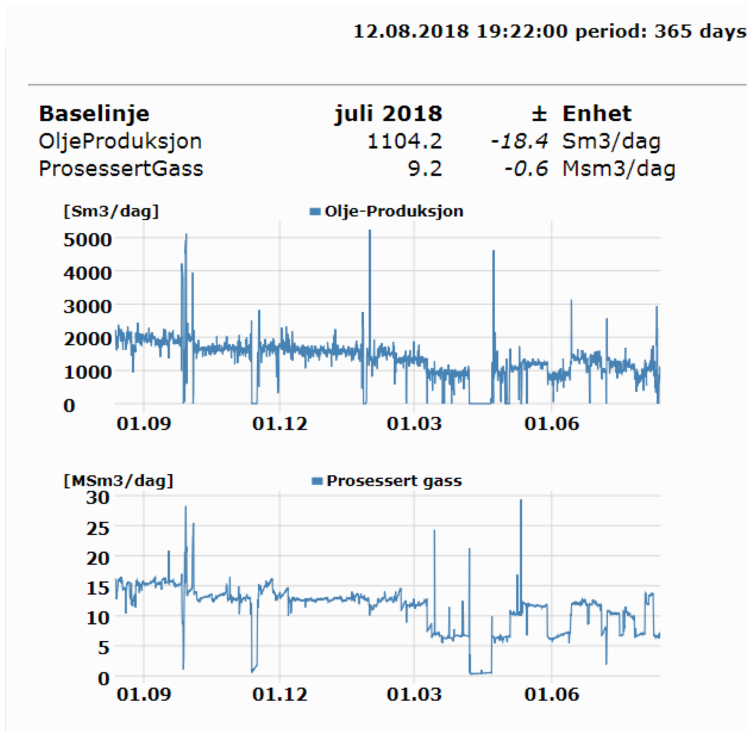


Figure 5.2: Screenshot from Equinor *Energiportalen*, August 12 2018.

Both the oil- and gas production in the `UniSim` model differ with a factor around two from the data shown in *Energiportalen* - Equinor's databank. Since most process parameters like pressures and temperatures matched with the data in the `UniSim` model, the deviation could be accounted for due to slight composition differences or differences between the volumetric inflows. No data is available for the fluid compositions or flow rates of the current operation, and therefore, the model is not modified to fit the outlet flow conditions illustrated in figure 5.2. Rather this is used to calculate the relative change in parameters of interest with the provided existing compositions. It should also be noted that the production of oil and gas changes all the time in the plot, however the trend is that both oil- and gas production is decreasing. Meaning, that if the model was fitted to the production rates of July, the production rates in November might be quite different and results will change of the modelling and optimisation.

6 Case study: Optimising current equipment

In this section the current equipment of the oil- and gas facility is optimised by performing several case studies.

6.1 Case studies

In the reinjection compressor units (26KA601/602/701/702) the different cases are done by decreasing the split factor by steps of 5% (closing the anti-surge valve) until the surge control line is reached. At this point the most amount of energy is saved. One would ideally want to completely close the anti-surge valves, however this could lead to the compressor going into surge.

For the recompression train (23KA500/501/502), different case studies are done changing the second- and third stage pressure, and lowering the anti-surge valve openings using the split factors. Constraints here are to not reach the surge control lines of the compressor, and to keep the RVP of the oil stream below 11.5 psi.

Data from case studies in the `UniSim` model is imported into an Excel verification sheet to analyse and organise obtained data. The results of each case study are compared to the base case, to ensure that all constraints are met. The plotted compressor performance curves are checked in the verification sheet to see if it is within the operational boundaries.

6.2 Optimal Operating Conditions

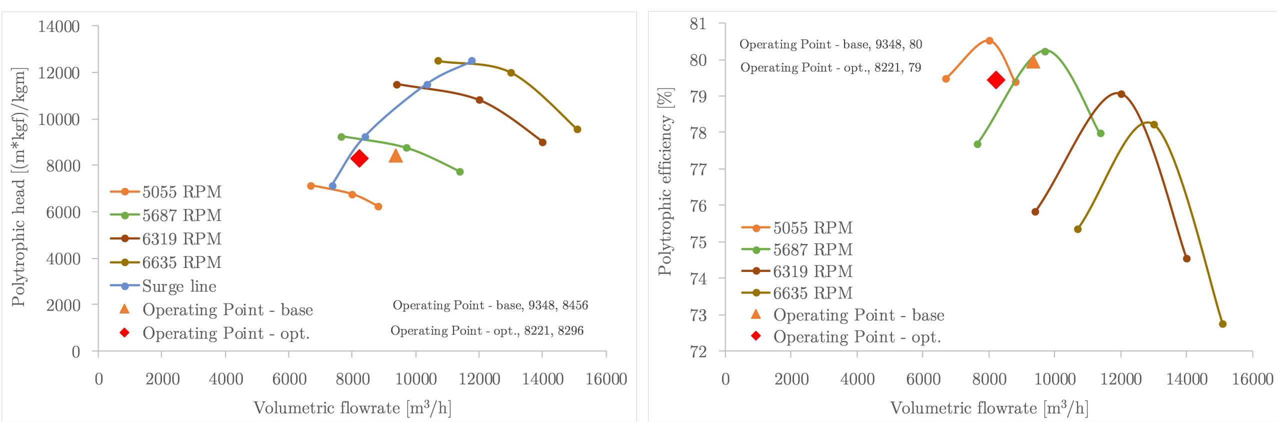
The optimal operation conditions is found from doing before mentioned case studies, and the optimised handles are given in table 6.1.

Table 6.1: Optimal operation conditions from case studies and base operation conditions.

	Base case	**Reinjection units optimised**	**All optimised**
23KA500 split factor (0-1)	0.31	0.31	0.20
23KA501 split factor (0-1)	0.30	0.30	0.25
23KA502 split factor (0-1)	0.75	0.57	0.71
26KA601/701 split factor(0-1)	0.2	0	0.00
26KA602/702 split factor (0-1)	0.2	0.07	0.07
2nd stage pressure [kPa]	2251	2251	2161
3rd stage pressure [kPa]	193.3	193.3	193.3
23KA500/501 speed [rpm]	5588	5588	5469
23KA502 speed [rpm]	6793	6793	6787
26KA601/701/602/702 speed [rpm]	9347	9156	9156

In the reinjection compressors the anti-surge valves could be closed significantly, where the 26KA601/701 compressors could be closed fully. For the recompression train, the anti-surge valves could not be closed significantly before reaching the compressor speed operating limit.

The plot of the compressor curve for 23KA500 is given in figure 6.1 below.



(a) Polytropic head curves

(b) Efficiency curves

Figure 6.1: Compressor 23KA500; polytropic head and efficiency curves

In figure 6.1 both the optimised operating point and base case operating point are presented into the compressor performance curves. From the figure it is possible to see that the compressor is operating close to the anti-surge control line, despite the anti-surge valve barely being closed. From the efficiency curves it can be seen that the optimised operating point has a lower polytropic efficiency than the base case operation point. This is not ideal, but the reduced mass flow will make up for a loss in efficiency. The compressor performance curves of the other compressors are given in section F in the appendix.

The reduced energy consumption for the compressors are given in table 6.2.

Table 6.2: Compressor duties for base-, optimised reinjectors- and all optimised case.

	Duty [MW]		
	Base case	**Reinjection units optimised**	**All optimised**
Total	65.23	56.01	55.69
23KA500	0.98	0.98	0.85
23KA501	0.61	0.61	0.53
23KA502	1.51	1.51	1.38
27KA500	4.61	4.61	4.62
26KA601	16.46	13.55	13.55
26KA602	12.30	10.61	10.61
26KA701	16.46	13.55	13.55
26KA702	12.30	10.61	10.61

From the optimisation of all compressors a total of 9.54 MW can be saved. Major energy savings from optimising the reinjection compressors, and minor energy saving from the recompression train are obtained.

One thing to note is that the export compressor 27KA500 uses slightly more energy compared to the base case. Since the anti-surge valve is closed in the base operation, and the discharge and suction pressures are constraints, no optimisation could be done for this compressor. However, by reducing the second stage pressure more intermediate hydrocarbons will go into the gas stream, and the export compressor must compress a slightly larger mass stream.

7 Case study: New compressors

An alternative to optimising the current operation is to consider buying new equipment. The equipment in the plant is sized for different flows than the ones in the current flows. Buying more adequately sized compressors could save more power, however, the cost of buying new equipment might outweigh the money saved from more efficient operation. With new compressors, the operating conditions can be changed to push more intermediates into the oil, which will improve the profitability of the plant, due to oil being more profitable than natural gas. However, the RVP specification of the oil must be within given constraints.

7.1 Replacement choices

As can be seen from table 6.1, for compressors 23KA500/501/502 (recompression train) the anti-surge valves can not be closed much compared to the other compressor units. For the compressors 26KA601/701 and 26KA602/702 (reinjection train), the anti-surge valves are almost fully closed in the optimisation. The reinjection compressors have their anti-surge valves almost closed, therefore, it is assumed that replacing these compressors would not lead to a great improvement. For the following case study only replacement of the recompression train is evaluated. The new compressors are assumed to operate at a polytropic efficiency of 80%.

By replacing the 23KA500/501/502 compressors, the intermediate pressures in between 23KA500/501 and 23KA501/502 becomes a handle for optimisation. As the compressor speeds are assumed to be optimal, and the anti-surge valves are closed, these will not be considered as handles in the following case study. This leaves the 2^{nd} and 3^{rd} stage pressures and the intermediate pressure between 23KA500/501 as handles, to maximise the profitability of the plant.

The revenue from the plant varies with the composition of the oil- and gas products, and the constraints in table 2.7 needs to be met. The case studies that were run in `UniSim` took these into account.

7.2 Economic pricing data

Table 7.1 shows the costing data received from Equinor, and was used in the investment analysis considering replacement of compressors 23KA500, 23KA501 and 23KA502 in section 7.4.

Table 7.1: Åsgard costing parameters.

Parameter	Value	Unit
Fuel gas price	1.567	NOK/Sm^3
CO_2 price	1.067	NOK/Sm^3
Alternative fuel cost	0.500	NOK/Sm^3
CO_2 production	2.429	kgCO2/Sm^3
CO_2 tax	1.040	NOK/Sm^3
Oil barrel volume	0.159	Sm^3
Oil/Gas equivalent	1000.0	Sm3 gas/Sm^3 oil
Compressor efficiency	0.360	fraction(0-1)
Density fuel gas	0.800	kg/Sm^3
Heating value	35.00	MJ/Sm^3

Table 7.2 shows the economic rates used in the calculations of the Net Present Value, NPV. The depreciation rate is used in calculations of depreciation on equipment [11]. The discount rate is provided by Equinor, and the tax rate is found in Altinn [12].

Table 7.2: Rates used in economic calculations.

Rate	Symbol	Used value
Depreciation rate	d_r	4 %
Tax rate	t_r	23 %
Discount rate	i	8 %

7.3 Economic calculations

7.3.1 Cost estimation

The sizing calculations for the possible new compressors are done based on the ideal energy consumption of the new, smaller compressors found during the optimisation. The calculations are done using equation 7.1 [5].

$$C = a + bS^n = 490000 + 16800 \cdot S^{0.6} \tag{7.1}$$

This equation is valid for compressor powers between 75- and 30000 kW. C is the material cost of the new equipment, S is the compressor power, and a, b and n are coefficients given in Sinnot: *Chemical Engineering Design*[5]. These are specific to centrifugal compressors.

Equation 7.1 yields the cost in a 2007 USD basis, and needs to be scaled to find the correct values.

$$C_{\text{now}} = C_{\text{then}} \frac{\text{Index}_{\text{now}}}{\text{Index}_{\text{then}}} \tag{7.2}$$

The index used is the CEPCI, the *Chemical Engineering Plant Cost Index* for pumps and compressor. The indices are found in the *Chemical Engineering Monthly Journal* [13, 14]. The indexes for October 2007[14] is 794.2, and for May 2018[13] is 1022.9.

Equation 7.1 and 7.2 yields the data given in table 7.3.

Table 7.3: Compressor sizing results.

Compressor	**Power** [kW]	**Price** [MUSD]	**Price** [MNOK]
23KA500	757	1.78	14.83
23KA501	393.4	1.41	11.71
23KA502	329	1.33	11.05
Total	1479.44	4.52	37.59

This cost only includes the price of materials, so some installation factors need to be included to reflect a more accurate price for the new compressors. This cost can be found using equation 7.3 [5].

$$C_{tot} = \sum_i C_i \left[1 + f_p + \frac{f_{er} + f_{el} + f_i + f_c + f_s + f_l}{f_m} \right] \tag{7.3}$$

Not all installation factors are used for this case, for example, it is assumed that the current pipelines of the facility are suitable for the new compressors. An overview of the possible installation factors and used installation factors is shown in table 7.4.

Table 7.4: Possible installation factors, with a specification if used.

Installation factor	**Symbol**	**Value for fluid process type**	**Included**
Piping	f_p	0.8	No
Equipment erection	f_{er}	0.3	Yes
Electrical work	f_{el}	0.2	No
Instrumentation and process control	f_i	0.3	Yes
Civil engineer work	f_e	0.3	Yes
Structures and buildings	f_s	0.2	No
Lagging, insulation and paint	f_l	0.1	Yes
Material (carbon steel = 1.0)	f_m	-	Yes

Only taking into account the used factors, equation 7.3 can be simplified to equation 7.4. The material factor is corresponding to the chosen material for the compressor. Assuming the new compressors will be made of carbon steel corresponds to a material factor of 1.0. This is what has been implemented.

$$C_{tot} = \sum_i C_i \Big[1 + f_{er} + f_i + f_c + f_l \Big] \tag{7.4}$$

Since the new equipment are three centrifugal compressors, the installation factors will be the equal for each compressor, resulting in equation 7.5.

$$C_{tot} = \Big[1 + f_{er} + f_i + f_c + f_l \Big] \sum_i C_i = 2.0 \sum_i C_i \tag{7.5}$$

Hence, the total cost of installing the new compressors is $2.0 \sum_i C_i$. In the following analysis, it is assumed that the compressors are being installed during a scheduled break in production or during a turnaround. The availability of the plant was not taken into account in the calculations, and was assumed to be 100%.

With the numbers from table 7.3, the total cost of the new compressors is approximately 75.2 MNOK.

7.3.2 Cash flows, Tax and depreciation

To determine if it is profitable to buy new compressors, the profits of the plant with new compressors must be calculated and compared with the calculated investment cost. Profits are calculated by the following equation:

$$\text{Profit} = \text{Revenue} - \text{Expenses} \tag{7.6}$$

The tax is calculated as the profits multiplied with a tax rate t_r. When a investment is made with decreasing value over time, the investment can be depreciated in the form of tax reduction. The tax rate was taken as 23% [12].

$$\text{Tax} = (\text{Profit} - \text{Depreciation}) \cdot t_r \tag{7.7}$$

A declining balance depreciation model was used to calculate the yearly depreciation. d_r is depreciation rate, taken as 4% for equipment. I_{in} is the initial investment:

$$\text{Depreciation}_{\text{in year } n} = I_{in} \cdot d_r(1 - d_r)^{n-1} \tag{7.8}$$

CO_2 tax is provided by Equinor and is given in table 7.1. The tax rate is 1.04 NOK/Sm^3. The only taxes that are taken into account is the cash flow- and the CO_2 tax. The CO_2 tax is considered an expense in this project.

7.3.3 Investment analysis

With new compressors being sized, a Net Present Value, NPV, calculation is done to determine whether it is profitable to change the compressors, or keep the old ones. The NPV calculation considers the cost of the new compressors and installation costs as initial investment. Calculation of the NPV takes into account the cash flow CF_n in year n, given by equation 7.9.

$$CF_n = \text{Profit} - \text{Taxes} \tag{7.9}$$

The yearly cash flows are calculated according to equation 7.9.

The cash flows calculated are the additional cash flows from getting new compressors, compared to the optimised case. The additional profit is the difference between the additional revenue (from the sale of oil and gas) and the reduction in operating costs (which include fuel gas costs and the tax for CO_2).

The relative profits will be the ones compared to the the optimised case and not the base case. This is to determine if it is more profitable to change the compressors, or to optimise the operation conditions of the old compressors. The taxes will be calculated after Norwegian tax law with the tax rate given in the Norwegian tax code. As new equipment is bought, tax depreciation for this equipment will be calculated.

The NPV is calculated from equation 7.10.

$$\mathrm{NPV} = \sum_{n=1}^{N} \frac{CF_n}{(1+i)^n} - I_{in} \tag{7.10}$$

CF_n is cash flow in year n as given in equation 7.9, N is the lifetime of the project, i is interest rate and I_{in} is the initial investment.

An alternative investment analysis method is to calculate the payback time. The payback time is a measurement for how long it will take to pay back the initial investment. Payback time, PBT, is given by equation 7.11.

$$\mathrm{PBT} = \frac{\text{Investment [NOK]}}{\text{Profit [NOK/year]}} \tag{7.11}$$

7.4 Case Study

The option for replacing compressor 23K500/501/502 is analysed. New operational conditions are identified as to which would maximise the NPV of the project.

In `UniSim`, the pressures of the 2^{nd} and 3^{rd} stage separators as well as the discharge pressure after compressor 23KA500 are handles that are used for running the case studies. For the $2^n d$ stage pressure the step size was 100 kPa withing the bounds of 1965 and 2365 kPa. For the $3^r d$ stage pressures the step size was 20 kPa within the bounds of 153 and 233 kPa. For the 23KA500 discharge pressure the step size was 100 kPa, within the bounds of 615 and 1015 kPa.

Apart from the original constraints used in earlier simulations, an additional constraint is that the new compressors should have a pressure ratio below 5. This is a rule of thumb used during design as most commercially available compressors are available in this pressure ratio range. Exceeding this specification would require more exotic designs and a higher investment cost.

The optimal case is found, taking the case with highest NPV for the project with the normal oil prices used for evaluation. The optimised pressure levels that the new compressors operate on was found from this are given in table 7.5.

Table 7.5: Optimised handles.

Pressure	**Value**
p, 2^{nd} stage separator [bar(a)]	23.65
p_d, 23KA500 [bar(a)]	9.14
p, 3^{rd} stage separator [bar(a)]	1.93

Since the oil price is the major variable, the feasibility of the project was checked under a range of expected oil prices as defined by Equinor for evaluating their projects (sensitivity analysis). The gas price is assumed to be constant at its normal value as there is less fluctuation in its prices.

The fuel gas price, gas/oil equivalent and oil barrel volume are used to calculate a current oil price (normal in table 7.6), using the data provided by Equinor, given in table 7.1. This is used to calculate the viability of the project at the various oil prices.

7.5 Results

The results of the cases with normal-, high- and low oil prices is given in table 7.6.

Table 7.6: Sensitivity analysis for three oil price cases.

Oil price [NOK/barrel]	**249.20** (low)	**498.30** (normal)	**747.45** (high)
Extra Total Profit [NOK/h]	1344	1869	2394
Extra Total Revenue [NOK/h]	399	923	1448
Reduction in OPEX [NOK/h]	945	945	945
Payback period [years]	6.4	4.6	3.6
NPV is positive in year	13	8	6
NPV in year 10 [MNOK]	-10.53	13.23	36.98

With a low oil price considered, changing the compressors will yield a negative NPV after less than ten years, and not achieving a positive NPV for the 12 years which is assumed as the lifetime of the facility. The project has a positive NPV when considering a normal oil price of 60 USD (498 NOK [ii]) per barrel with a payback period of 4.6 years.

[ii] With a NOK to USD exchange rate of 8.3 taken 07.11.18. [15]

8 Case study: Well pressure depletion

The aim is to find the bottlenecks in operating with the current equipment when the well pressures and flows are expected to drop, and find potential modifications that might be required.

With ageing wells, the pressure, volumetric flow and compositions will change. The production profile data is obtained by rigorous subsurface modelling. In the absence of any such models that are available for this study, a 20% drop in pressure and molar flow by the year 2030 (interpolated linearly for the years in between) was provided as input for the evaluation. Compositions for individual manifolds are kept constant for this analysis. The result is given in table 8.1.

Table 8.1: Linear Production Profile provided for analysis.

	Base Case @ 2018	**Low Production @ 2030**
Pressure [kPa]	4901	3920.8
Total Molar Flow [kmol/h]	37627	30102
Total Inlet Gas Flow [MSTD m^3/h]	0.77	0.69
Total Inlet Oil Flow [Act m^3/h]	171.7	150.4
Total Inlet Water Flow [Act m^3/h]	72.23	64.84

The constraints for this analysis are kept the same as previous analysis with the critical parameters being the maximum speed of export compressor, compressor 27KA500, and the required landing pressure of export gas at Åsgard B.

It is noticed that with the reduced 3rd stage pressure, the export compressor is unable to maintain the same landing pressure at Åsgard B with the gas flow rate (set in TEE-102) kept the same as the base case. Flow rate into the export compressor is redirected to reinjection compressors in order to meet the constraints in 27KA500.

With the well pressure dropping to 90% of the base case, the export compressor reached the surge control line while also operating at the maximum possible speed. Any further drop in the well pressure would see the landing pressure at Åsgard B drop, which is the one of the constraints provided by Equinor for this study.

Figure 8.1 shows the compressor performance curve with operation points of the base case and the operation points of the 10% pressure reduction (◆) case.

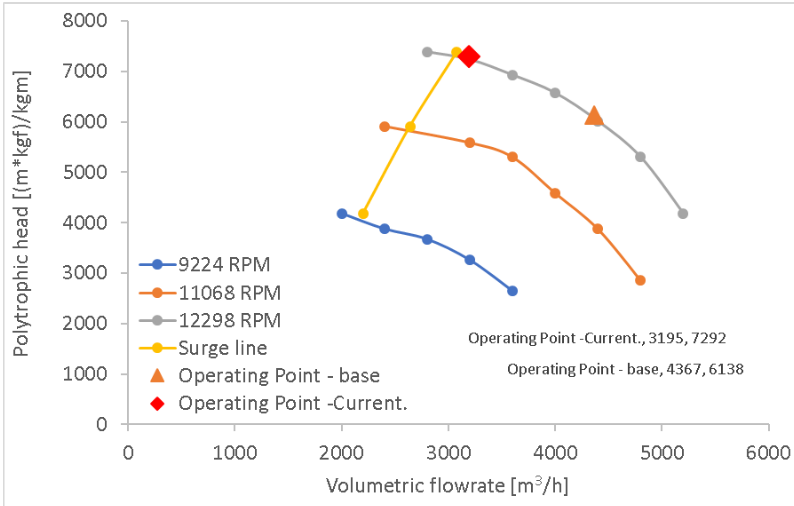


Figure 8.1: Operating point for 27KA500 with 10% drop in well pressure (◆).

Since the molar flow into the facility is reduced by 10%, the anti-surge valves must be opened more to prevent the compressors in the recompression train going into surge. This leads to increasing the inefficiency of operation in the compressor train. Anti-surge valve openings for the base case and the case with 90% of original well pressure are given in table 8.2.

Table 8.2: Fraction of flow recycled in different compressors

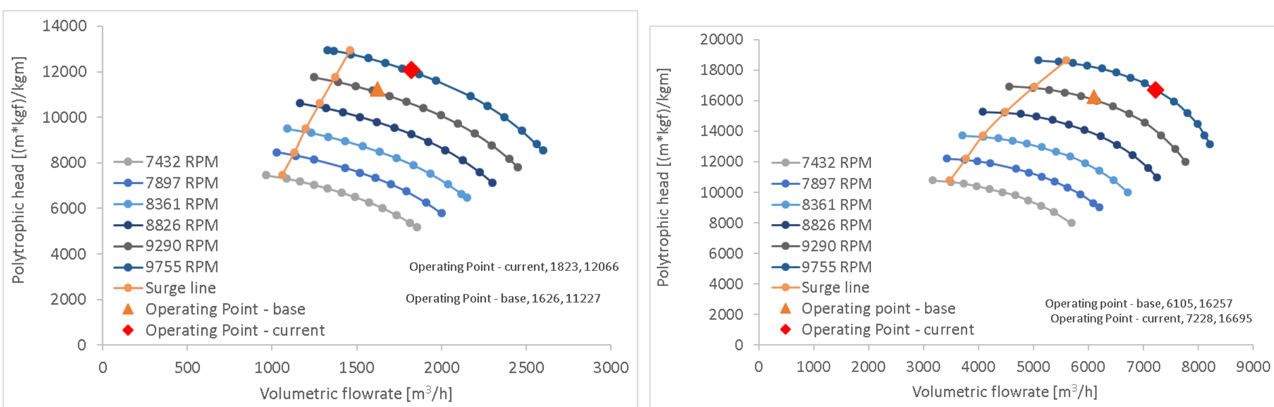
ASV Flow (Wt Fraction of total flow)	**Base case**	**90% pressure**
23KA500 ASV	0.25	0.32
23KA501 ASV	0.22	0.30
23KA502 ASV	0.71	0.80
27KA500 ASV	0.00	0.00
26KA601 ASV	0.00	0.00
26KA602 ASV	0.07	0.00
26KA701 ASV	0.00	0.00
26KA702 ASV	0.07	0.00

As can be seen from table 8.3, a decrease in pressure of well causes a reduction of oil and gas flow without any significant reduction of the total energy consumption.

Table 8.3: Power consumption and product flows

	Base case	90% pressure
Total Power Consumed [MW]	55.69	53.63
Molar Flow [kmol/h]		
To Åsgard B	9920	6432
To reinjection	21680	21680
Oil to tank	834	746

If well pressures drop below 90%, then the export compressor will be unable to maintain the landing pressure at Åsgard B. All gas produced in the facility could be sent into 26KA601/602/701/701 if needed. The reinjection compressors are able to accommodate this additional gas flow into the system if the export compressor is shutdown. This can be seen from the operating points (◆) for the compressors in figures 8.2a and 8.2b.



(a) 26KA601 **(b)** 26KA602

Figure 8.2: Operating points for compressors 26KA601/602 when all gas flow is routed to reinjection (◆).

It is not clear from the available process model if the gas from the recompression train can be rerouted to 26KA601/602/701/702 with the available piping in the plant. The potential piping modifications need to be compared to the benefits of modifications of the export compressor, as this compressor may need to be replaced later in the field life.

9 Discussion

The current operations as specified by the base model of Åsgard A provided by Equinor shows significant losses because of compressors operating with partially open recycle valves. The energy savings that could be achieved by closing the anti-surge valves and operating closer to the surge control line provides reduction in power consumption by 9.22 MW (if only the reinjection units are optimised) and by 9.54 MW (if both the recompression train and the reinjection sections are optimised).

Table 9.1: Savings by closing anti-surge valves

	Base model	**Reinjection optimised**	**All optimised**
Total Power [MW]	65.23	56.01	55.69
Power Savings [MW]	-	9.22	9.54
Monetary Savings [MNOK/year]	-	60.16	62.25

There is further scope of reducing power consumption by replacing the compressors in the recompression train (since they are still operating with partially open recycle valves to prevent these machines from going into surge) with smaller machines. The newer machines would also allow for operation of the separators under optimal pressure levels that would maximise the revenue of the plant by maximising oil production under current conditions while still meeting all the constraints specified. A constant production profile has been assumed for the economic analysis below.

Table 9.2: Sensitivity analysis of project to replace compressors in the recompression train.

Oil price [NOK/barrel]	**249.20** (low)	**498.30** (normal)	**747.45** (high)
Extra Total Profit [NOK/h]	1344	1869	2394
Payback period [years]	6.4	4.6	3.6
NPV positive in year	13	8	6
NPV in year 10 [MNOK]	-10.53	13.23	36.98

With ageing wells, the pressures and volumetric flow rates into the facility will reduce. Assuming a constant composition (No change in the increase in gas to oil Ratio as well depletes) and a linear drop in both pressure and volumetric flow, it was seen that if the pressure drops below 90% of current pressure, then the export compressor would be unable to deliver the gas at the same export pressure required. This gas could however be rerouted to reinjection as there is available capacity in the system 26. It has to be stressed that no change in the gas to oil ratio was assumed for this analysis as no detailed subsurface models were available at the time of this study. With the higher gas/oil ratio expected at the end of field life, the additional gas flows may exceed the capacity of the reinjection units and has to be considered before making a final decision.

10 Conclusion

The energy consumption in Åsgard A could be reduced by 9.22 MW by optimising the operation of compressors in the reinjection units and an additional 0.32 MW by optimally operating the compressors in the recompression train. Replacing the compressors in the recompression train would provide an additional power saving of 1.27 MW. The project has an NPV of 13.23 MNOK assuming a constant production profile at normal oil price of 60 USD per barrel. Equinor should consider the benefits of CO_2 reduction as well as the improvements in production that this project offers in comparison to similar opportunities that are available before making an investment decision.

Ageing of wells would decrease the pressure and flow into the facility. If the pressure drops below 90% of the current pressure, the export compressor would be unable to deliver gas to Åsgard B at the same landing pressure. This gas could however be redirected to reinjection instead of installing additional booster compressors for export. These results have assumed a linear drop in pressure and volumetric flow without any change in composition of fluids coming into the facility. The analysis should be conducted with updated production profiles from the subsurface team for accurate results.

Simen Bjorvand Trondheim, November 23, 2018

Anita Klavenes Trondheim, November 23, 2018

Sandeep Prakash Trondheim, November 23, 2018

Sander Wijnsma Trondheim, November 23, 2018

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A List of symbols

Symbol	**Unit**	**Description**
a	USD 2007 Basis	Costing parameter
b		Costing parameter
C	Pa$\cdot$(m^3)^n	Polytropic system constant
C	USD 2007 Basis	Material cost of new equipment
CF_n	NOK	Cash flow in year n
c_p	J/(K $\cdot$ mol)	Specific heat capacity at constant pressure
c_v	J/(K $\cdot$ mol)	Specific heat capacity at constant volume
d_r	-	Depreciation rate
η	-	Compressor efficiency
f_j	-	Installation factor, $j \in \{p, er, el, i, e, s, l, m\}$
ϕ_v	Sm^3/h	Standard volumetric flow rate
g	m/s^2	Gravitational acceleration
h	See appendix C	Head
i	-	Interest rate
I_{in}	NOK	Initial investment
γ	m/s^2	Heat capacity ratio $= {}^{c_p}/_{c_v}$
$\dot{m}$	kg/s	Mass flow rate
M	kg/mol	Molar weight
n	-	Polytropic exponent
n	years	Current year in economic calculations
N	mol	Moles
N	years	Lifetime in economic calculations
$\dot{N}$	mol/s	Molar flow rate
NPV	NOK	Net present value
R	J/(K $\cdot$ mol)	Ideal gas constant
ρ	kg/m^3	Density
RVP	Pa	Reid vapour pressure
p	Pa	Pressure
p_d	Pa	Discharge pressure
p_s	Pa	Suction side pressure
$\dot{q}$	m^3/s	Volumetric flow rate
S	(kW)	Sizing parameter, here compressor power
T	K	Temperature
T_d	K	Discharge temperature
t_r	-	Tax rate
T_s	K	Suction side temperature
TVP	Pa	True vapour pressure
V	m^3	Volume
$\dot{W}_s$	J/s	Compressor work
$\dot{W}_s^{\text{rev}}$	J/s	Reversible, useful compressor work
Z_i	-	Compressability factor of stream i

B Process parameters

B.i Feedstock specifications

Table B.1: Feed compositions for Manifold A

Compound	Mole fraction	Vapour phase	Liquid phase
H_2O	0.1720	0.0031	0.0014
Nitrogen	0.0047	0.0059	0.0004
CO_2	0.0363	0.0446	0.0178
Methane	0.6210	0.7700	0.1499
Ethane	0.0805	0.0979	0.0722
Propane	0.0429	0.0500	0.0949
i-Butane	0.0061	0.0067	0.0244
n-Butane	0.0117	0.0123	0.0600
Others	0.0248	0.0094	0.5789

Table B.2: Feed compositions for Manifold B

Compound	Mole fraction	Vapour phase	Liquid phase
H_2O	0.0583	0.0040	0.0017
Nitrogen	0.0052	0.0057	0.0004
CO_2	0.0400	0.0432	0.0164
Methane	0.7075	0.7699	0.1454
Ethane	0.0910	0.0973	0.0674
Propane	0.0486	0.0502	0.0875
i-Butane	0.0070	0.0069	0.0225
n-Butane	0.0133	0.0126	0.0551
Others	0.0290	0.0102	0.6035

Table B.3: Feed compositions for Manifold Test

Compound	Mole fraction	Vapour phase	Liquid phase
H_2O	0.0737	0.0030	0.0014
Nitrogen	0.0052	0.0057	0.0004
CO_2	0.0396	0.0433	0.0172
Methane	0.6974	0.7666	0.1479
Ethane	0.0920	0.0997	0.0732
Propane	0.0499	0.0525	0.0994
i-Butane	0.0071	0.0072	0.0259
n-Butane	0.0134	0.0131	0.0636
Others	0.0218	0.0089	0.5711

Table B.4: Manifold properties

Properties	Manifold A	Manifold B	Manifold Test
Molecular weight M [g/mol]	23.3249	24.2274	23.5017
Standard volumetric flow rate ϕ_v (STD) [Sm^3/h]	377771.1015	404772.9134	107139.2386
Average liquid density ρ_{liq} [$kmol/m^3$]	18.1962	16.6024	16.9558
RVP at 37.8 °C [kPa]	1050.2492	955.5336	999.7549
TVP at 37.8 °C [kPa]	4488.1083	4325.6482	4483.6494
Pressure [kPa]	4901.0000	4901.0000	4901.0000
Temperature [°C]	50.4427	56.2463	50.1635

C Unit conversion table for head (h) units

[y] = $\square \cdot$ [x]
$g = 9.81\text{m/s}^2$

Unit \ Unit	m	cm	kJ/kg	m^2/s^2	mm	ft	lbf-ft/lbm	m·kgf/kgm
m	-	100	$10^{-3}g$	g	1000	3.2808	3.2808	1
cm	0.01	-	$10^{-5}g$	$0.01g$	10	0.032808	0.032808	0.01
kJ/kg	10^3g^{-1}	10^5g^{-1}	-	1000	10^6g^{-1}	334.44	334.44	10^3g^{-1}
m^2/s^2	g^{-1}	$100g^{-1}$	10^{-3}	-	$1000g^{-1}$	0.3344	0.3344	g^{-1}
mm	10^{-3}	0.1	$10^{-6}g$	$10^{-3}g$	-	0.0032808	0.0032808	10^{-3}
ft	0.3048	30.48	0.00299	2.99	302.8	-	1	0.3048
lbf-ft/lbm	0.3048	30.48	0.00299	2.99	302.8	1	-	0.3048
kgf-m/kgm	1	100	$10^{-3}g$	g	1000	3.2808	3.2808	-

D Derivation of head equation 4.4

The reversible $pV-$work $\dot{W}_s^{\text{rev}}$ for a compressor can be expressed as in equation D.1.

$$\dot{W}_s^{\text{rev}} = \int_{p_s}^{p_d} \dot{q}dp \tag{D.1}$$

p_s is the suction pressure, p_d is the discharge pressure and q is the volumetric flow rate of the system. As shown in equation 4.3, $p(V)^n$ is constant for different states of a polytropic process. This means the integral in equation D.1 can be expressed as in equation D.2, with the constant term placed outside the integral boundary, this assuming n constant and not a variable of pressure, and here per units of time.

$$\dot{W}_s^{\text{rev}} = \dot{C}^{1/n} \int_{p_s}^{p_d} \left(\frac{1}{p}\right)^{\frac{1}{n}} dp \tag{D.2}$$

Here, n is the polytropic constant and $\dot{C}$ is a polytropic system constant, here in terms of flow per unit time. Solving the integral and rearranging yields equation D.3.

$$\dot{W}_s^{\text{rev}} = \dot{C}^{1/n}\left(\frac{n}{n-1}\right)\left(p_d^{\frac{n-1}{n}} - p_s^{\frac{n-1}{n}}\right) = \dot{C}^{1/n}\left(\frac{n}{n-1}\right)p_s^{\frac{n-1}{n}}\left[\left(\frac{p_d}{p_s}\right)^{\frac{n-1}{n}} - 1\right] \tag{D.3}$$

The constant C can be found using the definision of compressability factor Z together with equation 4.3. The compressability factor is defined as in equation D.4.

$$Z \hat{=} \frac{pV}{NRT} = \frac{p\dot{q}}{\dot{N}RT} \tag{D.4}$$

p is the pressure, V is the volume, $\dot{q}$ is the volumetric flow rate, N is moles, $\dot{N}$ is molar flow rate, R is the ideal gas constant and T is the temperature. Expressing equations 4.3 and D.4 in terms of V_s, the volume on the suction side of a compressor yields equation D.5.

$$\dot{C}^{1/n} = \frac{\dot{N}_s R T_s Z_s}{p_s^{\frac{n-1}{n}}} \tag{D.5}$$

Going back to the solved integral in equation D.3, plugging in the constant $C^{1/n}$ from equation D.5 leads to equation D.6.

$$\dot{W}_s^{\text{rev}} = \left(\frac{n}{n-1}\right)\dot{N}_s T_s Z_s R\left[\left(\frac{p_d}{p_s}\right)^{\frac{n-1}{n}} - 1\right] \tag{D.6}$$

To transfer the equation into an equation in terms of mass, the molar flow rate $\dot{N}$ can be expressed as in equation D.7.

$$\dot{N} = \frac{\dot{m}}{M} \tag{D.7}$$

$\dot{m}$ is the mass flow rate, and M is the molar weight of the fluid. Substituting with $\dot{N}_s$ in equation D.6 yields equation D.8.

$$\dot{W}_s^{\text{rev}} = \left(\frac{n}{n-1}\right)\frac{\dot{m}_s T_s Z_s R}{M}\left[\left(\frac{p_d}{p_s}\right)^{\frac{n-1}{n}} - 1\right] \tag{D.8}$$

Now looking at the relationship between head and work from equation 4.1, the polytropic head can be expressed as in equation 4.4[16].

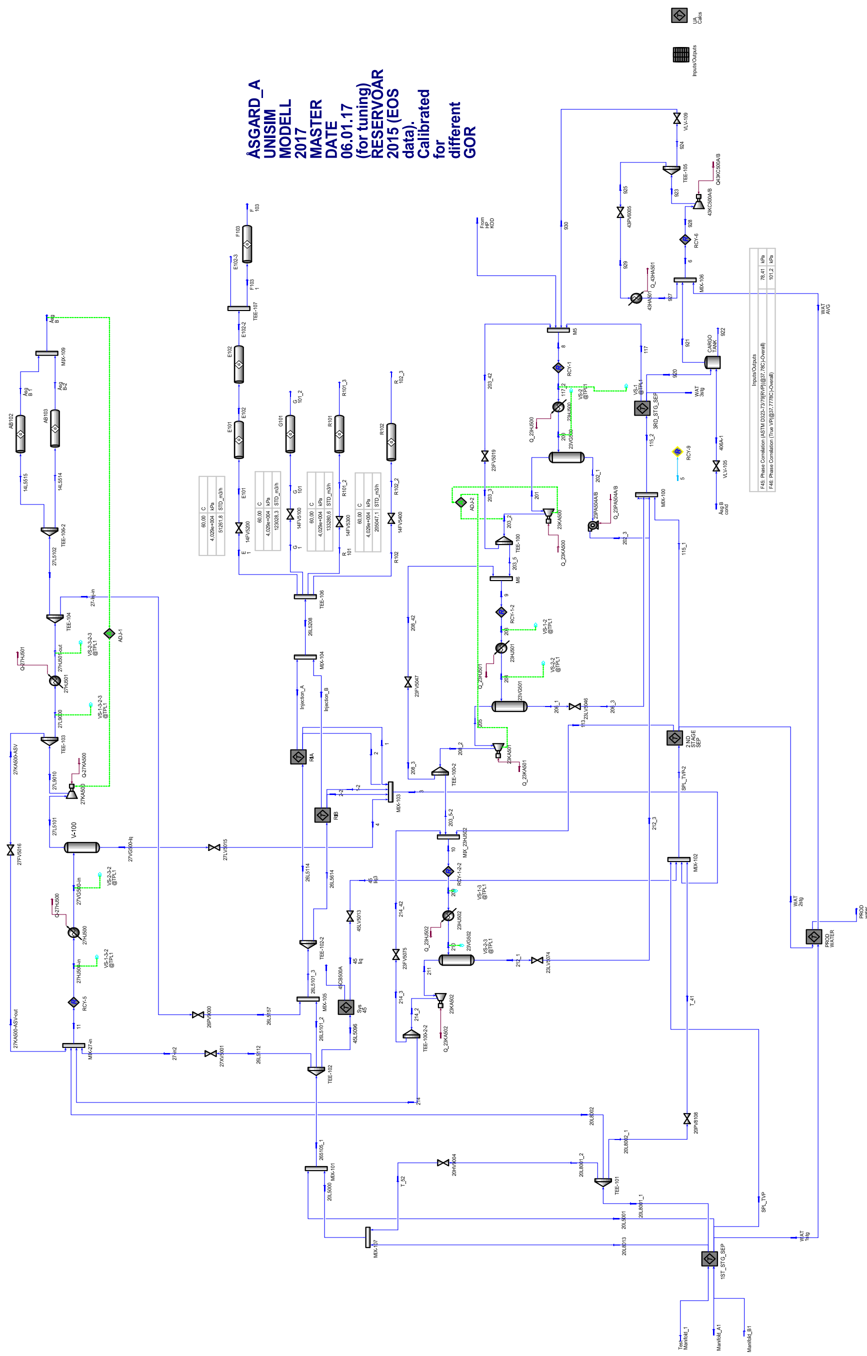
$$h_s^{\text{rev}} = \left(\frac{n}{n-1}\right)\frac{T_s Z_s R}{Mg}\left[\left(\frac{p_d}{p_s}\right)^{\frac{n-1}{n}} - 1\right] \tag{D.9}$$

This is still the ideal polytropic head, so to achieve the real head, equation D.9 must be divided by the polytropic efficiency of the compressor, η_{poly} as equation 4.2 shows, since $h \propto W$. The real polytropic head is shown in equation D.10.

$$h_{s,\text{poly}} = \left(\frac{n}{n-1}\right)\frac{T_s Z_s R}{Mg\eta_{\text{poly}}}\left[\left(\frac{p_d}{p_s}\right)^{\frac{n-1}{n}} - 1\right] \tag{D.10}$$

Equation D.10 gives head in units of meters. Since the variable head h may have many different units, some conversions may me made. Conversion formulas are given in appendix C.

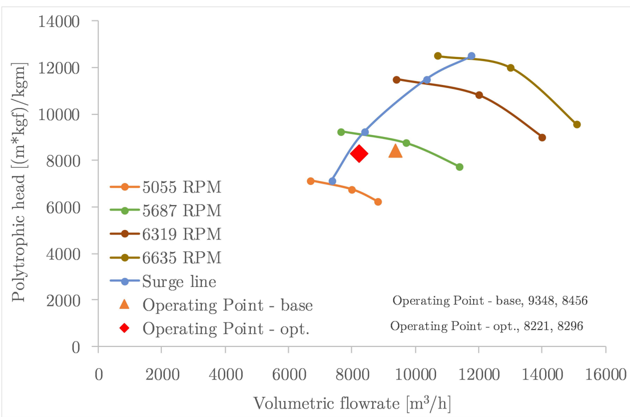
E UniSim model graphic



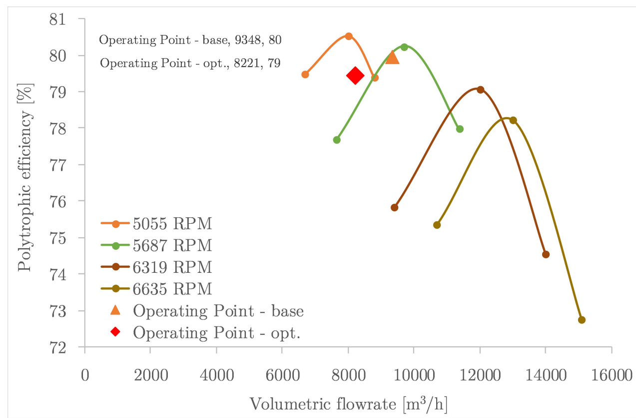
F Compressor curves with operating points

In this attachment the compressor curves are illustrated that are obtained. The compressor curves for each compressor consists of a polytropic head curve and a efficiency curve. These curves represent the compressor characteristics to give an overview. In the curves, both the operating point from the base model (▲) and the optimised model (◆) are given as points.

F.i Compressor 23KA500



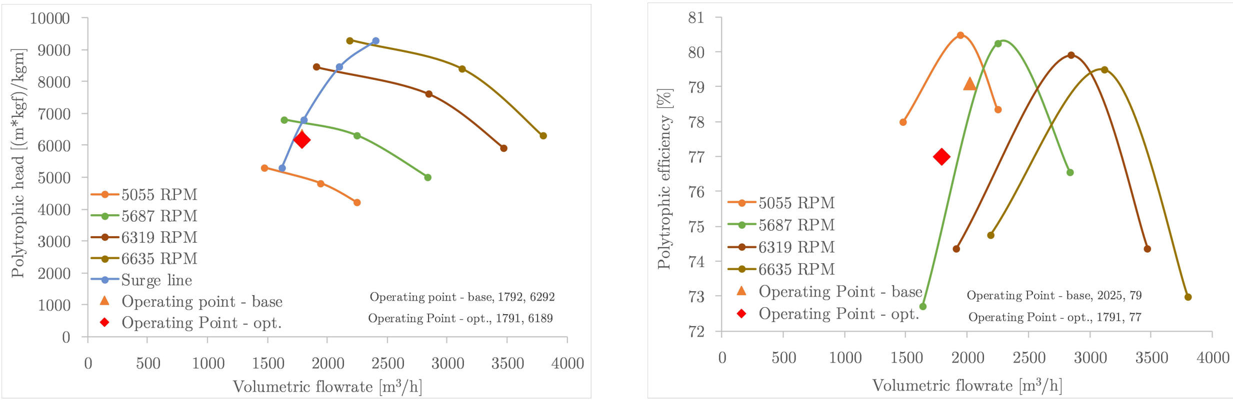
(a) Polytropic head curves .



(b) Efficiency curves.

Figure F.1: Compressor 23KA500; polytropic head and efficiency curves.

F.ii Compressor 23KA501

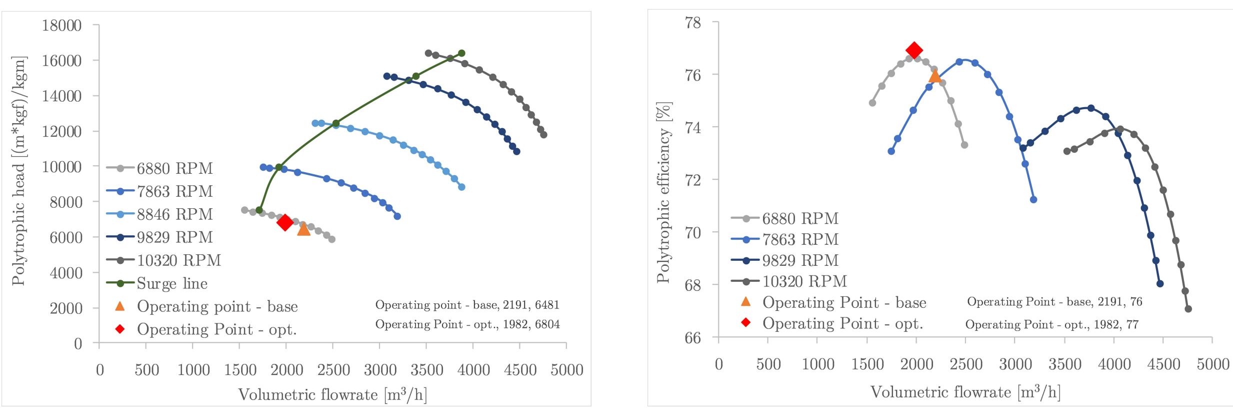


(a) Polytropic head curves .

(b) Efficiency curves.

Figure F.2: Compressor 23KA501; polytropic head and efficiency curves.

F.iii Compressor 23KA502

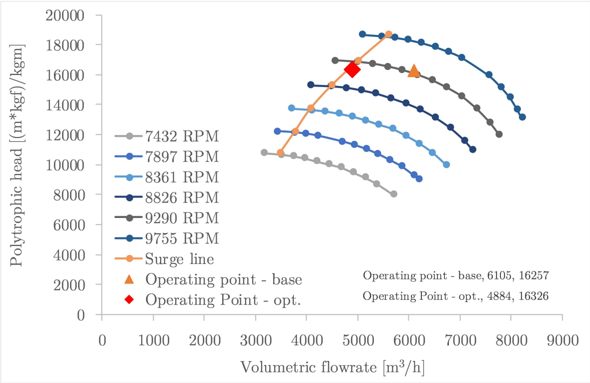


(a) Polytropic head curves .

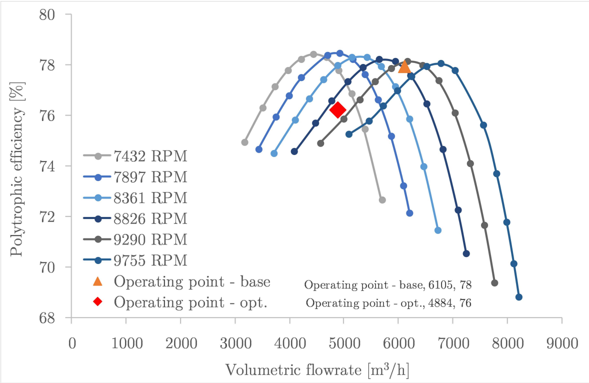
(b) Efficiency curves.

Figure F.3: Compressor 23KA502; polytropic head and efficiency curves.

F.iv Compressor 26KA601



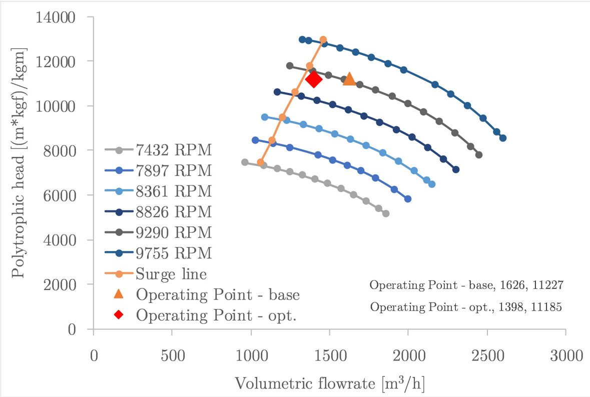
(a) Polytropic head curves .



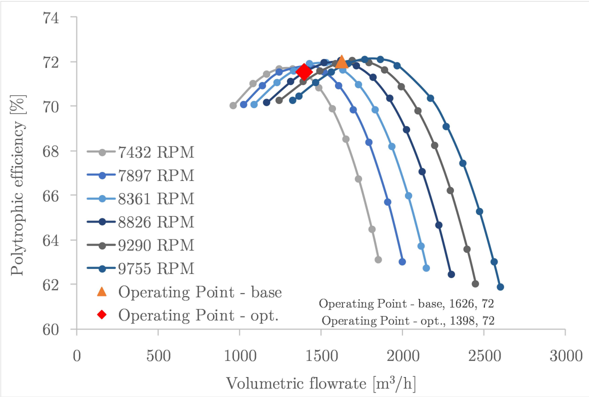
(b) Efficiency curves.

Figure F.4: Compressor 26KA601; polytropic head and efficiency curves.

F.v Compressor 26KA602



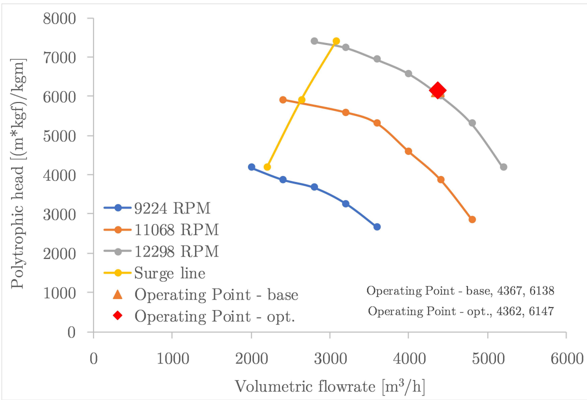
(a) Polytropic head curves .



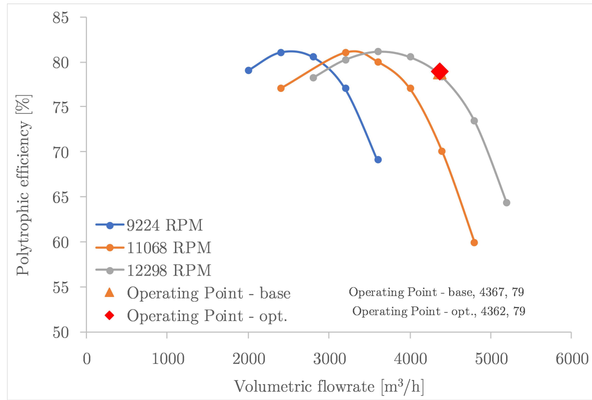
(b) Efficiency curves.

Figure F.5: Compressor 26KA602; polytropic head and efficiency curves.

F.vi Compressor 27KA500



(a) Polytropic head curves .



(b) Efficiency curves.

Figure F.6: Compressor 27KA500; polytropic head and efficiency curves.

G Project poster

The poster is featured on the next page.

A poster for the project is presented at the ”IKP dagen”, November 26 2018. The poster is in format A0 and was made in MS PowerPoint.

Compressor Train Optimisation

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Project description

In this project, the operating condition of an Equinor topside facility is studied, with a focus on the compressors (marked in red). Currently, the plant is inefficient due to a pressure depletion of the wells compared to the initial production in 2000. This causes a lower flow rate of gas through the compressors, forcing them closer to their surge points.

The goal of this project is to perform a techno-economical optimisation of the oil- and gas facility without exceeding given constraints of the process and product. This is done by reducing the total energy required. Also, the well pressure has depleted compared to initial production. Therefore, case studies about changing equipment are performed. To evaluate operational modes a UniSim model has been provided by Equinor. The battery limit for the project is taken as the topside facility only, excluding the subsea section and the production water streams, as illustrated in figure 1.

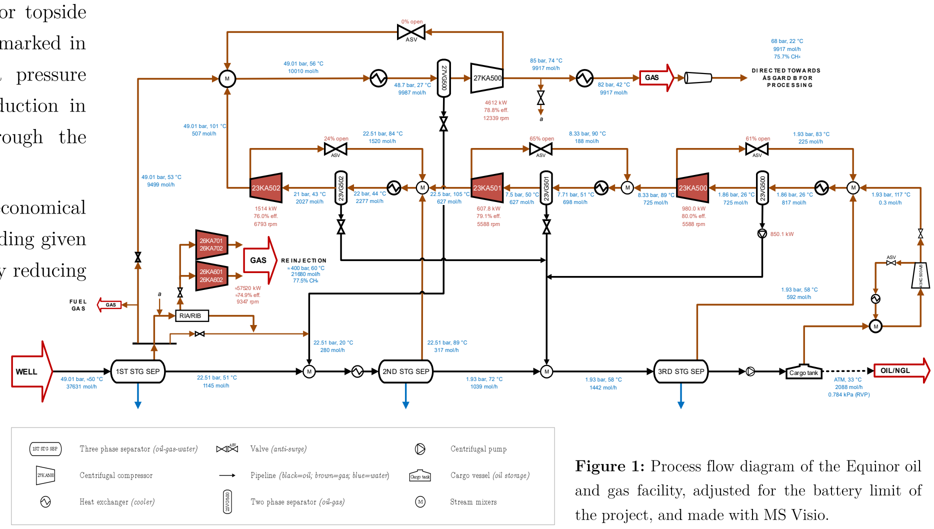

Figure 1: Process flow diagram of the Equinor oil and gas facility, adjusted for the battery limit of the project, and made with MS Visio.

Constraints

- RVP specification in product: 79.3 kPa.
- Well properties are constant.
- Export and injection pressures.
- Maximum duty delivered by turbines.
- Same compressor shafts (23KA500/501, 27KA601/602, 27KA701/702).

Handles

- Variable compressor motor speeds.
- Anti-surge valve openings.

The motor speeds and the anti-surge valve openings vary with the discharge pressures of the 1st and 2nd stage separator sections.

Model Fitting

- Heat exchangers are oversized and adjusted.
- Tightening the sensitivity of recycle blocks to remove mass balance errors.
- Removal of subsea and water production sections.

Case Studies - Optimised condition, New compressors, Well pressure depletion

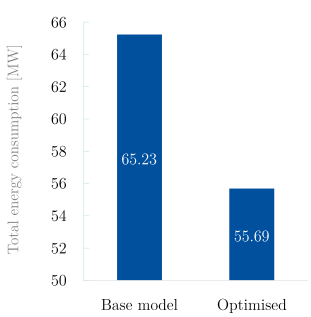

Figure 2: Total energy consumption in the received model compared to the optimised model, where the operation utilising the current equipment is optimised.

	Received	Optimised
Total energy consumption [MW]	65.23	55.69
26KA601/602 energy cons. [MW]	28.76	24.16
26KA701/702 energy cons. [MW]	28.76	24.16
23KA500/501/502 energy cons. [MW]	3.10	2.76
27KA500 energy cons. [MW]	4.61	4.62
23KA500 split factor (0-1)	0.31	0.20
23KA501 split factor (0-1)	0.30	0.25
23KA502 split factor (0-1)	0.75	0.71
26KA601/701 split factor (0-1)	0.2	0.00
26KA602/702 split factor (0-1)	0.2	0.07
2nd stage pressure [kPa]	2251	2161
3rd stage pressure [kPa]	193.3	193.3
23KA500/501 speed [rpm]	5588	5469
23KA502 speed [rpm]	6793	6787
26KA601/701/602/702 speed [rpm]	9347	9156

Table 1: Results from optimisation of operation utilising the current equipment. The total energy consumption is reduced by 9.54 MW.

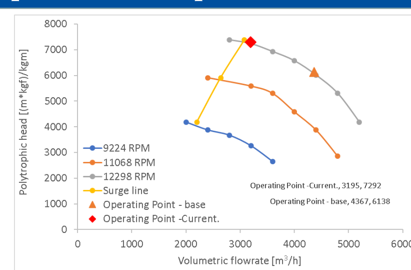

Figure 3: Compressor performance curve. Operating point for export compressor 27KA500 at its base point (▲) and 10% drop in well pressure (◆).

Oil price [NOK/barrel]	249.20 (low)	498.30 (normal)	747.45 (high)
Extra total profit [NOK/h]	1344	1869	2394
Extra total revenue [NOK/h]	399	923	1448
Reduction in OPEX [NOK/h]	945	945	945
Payback time [years]	6.4	4.6	3.6
NPV is positive in year...	13	8	6
NPV in year 10 [MNOK]	-10.53	13.23	36.98

Table 2: Results for the replacement of oversized compressors 23KA5000/501/502 (taking into account the assumptions made). Replacing the compressors in question is preferred because their anti-surge valves could not be fully closed in the optimisation (see table 1), giving these units a greater potential for improvement.

The total costs for replacement of the recompression train, is found to be approximately 75.2 MNOK. The values for the optimal case study (highest NPV value) are found by changing the discharge pressures of the 2nd and 3rd stage separator. A sensitivity analysis has been performed, taking into account three different oil prices.

Conclusion & Recommendation

The total energy consumption of the facility can be reduced by 9.54 MW by optimising the recompression- and reinjection compressor trains. Replacing the recompression train would provide additional power savings of 1.27 MW, with an NPV of 13.23 MNOK (assuming a constant production profile at a normal oil price of 498.3 NOK/barrel). This results in a beneficial CO_2 reduction as well as an improved oil production.

As a consequence of the ageing of the wells, the pressure and flow rate into the facility decreases. If the pressure of the well decreases below 90% of its current pressure, the export compressor (27KA500) is incapable to satisfy the pressure constraint. In this case, the exported gas can be reinjected instead of replacing the export compressor (when a linear flow- and pressure reduction, and equal composition is assumed).

equinor

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