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Optimale feil-posisjoner for reguleringsventiler i et kjølemedium-system

Birger Strand

Master i energi og miljø	
Innlevert:	juni 2014
Hovedveileder:	Truls Gundersen, EPT
Medveileder:	Rikke Mæhle, Aker Solutions
	Erik Jemsby, Aker Solutions

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MASTER THESIS

for

Student Birger Strand

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Optimal Valve Failure Positions for Cooling Medium Control Valves

Optimale feil-posisjoner for reguleringsventiler i et kjølemedium-system

Background and objective

Aker Solutions is involved in the Modifications Project for one of the fields in the North Sea with the objective to extend the lifetime of the existing installations of this field, as well as to prepare for the tie-in of a new installation. The project spans more than five platforms. The modification project consists of extensive upgrades and refurbishment over a wide range of platform systems.

On one of the platforms there is a large and complex cooling medium system with many consumers. Extensive modifications have already been carried out on this system during the 2013 shutdown. These modifications include the installation of new seawater/cooling medium coolers, replacement and installation of valves, and a large amount of pipe re-routing.

One item that requires a thorough study is the failure position of the control valves. In this study, focus will be on three specific valves (see item 3 in the list of tasks below). When the valve gets a failure signal (power outage, lack of air, problems in the control system, etc.), it will move to one of typically 3 different pre-determined valve positions (openings): (1) fully open, (2) fully closed, or (3) remain in its current position.

The main objective of this master thesis project is therefore to determine which failure position is the most optimal one in different operation and failure scenarios. This requires a study of the different scenarios and valve failure positions and also how the valve reaches its failure position.

The following tasks are to be considered:

1. Study the cooling medium system as a whole and getting familiar with the existing steady state simulation tool PIPENET.
2. The three alternative failure positions mentioned above should be simulated using PIPENET.

3. Study the control valves and the outlet streams of the coolers (1^{st} stage suction cooler, 2^{nd} stage after-cooler and dehydration suction cooler. The valves that this study concerns are scheduled for installation in the 2015 shutdown.
    - Identify failure scenarios
    - Understand the consequences of control valve failure in different positions
    - Suggest which failure position the valves should have

4. If time is available, understand how the valve actuator and solenoids work and propose the design.

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Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

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The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. Based on an agreement with the supervisor, the final report and other material and documents may be given to the supervisor in digital format.

☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department of Energy and Process Engineering, 14 January 2014

Olav Bolland
Department Head

Truls Gundersen
Academic Supervisor

Industrial contacts:

Rikke Mæhle, Senior Process Engineer, Aker Solutions, Stavanger
Phone: 51 85 12 43, E-mail: rikke.maehle@akersolutions.com

Erik Jemsby, Senior Process Engineer, Aker Solutions, Stavanger
Phone: 51 89 81 39, E-mail: erik.jemsby@akersolutions.com

Preface

This project thesis is written as a project at the Department of Energy and Process Engineering in the Energy and Environmental Engineering program at the Norwegian University of Science and Technology (NTNU). The thesis comprises 30 study points, which represents to the work load stipulated for the 10th semester.

I would like to thank Jeanette Hegdal at Aker Solutions (AKSO) for giving me the opportunity to write this master thesis. I would also like to thank Professor Truls Gundersen for accepting the responsibility to be my supervisor and for theoretical guidance during the project.

In addition to this I would like to thank my supervisors at AKSO, Rikke Mæhle and Erik Jemsby, for technical guidance and theoretical guidance. Finally, I would like to thank Arne Bratseth at AKSO for helping me with simulations in both PIPENET and HYSYS, and helping me to understand the simulation models.

Summary

As part of a modification project for one of the fields in the North Sea, the optimal failure position of three new globe valves has been studied. In a gas compression train, the gas needs to be cooled to a set temperature, and the globe valves control the amount of cooling medium required through the heat exchangers to reach this set temperature. A wrong choice of failure position could potentially cause expensive and important process equipment to get damaged.

The four failure positions that have been studied are fail open, fail close, fail close with travel stop and fail in position. Results were obtained using the simulation tools HYSYS and PIPENET, and calculations in Excel.

Fail close is not an optimal failure position, but it has been considered the most optimal as of today. It is very likely that this failure position will lead to a process shut down, which will give reduced production. The reason why it is still considered the most optimal solution is that no equipment will get damaged, and the actuator solution is well proven and trustable.

The results show that fail in position is the failure position which is best from a process point of view. There are however several questions connected to the actuator solution of fail in position. It is a relatively new failure position solution, which is why AKSO have not been able to get any feedback on how it works. It is very likely that fail in position will be used in similar installations in the future, when the actuator solution has been verified and improved if necessary.

It has been found that fail open is the least optimal failure solution. As the heat exchanger cooling medium demand will increase significantly when the valves are left fully open, there will be deficit of cooling medium to the other cooling medium consumers. A possible consequence of this is a process shutdown to avoid overheating of important process equipment. Another possible consequence of fail open as the failure position is hydrate formation, which in worst case can lead to damages to the compressors.

Sammendrag

Som en del av et modifikasjons prosjekt på et av feltene i Nordsjøen har den optimale feil posisjonen til tre nye globe ventiler blitt studert. Gassen i ett gass kompresjons tog må bli kjølt ned til en gitt temperatur, og globe ventilene kontrollerer hvilken mengde kjølemedium som trengs gjennom varmevekslerne for å nå denne temperaturen. Et galt valg av feil posisjon kan potensielt føre til skader på dyr og viktig prosessutstyr.

De fire feilposisjonene som har blitt studert er at ventilen åpner helt, stenger helt, stenger med en mekanisk sperre eller at ventilen blir låst i posisjonen den har når den feiler. Resultatene ble funnet ved hjelp av simuleringsverktøyene HYSYS og PIPENET, samt utregninger i Excel.

At ventilen stenger helt er ikke en optimal feilposisjon, men har blitt vurdert som den mest optimale per i dag. Denne feil posisjonen vil med stor sannsynlighet føre til en nedstengning av prosessen, som igjen vil gi redusert gass produksjon. Grunnen til at det likevel er ansett som den mest optimale feilposisjonen er at det ikke vil bli skader på utstyr, samt at aktuatorløsningen er velprøvd og til å stole på.

Resultatene og diskusjonen viser at den beste feilposisjonen fra prosess synspunkt er at ventilen forblir i posisjonen den har hvis den feiler. Det er dog flere spørsmål knyttet til denne feilposisjonens aktuatorløsning. Det er en relativt ny feil posisjon løsning, som er grunnen til at AKSO ikke har fått tak i tilbakemeldinger på hvordan den fungerer. Når aktuatorløsningen knyttet til denne feilposisjonen blir verifisert og forbedret om nødvendig, er det veldig sannsynlig at den vil bli brukt i liknende installasjoner i fremtiden.

At ventilen åpner helt er vurdert som den dårligste feilposisjonen. Varmevekslernes behov av kjølemedium vil øke kraftig ved denne feilposisjonen, noe som vil føre til underskudd på kjølemedium til de andre forbrukerne av kjølemedium. En mulig konsekvens av dette er nedstengning av prosessen for å unngå overheting av viktig prosessutstyr. En annen mulig konsekvens av denne feilposisjonen er hydrat formasjon, noe som i verste fall kan føre til skader på gas kompressorene.

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Tables

Nomenclature

A	[m^2]	Heat transfer area
A, B, C	[-]	Pump performance coefficients
$c_{p(CM)}$	[J/(kgK)]	Specific heat capacity of the cooling medium
$c_{p(gas)}$	[J/(kgK)]	Specific heat capacity of the gas
C_v	[-]	Valve flow coefficient
D	[m]	Diameter of a pipe
f	[-]	Fanning friction factor
F	[-]	Heat exchanger correction factor
g	[m/s^2]	Gravity constant
K	[-]	K-factor
K_v	[-]	Valve flow coefficient
L	[m]	Pipe length
LMTD	[K]	Logarithmic mean temperature difference
$\dot{m}_{CM}$	[kg/s]	Mass flow of cooling medium
$\dot{m}_{gas}$	[kg/s]	Mass flow of gas
ΔP_{CV}	[N/m^2]	Pressure drop due to a control valve
ΔP_{elev}	[N/m^2]	Pressure drop due to change in elevation
$\Delta P_{fitting}$	[N/m^2]	Pressure drop due to fittings within a pipe
ΔP_{fric}	[N/m^2]	Pressure drop due to friction and fittings
$P_{gas(out)}$	[N/m^2]	Heat exchanger outlet pressure
ΔP_{pump}	[N/m^2]	Pressure increase produced by a pump
ΔP_{loss}	[N/m^2]	Total pressure drop in a pipe
$P_{PIC(out)}$	[N/m^2]	Pressure indicator controller outlet pressure
ΔP_{pipe}	[N/m^2]	Pressure drop along the pipe
ΔP_{plate}	[N/m^2]	Pressure drop due to an orifice plate within a pipe
ΔP_{pump}	[N/m^2]	Pressure increase in a pump
$P_{valve(out)}$	[N/m^2]	Control valve outlet pressure
q_{CM}	[W]	Heat transfer of cooling medium
q_{gas}	[W]	Heat transfer of gas
q_{HX}	[W]	Heat exchanger heat transfer
Q	[m^3/s]	Volumetric flow rate

Q_{CM}	[m^3/s]	Volumetric flow rate of cooling medium
s	[-]	Valve setting
$T_{boil(CM)}$	[K]	Boiling temperature of the cooling medium
$T_{CM(in)}$	[K]	Heat exchanger cooling medium inlet temperature
$T_{CM(out)}$	[K]	Heat exchanger cooling medium outlet temperature
$T_{gas(in)}$	[K]	Heat exchanger gas inlet temperature
$T_{gas(out)}$	[K]	Heat exchanger gas outlet temperature
$T_{HH(gas)}$	[K]	Platform high high gas temperature
ΔT	[K]	Temperature difference
ΔT_{CM}	[K]	Temperature difference cooling medium
ΔT_{gas}	[K]	Temperature difference gas
u	[m/s]	Fluid velocity
U	[W/(m^2K)]	Overall heat transfer coefficient
z	[m]	Change in elevation of a pipe
ρ	[kg/m^3]	Fluid density
ρ_{CM}	[kg/m^3]	Cooling medium density
ρ_0	[kg/m^3]	Density of water at 20^{o}C

1. Introduction

Aker Solutions (AKSO) is involved in a modification project at an oilfield in the North Sea. Due to client confidentiality the operator and platform names covered in this thesis cannot be disclosed. Instead, Imaginary names will be given to the platforms of the field. The field consists of four platforms plus a platform that is controlled from the field. The names that will be given are the following: Midgard A, Midgard B, Loke and Dagrun. In addition to this, the platform that is controlled from the field will be called Munin. The function of the platforms will be explained in the platform process description (Chapter 2).

A part of the modification project is the installation of three new control valves connected to the cooling medium system at Loke. In normal operation, these valves control the flow of cooling medium through the 1st stage recompression suction cooler, dehydration suction cooler and the 2nd stage recompression cooler respectively, to give a set gas temperature at the outlet of the heat exchangers. Chapter 2 will present functions of the platforms in general, and a more detailed presentation of the cooling medium system at Loke.

This project will focus on finding the optimal failure position for the three control valves. The failure position means which position the valve should have if the actuator for some reason fails. The failure position could be, fail open, fail close, fail close with travel stop or fail in position. Fail open means that the valve is set to a 100% opening when a failure scenario occurs, fail close means that the valve will get completely closed, while fail in position means that the valve will stay locked in the position it has if the actuator fails. Fail close with travel stop means that the valve closes, but is left with a small predetermined opening. A mechanical component ensures that the valve cannot go to openings below this set travel stop.

From a process point of view, the optimal failure position would be the one that gives the smallest impact on the rest of the system. This is however not the only factor to take into account when choosing the failure position. The actuator solution connected to the failure position must be solid. The control valves should originally have been installed during a shutdown in summer 2013, but AKSO and the operator could not decide the failure position in time. A decision was however made during autumn 2013, and the new control valves will be installed during a shutdown in 2015. There was a lot of disagreement and discussion on which failure position that would be the most optimal, which is why AKSO wants a new study of the valves and their failure position.

The new control valves are globe valves. Globe valves are better suited for control compared to butterfly valves, which is the type of control valves in operation today (FEED Report, 2010). In Chapter 3 control valve and actuator theory will be presented along with theory about the new control valves that will be installed at Loke. In addition possible failure scenarios and possible consequences with the different failure positions will be presented in Chapter 3.

In order to obtain results, the simulation tools PIPENET and HYSYS were used in combination with calculations in Excel. PIPENET is a modelling tool which enables accurate simulation of the flow of fluid through a network of pipes and other components, and will be used in this project. A model of the cooling medium system has already been built in PIPENET, and this enables the simulation of different failure positions of the three valves. The two simulation tools will be more thoroughly reviewed in Chapter 3. Chapter 4 presents how PIPENET and HYSYS in combination with calculations in Excel were used to obtain the results. These results are presented in Chapter 5, while the results are discussed in Chapter 6. The conclusion, based on the results and discussion along with theory, are presented in Chapter 7. Chapter 7 also suggests what further work that is needed.

1.1 Objective

The purpose of doing this project is to evaluate which is the most optimal failure position of the three new control valves at Loke. Because of the discussion related to which failure position to choose, the installation of these valves got delayed. This is also why AKSO wanted a new and independent analysis of the failure positions for the valves.

This project mainly consists of three parts:

- Get familiar with the cooling medium system at LOKE and learn how to use PIPENET and HYSYS.
- Evaluate control valve and actuator theory, both in general and for the new control valve systems at Loke
- Simulate case studies using PIPENET, HYSYS and Excel calculations in combination, and analyse the results

1.2 Scope

- Study three control valves to obtain the most optimal failure position of the valves
- The mass flow of gas at normal operation was used. For fail in position the mass flow of gas was changed to evaluate how much it can change before unwanted effects occur
- No heat loss to the surroundings has been assumed

2. Platform process description

The modification project comprises all work scope on the existing platforms Midgard A, Midgard B, Dagrun, Loke and Munin, plus the building of a new platform which in this project will be given the name Odin. Today the production is taking place at Midgard A, Midgard B and Munin. The produced oil and gas at Midgard B is processed at the same platform, while the production at Midgard A and Munin is processed at Dagrun. Processed gas is sent further to another field and from there in pipelines to Germany and England. From the summer 2015, a direct changeover from Dagrun to the new systems on Odin is planned. The separator and the test separator, which today are located at Dagrun and Midgard A respectively, will be disconnected and replaced with by a new test- and production separator located at Odin after the shut down during the summer 2015.

The platform with the three control valves which were studied in this project, Loke, was installed in 1999. It is a water injection platform, and has a large gas compressor train which enables gas lift and gas injection. In addition, it has a considerable power plant, which is supplying the field. The required gas for the gas injection and gas turbine is supplied from Dagrun today. It is processed at Loke before it is consumed. After the shut down during the summer 2015 the gas needed at Loke will be provided from Odin.

2.1 Loke cooling medium system

There is a large cooling medium system at Loke with many consumers, such as the gas lift and gas injection compressor package, fuel gas compressor package and the main turbine generator package. The cooling medium consists of 60% water and 40% glycol. It is provided by three cooling medium pumps, which sends the cooling medium further to three cooling medium coolers, where the cooling medium is cooled with filtered seawater. One of the coolers will have seawater supply from Odin, and is only to be used in cases where the fine filters experience blocking. A simplified sketch of the cooling medium system can be seen in Figure 1.

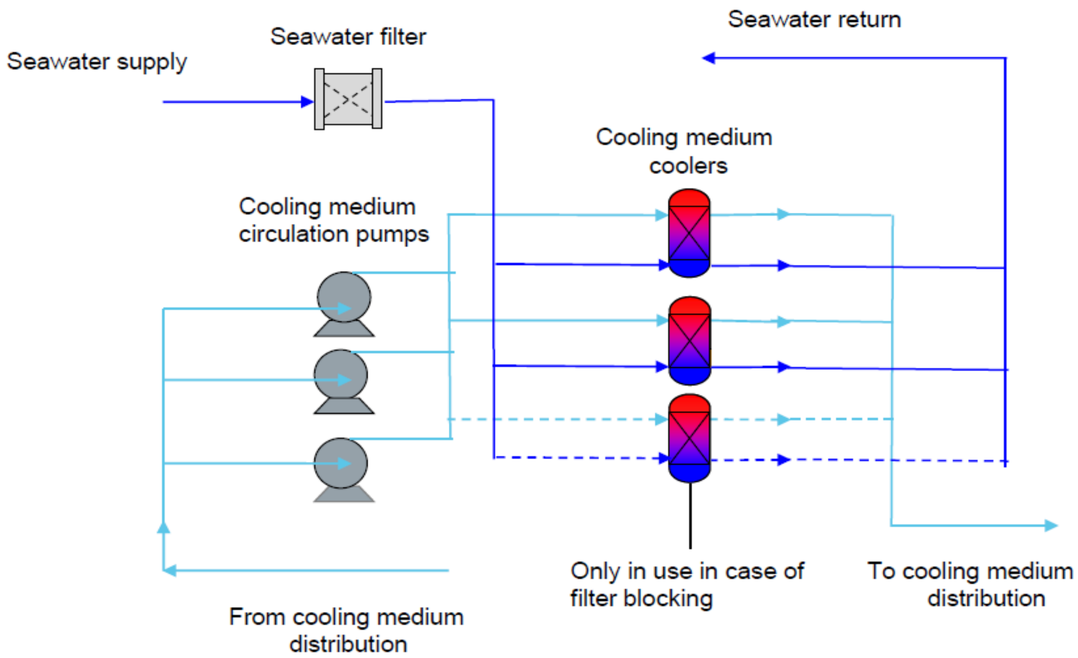

Figure 1 Simplified sketch of the cooling medium system

New cooling medium coolers were installed in 2013, and have increased capacities compared to the old coolers. This increased capacity is achieved by a larger heat transfer area between seawater and cooling medium in the new coolers. As well as the increased capacity, the new coolers ensure less dependent on the fine filters, which experience algae blocking in certain periods of the year. It should also be mentioned that the dehydration suction cooler has a maximum cooling medium flow of 210 000kg/s to avoid vibrations (FEED report, 2010).

Before the shutdown in 2013, there were only a few cooling medium consumers which had control valves to ensure that the correct flow rate was directed to the consumers. The result of this was a surplus of cooling medium to some consumers, causing a deficit of cooling medium in other parts of the system. During the shutdown in 2013 manual valves were installed to direct the cooling medium flow to the required coolers. These valves were manually adjusted during commissioning until the flows and temperatures were as required and optimised. No further adjustments will be required unless there is a change in the cooling medium system.

When the cooling medium has gone through the cooling medium coolers, it is distributed to the consumers. Among these consumers are the three heat exchangers which will be studied in this project; the 1st stage recompression suction cooler, the dehydration suction cooler and the 2nd stage recompression after-cooler.

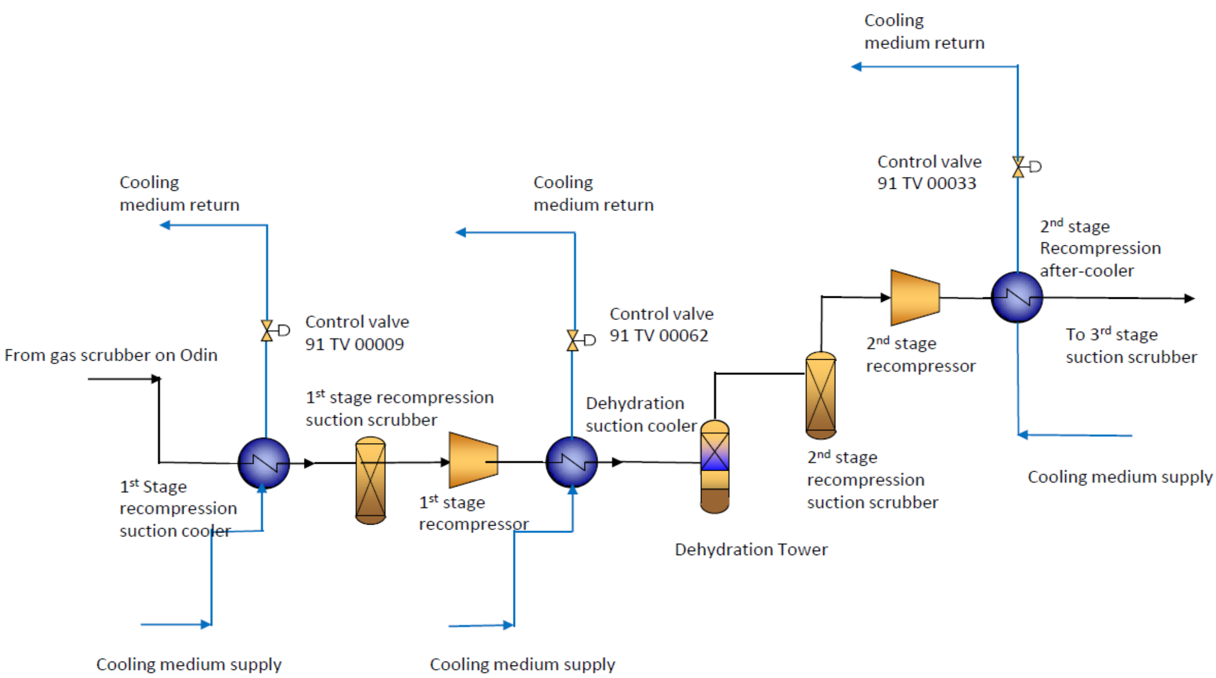

Figure 2 simplified sketch of the Loke compressor train

Figure 2 shows a simplified sketch of the Loke compressor train from the 1st stage recompression suction cooler to the 2nd stage recompression. The control valves that control the cooling medium flow through the heat exchangers can be seen above the heat exchangers. These are the valves that will be analysed to obtain the most optimal failure position. The 1st stage recompression suction cooler will normally not be in operation after the summer 2015 shutdown, as the gas will be cooled at Odin. However, it will operate if the heat exchanger at Odin for some reason is out of operation. It will also have a minimum flow even though the Odin cooler is running. This will keep the control valve system going, and if the gas temperature differs compared to the set temperature, the control valve can adjust this by sending more or less cooling medium through the 1st stage recompression suction cooler. The 1st stage suction cooler will also need to be operational in case gas is recycled from the 1st stage compressor through the anti-surge valve.

3. Theory

3.1 Control valves

Process plants consist of many control loops that are all networked together to produce a product that will be offered for sale. Each control loop is designed to keep some important process variable such as pressure, temperature or flow rate within a required operating range to ensure quality of the end product. The loops receive, and internally create disturbances that affect the process variable. A controller reduces the effect of these disturbances. Sensors and transmitters collect information about the process variable, and their relationship to a desired set point. A controller processes this information and decides what must be done to get the process variable to where it should be after a load disturbance occurs. Some type of final control element must implement the strategy selected by the controller when all the measuring, comparing and calculation are done. In the process industry, the most common final control element is the control valve (Emerson Process Management, 2005).

The valve systems at Loke that will be studied in this project are automatic control systems. These systems consist of (Nesbitt, 2007):

- The fluid to be controlled
- A sensor for the process variable
- A controller that affects the actuator
- An actuator which modulates the valve
- A control valve to control the flow

When we talk about a control valve, this control system is often referred to as a control valve assembly. Even though the control valve is an important part of a control valve assembly, it is not accurate to say that the control valve is the most important part. The control loop can be looked at as an instrumentation chain. Like any other chain, the whole chain is only as good as its weakest link (Emerson Process Management, 2005).

There are several different types of control valves, such as ball valves, butterfly valves, globe valves, pinch valves and eccentric rotating plug valves. The existing valves at Loke are butterfly valves, while the new valves, where the optimal failure position will be studied, are globe valves. Therefore, these valves will be further analysed in this project.

3.2 Actuators

The Function of an actuator is to adjust the position of the valve to ensure correct control of the process fluid. The position of the valve may only be open or closed as in the case for isolating valves, or in any intermediate position for control valves. In order to operate effectively, the actuator must be sufficiently powerful to produce a positive, accurate and quick response to a control signal. In the case of the three control valves that will be studied in this project, the actuator is required to return the valve to a predetermined position or to hold its current position in the event of signal failure. In order to meet the demands of the process, reliability and economy it is important to specify the correct type and size for the actuator (Nesbitt, 2007).

There are many different types of actuators. Pneumatically operated control valve actuators are the most popular type in use. Other types that are widely used are electric, hydraulic and manual actuators. The type of actuators that will be installed to the three new control valves at Loke are pneumatically operated diaphragm actuators. Diaphragm actuators can be direct-acting or reverse-acting. Direct-acting means that increasing air pressure pushes down a diaphragm and extends the actuator stem, while reverse-acting means that increasing air pressure pushes up a diaphragm and retracts the actuator stem (Emerson Process Management, 2005).

3.3 Butterfly valves

The control valves controlling the cooling medium flow to the 1st stage recompression suction cooler, the dehydration suction cooler and the 2nd stage recompression after-cooler are as mentioned butterfly valves as of today. Butterfly valves are rotary valves where a disc-shaped closure member is rotated through 90^{o}, to open or close the flow passage (Smith & Zappe, 2004).

Butterfly valves are widely used in the process industry. Economy is the main reason for this. The installation costs are lower than other valves since pipework reducers are generally not required as butterfly valves tend to be line size. In addition, butterfly valves use less metal in their manufacture which makes them cheaper to purchase. Butterfly valves offer an advantage in terms of flow capacity per investment. Figure 3 shows a butterfly valve, with examples of the disc fully open and closed.

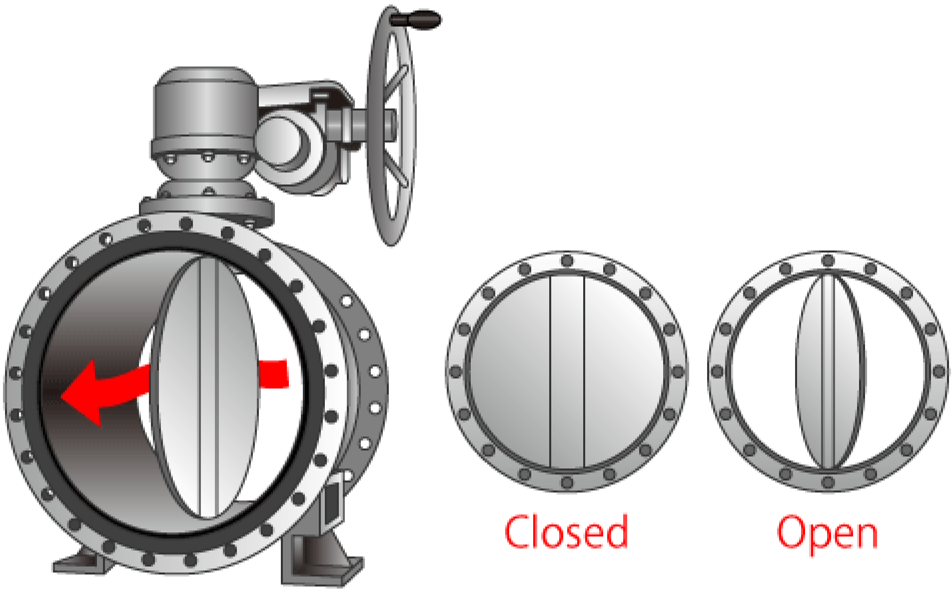

Figure 3 Butterfly valve (TLV.com)

Butterfly valves are typically installed in secondary loops where control is not deemed as critical, or in loops where the pressure drop across a system must be limited. Using a butterfly valve in these applications could work fine, but often it will cause problems. The conventional butterfly valves have useable controllable range at valve openings between 25 and 50 percent. At low openings, they generally have a very high gain, which is the change in flow per unit % change in the valve opening. This means that at low flow rates, a small change in the valve opening will result in a much larger change in the flow rate. In the other end, above 50% valve opening, one gets little change in flow rate with a large change in valve opening. Conventional butterfly valves have a characteristic called quick opening, which means they have a high gain at low valve travel. However, there is a new type of butterfly valve on the market, which has a characteristic called equal percentage, which means that whenever the valve opening is changed, the percentage change in flow is equal to percentage change in the valve opening. The new type of butterfly valves has a useable control range between 15 and 70 percent, which is close to the control range of a globe valve (valve user, 2009).

In addition to quick opening and equal percentage flow characteristics, linear flow characteristic is a commonly observed flow characteristic type. The linear flow characteristic has a constant slope, which means that valves of this type have a constant gain through the complete range of flows. The different flow characteristics can be seen in Figure 4.

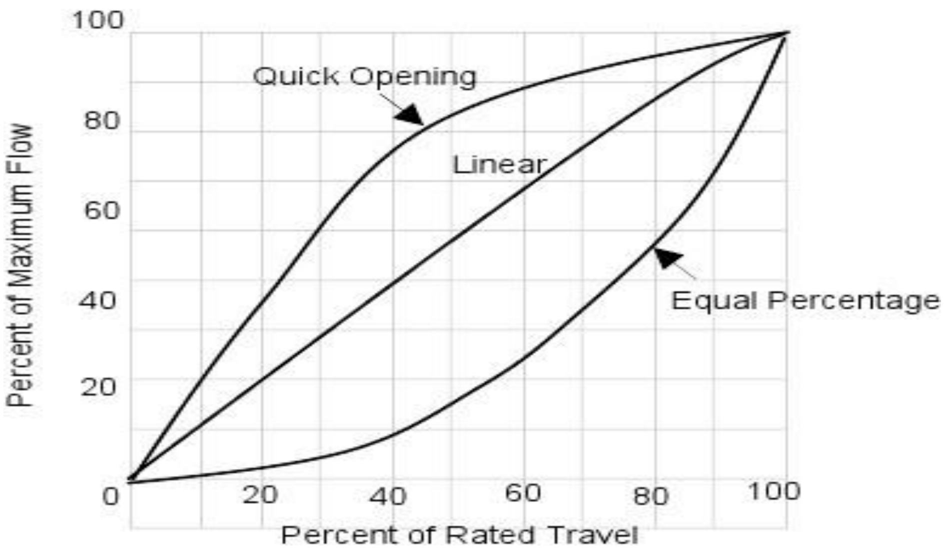

Figure 4 The three most commonly used valve flow characteristics (globalspec.com)

3.4 Globe valves

Globe valves are valves where the closure member is moved squarely on and off the disc. The closure member is normally referred to as a disc, irrespective of its shape. By this method of disk travel, the seat opening varies in direct proportion to the travel of the disk. Globe valves are ideally suited for duties involving control of flow rate because of this proportional relationship between valve opening and disc travel (Smith et al., 2004).

Even though the new butterfly valves have a highly improved useable control range compared to the conventional type, globe valves still have a higher useable control range, and is therefore better suited as control valves (valve user, 2009). This was also the reason why AKSO decided to change the three control valves from butterfly valves to globe valves. Figure 5 shows an illustration of a globe valve, with names of the main components. As the figure shows, the valve stem keeps the valve in a closed position. By turning the handle, the valve will open. Flow rate control is determined by the lift of the valve stem.

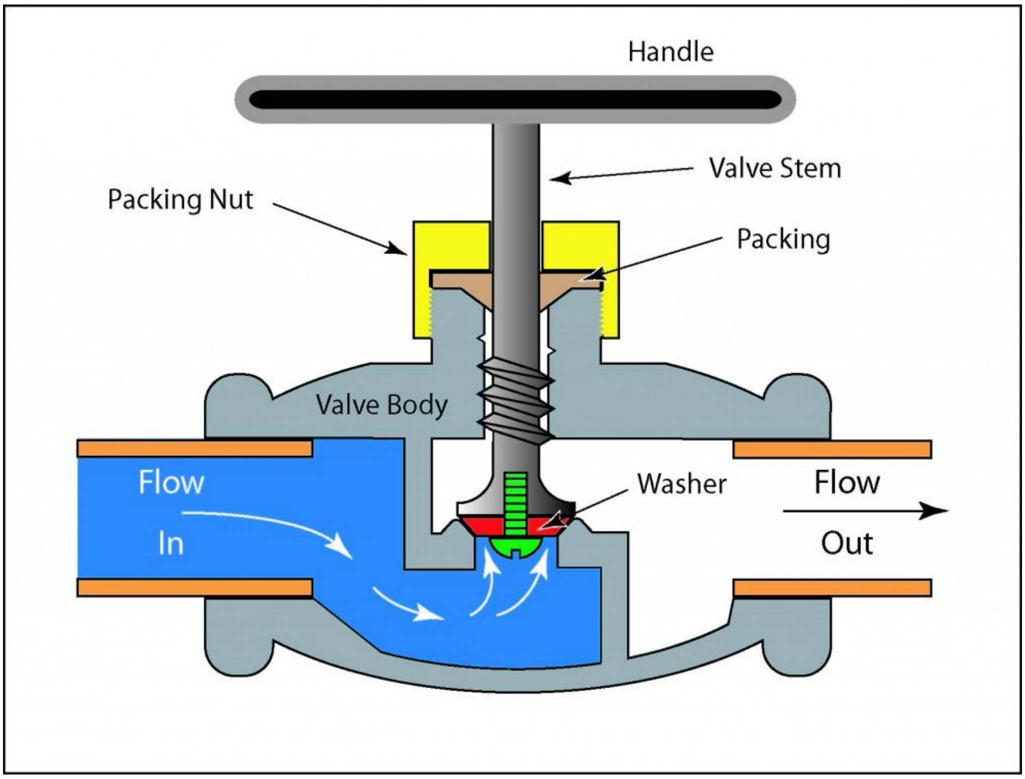

Figure 5 Illustration of a globe valve (ctgclean.com)

In addition to its precise control, other advantages related to globe valves are their use in high pressure systems, and the fact that they can be used for both gas and liquid systems. Another feature of globe valves is that there is less risk of damage to the valve seat or valve plug by the fluid than other types of manual valves, even when it is used in partially open position.

When it comes to disadvantages for globe valves, there are several. Because of its S-shaped flow path, the pressure drop is higher than of many other types of valves. The valve stem must be turned several times in order to open and close the valve, which means that valve operation time is longer. This may eventually cause leakage of the gland seal (packing) (tlv.com).

3.5 New control valves and actuators

The new control valves at Loke are pneumatically operated single seat globe valves. The valve characteristics curve is equal percentage, which can be seen in Figure 6.

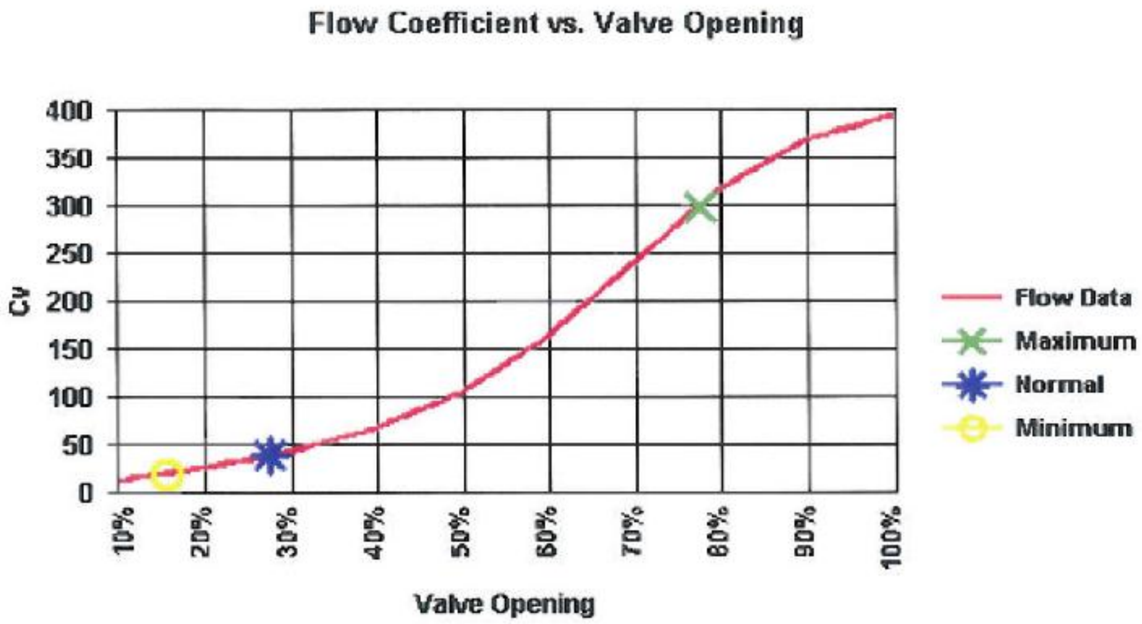

Figure 6 Flow characteristics for one of the new control valves at Loke

As can be seen from Figure 6, the valve has a minimum and maximum valve opening. In general it can be said that a Fisher globe valve has good regulation at valve openings between 15% and 85%. It can also give a good control at valve openings less than 15%, but other unwanted effects can occur at low openings. In liquid applications, these effects are often related to erosion and cavitation. If a valve operates at openings less than 15% over a long period, there can be possibilities for such damages (conversation with valve vendor).

While it was decided early in the project that the new control valves should be globe valves, the actuator solution was not decided before autumn 2013. The new valves should have been installed during a shutdown in the summer 2013, but due to the discussion on which actuator solution to choose it was as mentioned earlier delayed.

It may very well be that one failure position is the most optimal from a process perspective, but that another solution is the most optimal when other factors are taken into account. The new globe valves are going to be installed at the same place as the butterfly valves are located today. Early in the project limited space for the installation of the new control valves was an issue, but a pipe was removed, which made all the potential actuator solutions possible to install. An important factor when the most optimal failure solution is decided is that it must have a reliable actuator solution.

The figures below show the actuator solutions that were discussed by AKSO and the owner of the platform. They all use air to control the valve opening. There is an air supply and a filter regulator that regulates the air inlet pressure to a set value. The filter regulator also filters small particles. The controller gets an electrical signal which tells if more or less valve opening is needed, and controls the air pressure based on this signal (conversation with valve expert AKSO).

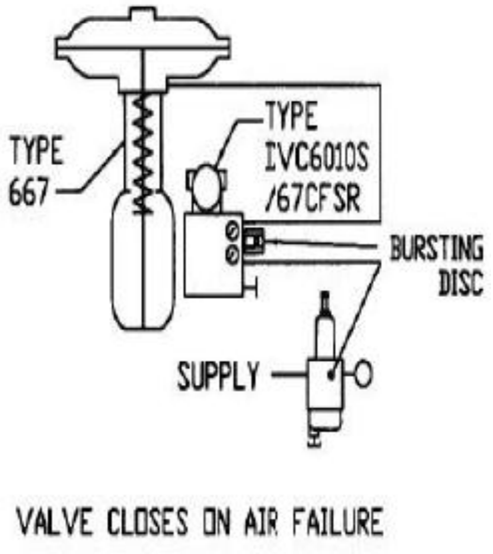

Figure 7 Actuator solution with fail close as the failure position (emersonprocess.com)

Figure 7 shows the Fisher 667 diaphragm actuator. It is a reverse-acting, spring opposed diaphragm actuator. This means that increasing air pressure results in a larger valve opening (emersonprocess.com).

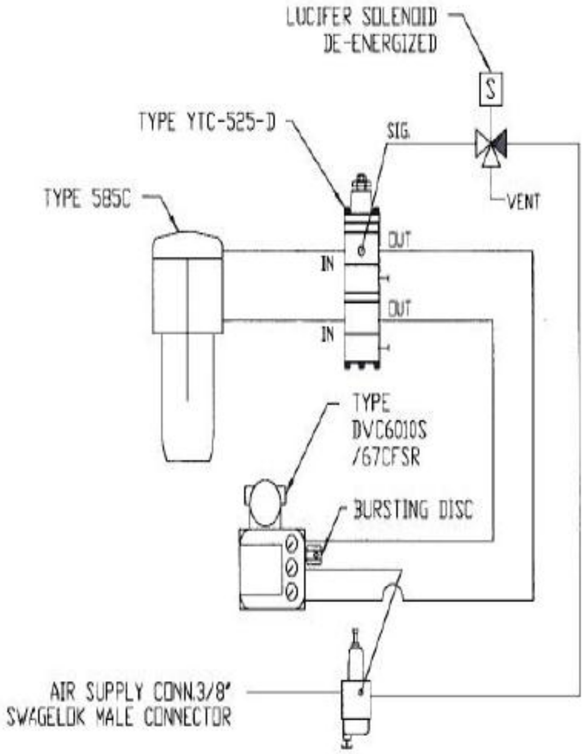

Figure 8 Double acting actuator solution with fail in position as failure position (emersonprocess.com)

Figure 8 shows the Fisher 585C linear double acting piston actuator. Double acting means that pressurized air can be added on both sides of the piston, dependent on whether a higher or lower valve opening percentage is needed. This is a fail in position actuator solution. This means that both sides of the piston must have air at the same pressure to ensure that it stays in its failure position.

Two new components are introduced in Figure 8 compared to Figure 7, a solenoid valve and a snap acting relay (YTC-525-D). A solenoid is a three way valve that in normal operation has pressurized air flowing through. When the actuator fails, the solenoid valve closes and pressurized air bleeds to vent. The position of the solenoid valve in normal operation and when the actuator fails can be seen in Figures 10 and 11. In normal operation enough air to keep the YTC-525-D snap acting relay open is sent from the filter regulator and then through the solenoid valve. If a failure scenario occurs, the solenoid valve is closed like Figure 8 shows. Then the air from the controller is directed through the snap acting relay and from there towards the solenoid valve, where it bleeds out to vent (emersonprocess.com).

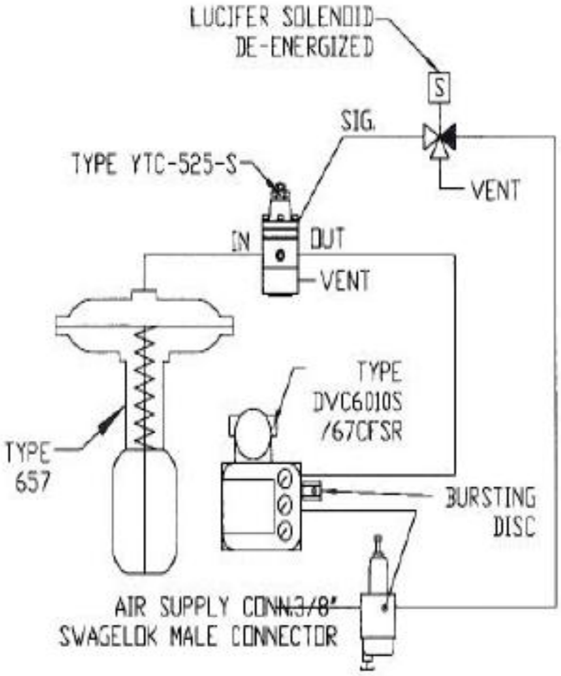

Figure 9 Actuator solution which were discussed with fail in Position as failure position (emersonprocess.com)

Figure 9 shows the Fisher 657 spring actuator which is a direct-acting, spring opposed diaphragm actuator. This solution is similar to the 667 solution, but as it is a fail in position solution it has two extra components; the YTC-525-S snap acting relay and the solenoid valve. If the valve fails, the snap acting relay changes the direction of the supply air towards the solenoid valve, which bleeds air to vent. This ensures that the pressure on the valve spring is kept constant, which keeps the valve in the position where it failed.

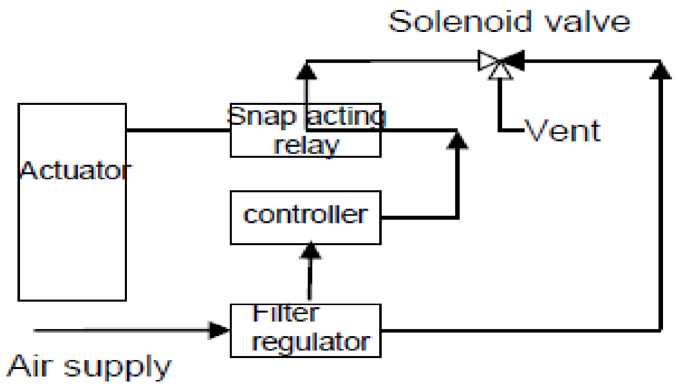

Figure 10 Air flow when the actuator fails

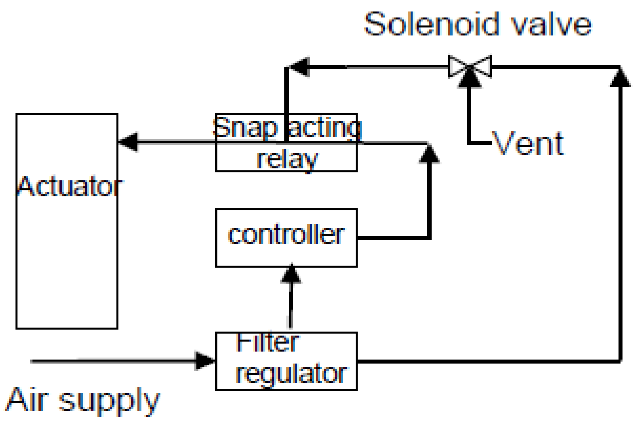

Figure 11 Air flow normal operation

3.6 Hydrates

Gas hydrates are a form of clathrate, which is any compound wherein guest molecules are entrapped in a cage structure composed of host molecules. With the gas hydrates the lattice is formed by water molecules (Campell, 1992). Gas hydrates look like ice, but in contrast to ice, hydrates can form at temperatures up to 25°C. Formation of hydrates in pipelines can cause the pipelines to get partially blocked, or in worst case totally blocked (Larsen, 2002). The worst scenario in this project is that a hydrate plug gets entrained in the gas flow. A hydrate plug can cause deformation of the compressor blades, which can cause serious damage or in the worst case destroy the compressor.

Roughly speaking, one can say that hydrates can form where there is free water, the right type of gas molecules, and relatively high pressure as well as relatively low temperature (Larsen, 2002). Figure 12 is an example of a phase envelope with its hydrate curve, which is the light blue curve. Hydrates can form at conditions which are to the left of the hydrate curve. As it can be seen, hydrates can form at temperatures up to 20°C in this case.

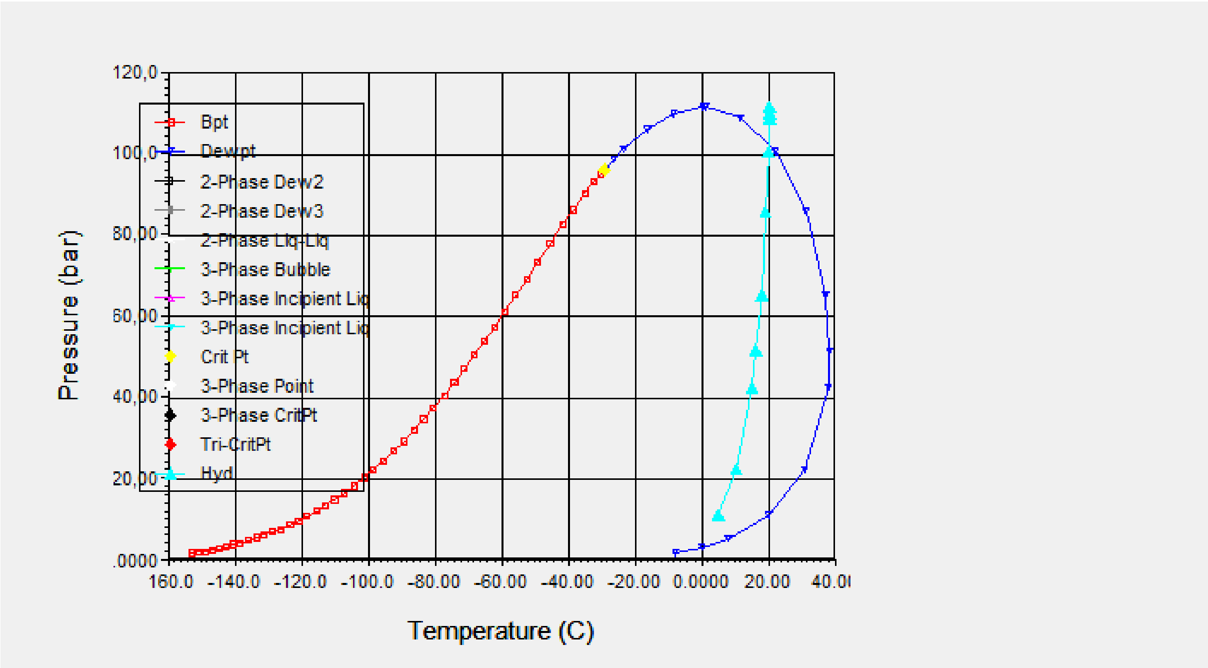

Figure 12 Phase envelope and hydrate curve from HYSYS

3.7 Failure scenarios and consequences

There are two main possible failure scenarios connected to the new control valves at LOKE. One possibility is if the control signal of the actuators for some reason fails, and another is if the air supply pressure to the actuator fails. This can for instance be if the tubing connected to the actuator solution gets damaged. If one of these failure scenarios occurs, the valve that it affects will go to its failure position. The possible consequences of a valve failure are as one would expect different for the failure positions.

A possible consequence if the failure position is fail open is as mentioned hydrate formation. This will obviously not be an issue for the case of fail close as the failure position. There are several ways to prevent gas hydrates from plugging pipes and flow lines, such as thermodynamic and kinetic inhibitors. This will however not be used at the Loke gas line. Therefore, it is important to ensure that hydrates cannot be formed by keeping the temperature above the region where hydrate formation can occur. In addition to possible hydrate formation, the case of fail open as the failure position might affect the other cooling medium consumers. There is a pressure indicator controller (PIC) downstream the cooling medium coolers. PIPENET normally fails if the pressure is below 6.5bara out of this PIC. The result of this is that no calculations were done using PIPENET. This pressure is the pressure that ensures that all cooling medium consumers get their required cooling. That some process equipment has a cooling medium deficit can lead to a shutdown as the temperature of the gas flow through that equipment will increase.

If the failure position is fail close, the gas temperature downstream the heat exchanger may get sufficiently high to trip a temperature high high alarm, which is present downstream all of the three heat exchangers to avoid overheating of the gas compressors. As well as this, fail close as the failure position might lead to boiling of the cooling medium. In addition to fail close, where the valve closes completely, the failure position of fail close with travel stop will be simulated. Fail close with travel stop means as mentioned that the valve will be left with a small opening, and is simulated to evaluate whether some of the consequences related to fail close can be avoid by choosing fail close with travel stop as the failure position.

If the failure position is fail in position there are two main issues. It will lock the valve in the position it has when it fails, which means that there will be a constant cooling medium flow. This means that if the gas flow increases, the gas temperature downstream the heat exchanger will increase as there is more gas to cool for the same amount of cooling medium. In the opposite case the gas temperature downstream the heat exchanger will decrease. In addition to this, the actuator solution has to be analysed for the case of fail in position. The reason for this is the fact that this actuator solution is relatively new compared to the other failure position actuator solution, which has been used in installations for decades (conversation with valve expert AKSO).

3.8 PIPENET

The PIPENET Vision programs have been designed to enable the accurate simulation of the flow of fluid through a network of pipes and other components. It is used in the oil and gas, power, petrochemical and shipbuilding industries. The program consists of three different modules; standard module, spray/sprinkler module and transient module (sunrise-sys.com).

The standard module was used to build the Loke cooling medium system model in PIPENET. It is a tool for solving general flow problems with liquids, gases or steam in pipe and duct networks, cooling water systems, steam distribution systems and HVAC systems. The spray/sprinkler module is for analysis of fixed fire-protection systems employing water, while the transient module is for analyses of transient flow in liquids, within all types of networks (Sunrise systems, 2006).

PIPENET Vision Standard module uses equations to model the flow behaviour in pipes, pumps, valves, etc. In the past, fluid flow analyses were performed by engineers using manual calculations. To do this analysis for a large network takes considerable effort. The PIPENET Vision Standard Module can help the user by providing faster and more reliable solutions (Sunrise systems, 2006).

3.8.1 Pressure drop in a pipe

In this project, the PIPENET Vision Standard module is used to find pressure drop through pipes, valves and pumps, and to find the flow rate that passes through the three control valves. The pressure drop in a pipe is modelled in PIPENET Vision Standard module using the following equation:

$$\Delta P_{loss} = \Delta P_{fric} + \Delta P_{elev} + \Delta P_{plat} \qquad (1)$$

Where: ΔP_{loss} = total pressure drop

ΔP_{fric} = pressure drop due to friction and fittings

ΔP_{elev} = pressure drop due to change in elevation

ΔP_{plat} = pressure drop due to any fitted orifice plate

To calculate the pressure drop, it is necessary to determine the values of ΔP_{fric}, ΔP_{elev}, and ΔP_{plat}. ΔP_{fric} is the sum of the pressure dropes along the pipe (ΔP_{pipe}), and any loss in pressure due to the presence of pipe fittings ($\Delta P_{fitting}$) (Sunrise systems, 2006). The pressure drop along the pipe can be found using the following equation, based on the work of Darcy (Darcy, 1857):

$$\Delta P_{pipe} = \frac{2fL\rho u^2}{D} \tag{2}$$

Where:

f = the fanning friction factor
L = pipe length
ρ = fluid density
u = fluid velocity
D = diameter of the pipe

The pressure drop due to fittings within the pipe, $\Delta P_{fitting}$, is found as follows:

$$\Delta P_{fitting} = \frac{K\rho u^2}{2} \tag{3}$$

Where: K = K-factor, a dimensionless number depending on the fitting type

This gives the following pressure drop due to frictional effects:

$$\Delta P_{fric} = \Delta P_{fitting} + \Delta P_{pipe} \tag{4}$$

The pressure drop due to difference in elevation between two ends of a pipe, P_{elev}, is expressed as follows:

$$\Delta P_{elev} = \rho \times g \times z \tag{5}$$

Where:

z = change in elevation of the pipe
g = acceleration due to gravity

An orifice plate within a pipe can cause the pressure to change by some incremental value. In PIPENET the user can choose the value that decides the pressure drop due to an orifice plate present (Sunrise systems, 2006).

3.8.2 Pressure changes due to pumps

A simple pump model uses the performance curve. The user inputs flow rate against head data at 100% rpm. The performance curve at other speeds can then be calculated by PIPENET using the homogenous relationships for pumps. The modelling equation used for pump is as follows (Sunrise systems, 2006):

$$\Delta P_{pump} = AQ^2 + BQ + C \tag{6}$$

Where:

Where:	ΔP_{pump}	=	pressure increase produced
	Q	=	volumetric flow rate through the pump
	A, B, C	=	pump performance coefficients

3.8.3 Pressure drop due to control valves

The valve can be characterised by one of two built-in models which require either a K-factor and a port area, or a flow coefficient. For the PIPENET model of the Loke cooling medium system has used the model that requires a flow coefficient. The modelling equation is as follows:

$$\Delta P_{CV} = \frac{\rho Q^2}{\rho_0 C_v^2} \tag{7}$$

Where:	ΔP_{CV}	=	pressure drop across the valve
	Q	=	volumetric flow rate through the valve
	ρ	=	fluid density
	ρ_0	=	density of water at 20^oC
	C_v	=	valve flow coefficient

It should be mentioned that the flow coefficient used in PIPENET has been converted from C_v to K_v. While C_v uses English dimensions, K_v uses metric dimensions. The relation between C_v and K_v is as follows:

$$K_v = 0.865C_v \tag{8}$$

3.9 HYSYS

Aspen HYSYS is a process modelling system used by oil and gas producers, refineries and engineering companies to optimize process design and operations (Aspentech, 2013).

It is possible to realistically model process equipment like pumps, valves, separators, compressors etc. by using HYSYS. Parameters like mass flow, sizing of equipment, pressure, temperature etc. can be found in HYSYS.

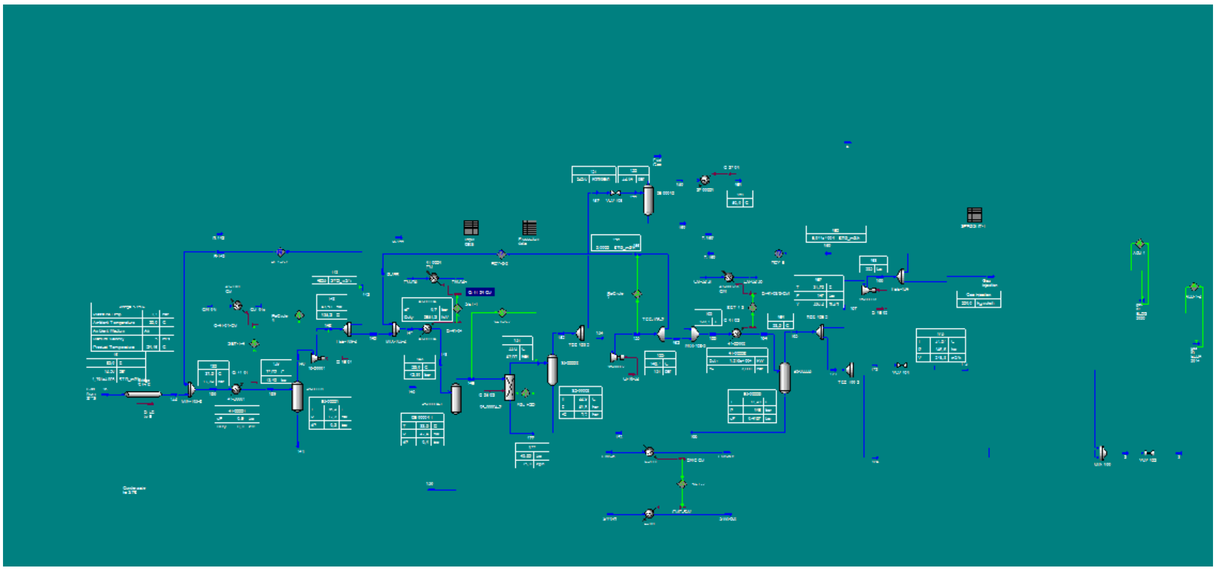

Figure 13 Loke compressor train model in HYSYS

The aim is to find the gas temperature at different failure positions of the globe valves. This will be found from calculations in Excel. HYSYS will be used to find the inlet gas temperatures and gas properties, as well as to generate hydrates curves. The model that will be used was built by AKSO, and can be seen in Figure 13.

4. Methodology

4.1 Definition of the problem

There are two main questions that will be answered in this project. The primary question is to evaluate which is the optimal failure position for the three new globe valves. The secondary objective is to study the actuator solutions that were suggested by AKSO, and evaluate the design. The two questions are connected to each other as the optimal failure position cannot be given without knowing that it has a solid actuator solution.

Simulations in PIPENET and data collection from HYSYS in combination with analysis and calculations will be done to obtain the answer to the first question. The second question will be answered by studying the actuator solutions that were proposed by AKSO. Each of the solutions will be analysed before a design is proposed.

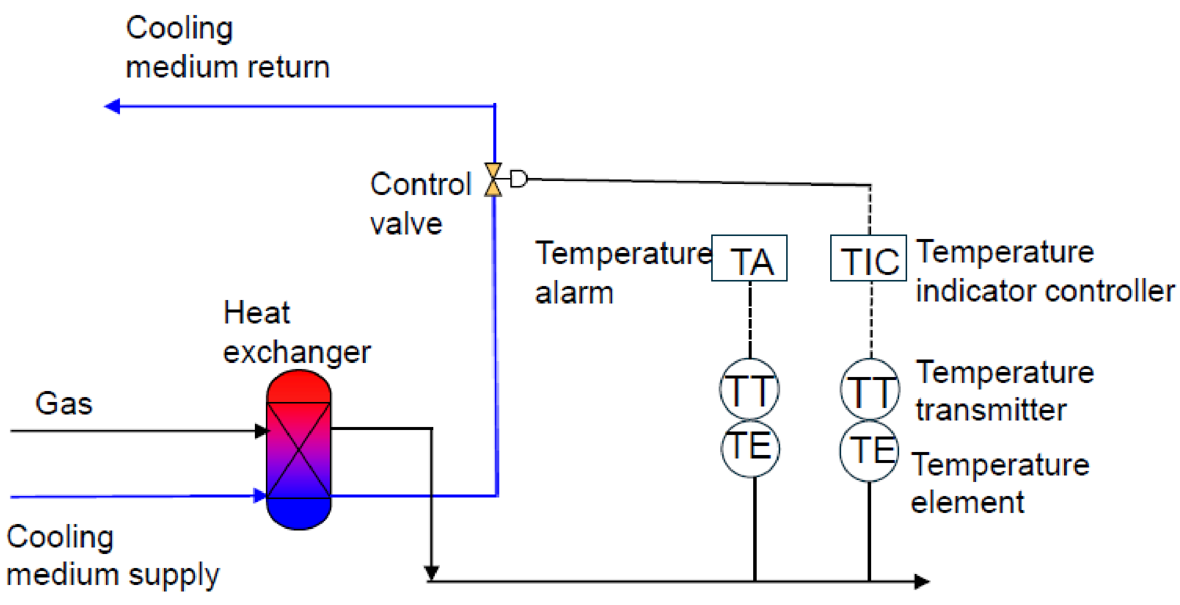

Figure 14 sketch showing how the gas temperature downstream the heat exchangers are controlled

Figure 14 shows what the system with the heat exchanger and the globe valve that controls the gas temperature downstream the heat exchanger looks like. The gas temperature is normally measured by the temperature element (TE) to the right, while the temperature transmitter (TT) sends an electrical signal to the temperature indicator controller (TIC), which regulates the control valve based on this signal. If the actuator solution fails, there is a backup system with a temperature alarm. If the gas temperature exceed the high high temperature this temperature alarm will trip, which will cause a process shut down.

4.2 Methodology for fail open

The methodology to obtain how the system is affected by fully opening the valves in the case of a failure action was divided in three parts. The first part was done in PIPENET, where the cooling medium flow was found when the valves were set to fully open. While one valve at a time was simulated fully open, the two other valves were kept in their normal position. In this way, the cooling medium demand for each heat exchanger in the case of fail open as the failure position was found, as well as the control valve outlet pressure.

The valve downstream the 1st stage recompression suction cooler was simulated first. To ensure that the simulation did not fail, the pressure controller downstream the cooling medium coolers were turned off. Normally there is a pressure indicator controller (PIC), and, as mentioned, PIPENET does normally fails if the supply pressure to the cooling medium drops below 6.5bara. This pressure is the pressure that will ensure that all cooling medium consumers get their required amount of cooling medium. Even though the pressure controller was turned off, the pressure was checked after the simulation to evaluate if a fail open failure position will lead to a cooling medium deficit among the other cooling medium consumers.

When the failure position of fail open of the three control valves were simulated in PIPENET, the volumetric flow rate of cooling medium through them could be found. The density of the cooling medium was found in the heat exchanger vendor sheets. When both the density and volumetric flow rate are known, the mass flow of cooling medium can be found as follows:

$$\dot{m}_{CM} = \rho_{CM} \times Q_{CM} \tag{9}$$

Where: $\dot{m}_{CM}$ = mass flow of the cooling medium

ρ_{CM} = density of the cooling medium and

Q_{CM} = volumetric flow rate of the cooling medium.

The value of ρ_{CM} is found from the vendor data sheets.

The mass flow of cooling medium has to be found to be able to find the duties and the temperatures of the gas and cooling medium leaving the heat exchangers. The temperatures of the cooling medium and the gas entering the heat exchangers are known. To find the two unknown temperatures and the duty, three equations were used. The heat transfer of cooling medium and gas can be found from the following two equations:

$$q_{CM} = \dot{m}_{CM} \times c_{p(CM)} \times \Delta T_{CM} \tag{10}$$

$$q_{gas} = \dot{m}_{gas} \times c_{p(gas)} \times \Delta T_{gas} \tag{11}$$

Where:

Where:	q_{CM}	=	heat transfer of cooling medium
	q_{gas}	=	heat transfer of gas
	$\dot{m}_{CM}$	=	mass flow of cooling medium through the heat exchanger
	$\dot{m}_{gas}$	=	mass flow of gas through the heat exchanger
	$c_{p(CM)}$	=	specific heat capacity of the cooling medium
	$c_{p(gas)}$	=	specific heat capacity of the gas
	ΔT_{CM}	=	temperature difference cooling medium
	ΔT_{gas}	=	temperature difference gas

No heat loss to the surroundings is assumed. The amount of heat removed from the gas is then always equal to the heat added to the cooling medium. Therefore the expressions for q_{CM} and q_{gas} were set equal to each other. The equations were manipulated in order to find an expression for the cooling medium temperature leaving the heat exchangers. This was combined with the heat transfer equation:

$$q_{HX} = U \times A \times F \times LMDT \tag{12}$$

Where:	q_{HX}	=	heat exchanger heat transfer
	U	=	heat transfer coefficient
	A	=	heat transfer area

F = heat exchanger correction factor

LMTD = logarithmic mean temperature difference

LMTD is defined as follows for a counter-current heat exchanger:

$$LMTD = \frac{\Delta T1 - \Delta T2}{ln\frac{\Delta T1}{\Delta T2}} \quad (13)$$

Where:

$\Delta T1$ = $T_{gas(in)} - T_{CM(out)}$

$\Delta T2$ = $T_{gas(out)} - T_{CM(in)}$

$T_{gas(in)}$ = heat exchanger gas inlet temperature

$T_{gas(out)}$ = heat exchanger gas outlet temperature

$T_{CM(in)}$ = heat exchanger cooling medium inlet temperature

$T_{CM(out)}$ = heat exchanger cooling medium outlet temperature

Both the dehydration suction cooler and the 2nd stage after-cooler have data sheets with a thorough review on how to calculate the duties for the respective heat exchangers. Two different areas are given; area provided and area required clean. The area required clean was used in the calculations. There were also two different values given for the heat transfer coefficient; required and clean heat transfer coefficient. The clean heat transfer coefficient was used in the calculations.

The F-factor describes the deviation from a pure counter current heat exchanger. The F-factor values were found from the vendor data sheets. The calculations of the 1st stage recompression suction cooler were done based on the data from the dehydration suction cooler since they are the same type of heat exchangers. This means that the relationship between the clean and effective area and the required and clean heat transfer coefficient is the same for the 1st stage recompression suction cooler and the dehydration suction cooler. The F-factor used in the calculations of the 1st stage recompression suction cooler and the dehydration suction cooler were therefore equal.

The gas c_p-values were found from HYSYS. Each heat exchanger calculation has one cold side and one hot side c_p-value. The average of those two was used in the Equations 9 and 10. The cooling medium c_p-values were found from vendor data sheets.

Equations 9, 10 and 11 were combined in Excel to find the heat exchanger outlet gas temperature and cooling medium temperature. The expression found for the cooling medium outlet temperature included the gas outlet temperature. By changing the gas outlet temperature, the cooling medium temperature would also change. The gas temperature was adjusted until the expressions in Equation 9, 10 and 11 gave the same answer. When this is found, the outlet gas and cooling medium temperature is found as well as the heat exchanger duty.

For the case of fail open, it is the gas temperature and the pressure downstream the heat exchangers that is of interest. While the heat exchanger gas outlet temperature has to be calculated, the heat exchanger outlet pressure has a set value and is found from the HYSYS gas compressor train model. The heat exchangers have a set pressure drop, which means that the heat exchangers outlet pressure will not change if the heat exchanger duty increases or decreases. When the pressure and temperature of the gas is known, the hydrate formation temperature can be found from HYSYS. Based on this curve it is evaluated if there is risk of hydrate formation. This is most relevant for the 1st stage recompression suction cooler and the dehydration suction cooler. The reason is the fact that the gas has been through two scrubbers and the dehydration tower when it arrives at the 2nd stage after-cooler, and the water content in the gas is therefore too low for hydrate formation to occur.

It was also considered whether fail open is even a possible solution for the dehydration suction cooler, which has a maximum cooling medium flow of 210 000kg/s. If the required cooling medium flow in a fail open failure scenario is higher than that, the dehydration suction cooler cannot have fail open as its failure position.

4.3 Methodology for fail close

There are as mentioned two opportunities if fail close is found as the optimal failure position; fail close, and fail close with travel stop. When it comes to fail close there is nothing to simulate in PIPENET as the valves are completely closed. The gas temperatures downstream the three heat exchangers are found in HYSYS, as the inlet and outlet temperatures will be the same in the case of closed control valves. If the temperature downstream the heat exchangers increase above a set maximum temperature, a process shut down (PSD) will lead to shutdown of the compressor train.

Another issue with fail close is the cooling medium temperature, as the temperature will approach the gas inlet temperature. Then there can be a problem that boiling of the cooling medium occurs, which is why the boiling point of the cooling medium has to be found. This will be found from Figure 15 which shows how the boiling point of water/MEG varies with pressure.

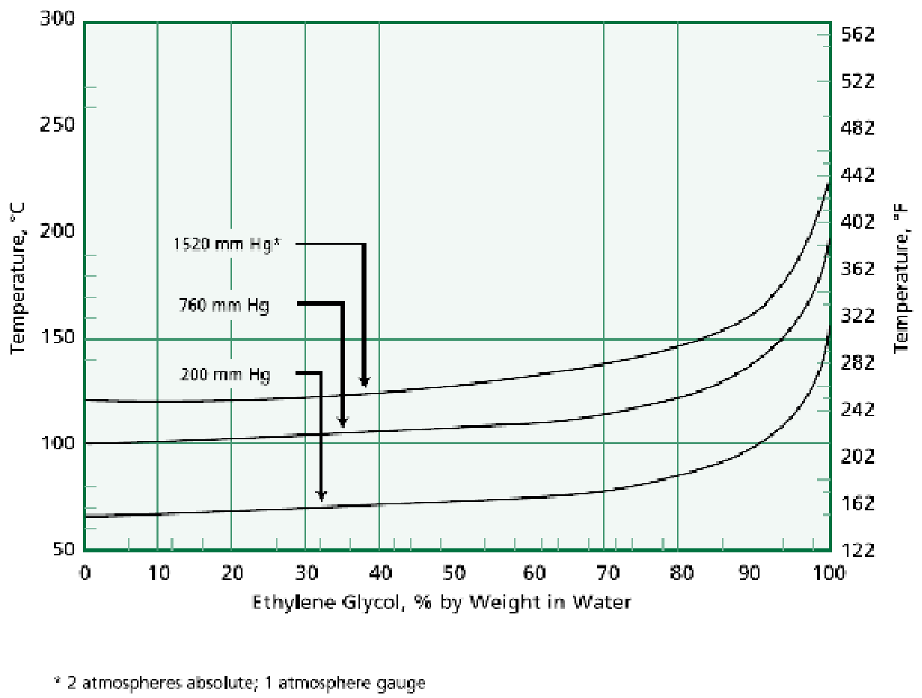

Figure 15 Boiling point curves for MEG/water solutions (meglobal.biz/)

The upper curve, which shows how the boiling point varies with the composition at 2bara, will be used to find the boiling point of the 60%/40% water/MEG solution. That curve will be used as the valve outlet pressure is very close to 2bara for all of the three valves.

4.4 Methodology for fail close with travel stop

The objective of fail close with travel stop is the same as for fail close, to find the gas- and cooling outlet temperatures of the heat exchangers. The methodology does however differ as there is a small amount of cooling medium flowing through the heat exchangers in the travel stop alternative. Therefore PIPENET was used to find the cooling medium flow at openings between 10 and 20%. The reason why lower openings were not simulated are the flow characteristics curves from the vendor. Flow characteristics between 0 and 10% were not delivered because the minimum opening of the valves varies from 10% at the valve downstream the dehydration suction cooler to 17% for the valve downstream the 2nd stage after-cooler. A fail close with travel stop solution can never have a valve opening percentage that is larger than the minimum opening that the valve operates at. However openings between 10 and 20% were simulated for each of the valves to evaluate the trends. The duty and temperatures at these openings were found by combining the same equationsthat were used for the case of fail open.

The gas temperature downstream the coolers will decrease with fail close with travel stop compared to fail close, as there is a small amount of cooling medium going through the heat exchanger. Therefore it might be that the gas temperature is above the set maximum gas temperature downstream the heat exchangers for the case of fail close, while it is below for the case of fail close with travel stop. Figure 15 will be used in the same way as for fail close to evaluate if boiling of the cooling medium will occur at openings between 10 and 20%.

4.5 Methodology for fail in position

The evaluation of fail in position as the failure position has been done by analysis and calculations in combination. The calculations were done based on the fact that the valves fail at the normal operation point. Under normal operation both the heat exchanger gas inlet and outlet temperatures and the gas flow rate are known. The water outlet temperature was found by combining Equations 10 and 11.

As long as the gas flow rate is steady, the heat exchanger outlet temperature will not change as the cooling medium flow rate is steady. In normal operation the gas flow is quite stable. If the gas flow rate should change it is however of interest to evaluate how much the gas flow rate can increase without tripping the high high temperature alarm, and how much the flow rate can decrease without risking hydrate formation. A changed mass flow of gas can for instance happen if a well is shut down or is going to be tested. The duty was assumed constant, and the temperatures were set to the temperature which trips the high high alarm downstream the heat exchangers. Then the mass flow of gas could be found by using Equation 11, and compared to the mass flow at normal operation. The hydrate curves for the gas flow line downstream the heat exchangers were found in HYSYS to see how much the gas flow could decrease without risking hydrate formation. The gas flow at this temperature was calculated by using Equation 11. These calculations were not done for the 1st stage recompression suction cooler, as the gas is precooled at Odin. If the gas flow changes at Odin, the heat exchanger at Odin will be regulated and the temperature at the 1st stage recompression suction cooler will remain constant.

The actuator solution has to be thoroughly analysed for the case of fail in position. It is of course important to have a solid actuator solution regardless of which failure position is chosen. However, the solution of fail in position is relatively new, while the other alternatives have been used for decades.

4.6 Parameter settings

The standard PIPENET module has been used in the PIPENET cooling medium model. In the HYSYS compressor train model, the Peng-Robinson equation of state model has been used.

4.6.1 Parameter settings for fail open

The following parameter settings were used to obtain the gas and cooling medium temperatures downstream the heat exchangers and to evaluate if fail open was going to affect the other cooling medium consumers:

1st stage recompression suction cooler:

- $c_{p(gas)}$ = 2.05J/(kgK)
- U = 846.1 W/(m^2K)
- A = 373.1m^2
- $P_{gas(out)}$ = 12.49Bara
- F = 0.988

Dehydration suction cooler:

- $c_{p(gas)}$ = 2.43J/(kgK)
- U = 851.2 W/(m^2K)
- A = 384.1m^2
- $P_{gas(out)}$ = 42.8bara
- F = 0.988

2nd stage after-cooler:

- $c_{p(gas)}$ = 3.20kJ/(kgK)
- U = 1949 W/(m^2K)
- A = 307,3m^2
- $P_{gas(out)}$ = 148bara
- F = 0.783

The following parameter settings were the same for the three heat exchangers for all the simulations:

- $T_{CW(in)} = 15^{o}C$
- $\rho_{CM} = 1044 kg/m^3$
- $\rho_0 = 1000 kg/m^3$
- $c_{p(CM)} = 3.71 kJ/(kgK)$

The heat transfer areas and heat transfer coefficient are the same as the ones above for all the simulations.

4.6.2 Parameter settings for fail close with travel stop

The parameter settings for studying fail close with travel stop as the failure position were as follows:

1st stage recompression suction cooler:

- Valve openings of 10, 15 and 20% were simulated

The remaining parameters were equal to the 1st stage recompression suction cooler parameters from the fail open case.

Dehydration suction cooler:

- Valve openings of 10, 15 and 20% were simulated

The remaining parameters were equal to the dehydration suction cooler parameters from the fail open case.

2nd stage after-cooler:

- Valve openings of 10, 15 and 20% were simulated
- $c_{p(gas)} = 2.85 kJ/(kgK)$

The remaining parameters were equal to the 2nd stage after-cooler parameters from the fail open case.

4.6.3 Parameter settings for fail in position

The openings that correspond to the normal operation point were used for the calculations done for the evaluation of fail in position as the failure position. The following parameters were used:

Dehydration suction cooler:

- Valve opening of 57% was simulated

The remaining parameters were equal to the dehydration suction cooler parameters from the fail open case.

2nd stage after-cooler:

- Valve opening of 75% was simulated

The remaining parameters were equal to the 2nd stage after-cooler parameters from the fail open case.

5. Results

The following chapters show the results of the calculations and simulations connected to the different failure scenarios. Temperatures are given in ^{o}C, pressures are given in bara, mass flows are given in kg/s while the heat exchanger duties are given in kW. Tag numbers were used, and 41-00001 represents the 1st stage recompression suction cooler, 41-00004 represents the dehydration suction cooler while 41-00002/3 represents the 2nd stage after-cooler.

5.1 Results for fail open

The following table shows the heat exchangers inlet and outlet temperatures for the gas and the cooling medium. The mass flow of cooling medium and gas through the heat exchangers, the heat exchangers gas outlet pressure, the supply pressure to the cooling medium system ($P_{PIC(out)}$) and the exchanged heat duties can also be seen in the table.

	41-00001	41-00004	41-00002/3
$\dot{m}_{CM}$	86.24	86.07	127.20
$\dot{m}_{gas}$	36.58	35.50	44.19
$T_{CM(in)}$	15	15	15
$T_{CM(out)}$	17.9	46.0	45.1
$T_{gas(in)}$	27.62	135.3	123.1
$T_{gas(out)}$	15.4	20.8	22.6
$P_{PIC(out)}$	6.26	6.31	6.43
$P_{gas(out)}$	12.49	42.8	148
q_{HX}	915	9887	14200

Table 1 Results fail open

The temperatures get significantly lower in the failure position of fail open compared to normal operation; therefore the possibilities for hydrate formation must be evaluated. Figures 16 to 18 show the phase envelopes and the hydrate curves for the gas outlet line of the 1st stage recompression suction cooler, dehydration suction cooler and 2nd stage after-cooler respectively, found from HYSYS. The black dot shows where in the phase envelope the gas is at the heat exchanger outlet.

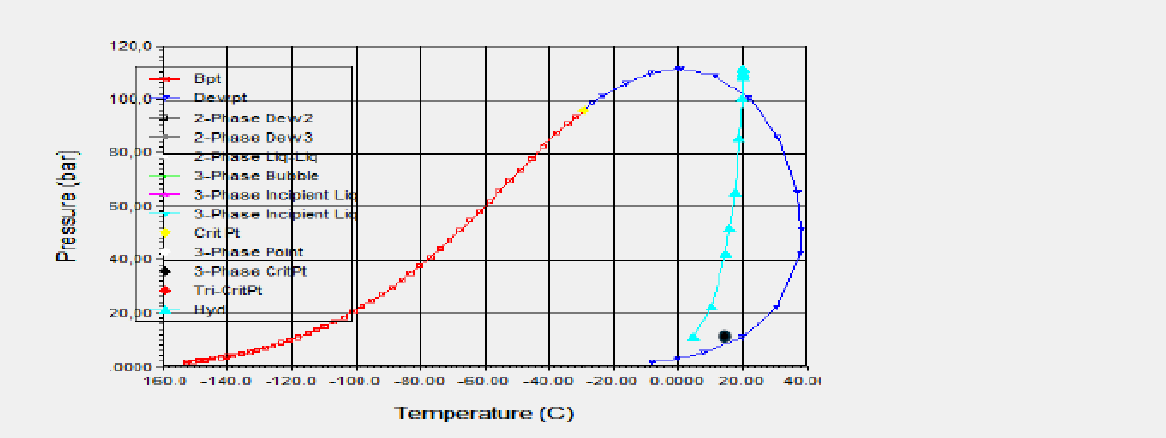

Figure 16 Phase envelope and hydrate curve for the gas composition at the 1st stage recompression suction cooler outlet

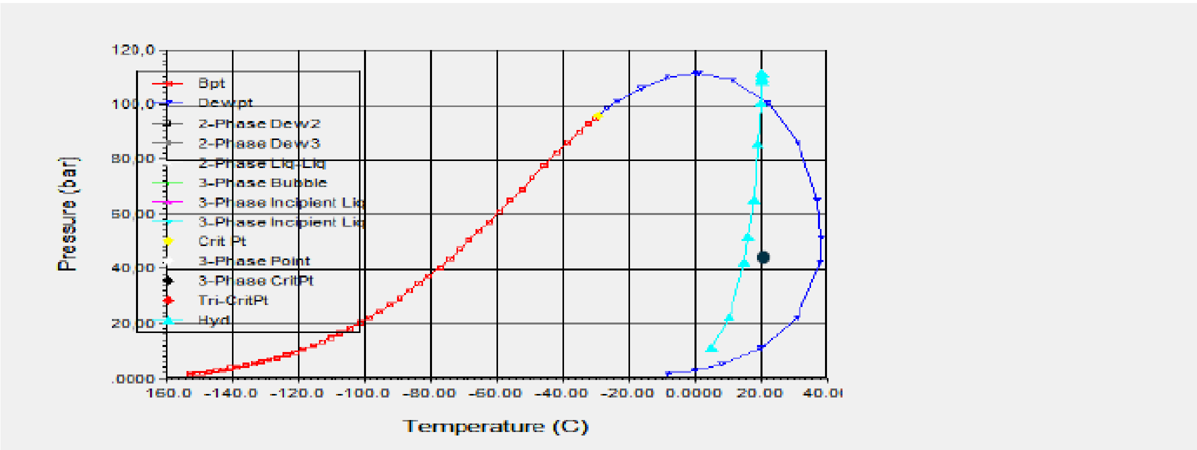

Figure 17 Phase envelope and hydrate curve for the gas composition at the dehydration suction cooler outlet

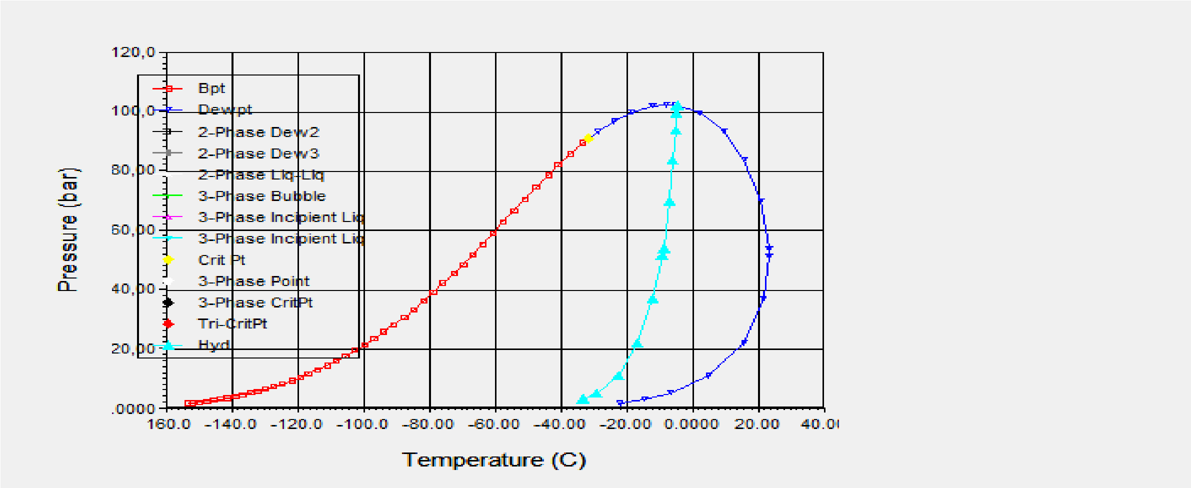

Figure 18 Phase envelope and hydrate curve for the gas composition at the 2nd stage after-cooler outlet

The light blue curve in the three figures above is the hydrate curve. As the figures show, the x-axis shows temperature in ^oC and the y-axis shows pressure in bara.

The black dots at Figure 16 and 17 are both close to the hydrate curve. They are just outside the area where hydrate formation occurs. From the gas outlet temperature and pressure of the 2nd stage after-cooler it is confirmed in Figure 18 that hydrate formation will not be a problem.

5.2 Results for fail close

The following table shows the cooling medium- and gas heat exchanger inlet and outlet temperatures, the mass flows through the heat exchangers, the valve pressure out of the three control valves, the boiling temperatures of the cooling medium at the given conditions ($T_{boil(CM)}$) and the gas high high temperatures ($T_{HH(gas)}$) downstream the heat exchangers that will cause a process shutdown. The boiling temperatures were found from Figure 15. The valve outlet pressure were found from PIPENET at valve openings of 10%, as the cooling medium will not start boiling before the valve opening approaches fully closed.

Until the valves are closed, there will be cooling medium flowing through the heat exchangers, and therefore also exchanged heat duties. The reason why cooling medium temperatures are given is to show that they approach the gas inlet temperature when the cooling medium flow gets low.

	41-00001	41-00004	41-00002/3
$\dot{m}_{CM}$	0	0	0
$\dot{m}_{gas}$	36.58	35.50	44.19
$T_{CM(in)}$	15	15	15
$T_{CM(out)}$	27.62	135.3	123.1
$T_{gas(in)}$	27.62	135.3	123.1
$T_{gas(out)}$	27.62	135.3	123.1
$T_{HH(gas)}$	51.7	46.1	55.0
$P_{valve(out)}$	2.07	2.03	2.04
$T_{boil(CM)}$	130	127	128
q_{HX}	0	0	0

Table 2 Results fail close

Only the boiling temperatures out of the control valves are given here and for the results of fail close with travel stop. The reason for this is the fact that the inlet pressures of the valves are significantly higher (around 4.7bara), and it can be seen from Figure 15 that boiling cooling water will not be an issue at those conditions.

5.3 Results for fail close with travel stop

The following tables show the heat exchanger gas- and cooling medium outlet temperatures, mass flows of cooling medium and gas, the control valve outlet pressures, the gas high high temperatures, the cooling medium boiling temperatures and the exchanged heat duties. The simulations were done at valve openings of 10%, 15% and 20% to be able to see how much a change in opening affects the gas- and cooling medium outlet temperatures. The boiling temperatures were found from Figure 15, while the valve outlet pressures were found from PIPENET.

	41-00001	41-00004	41-00002/3
Opening	0.1	0.1	0.1
$\dot{m}_{CM}$	4.56	4.34	5.83
$\dot{m}_{gas}$	36.58	35.50	44.19
$T_{CM(in)}$	15	15	15
$T_{CM(out)}$	27.619	135.3	123.089
$T_{gas(in)}$	27.62	135.3	123.1
$T_{gas(out)}$	24.8	112.9	104.5
$T_{HH(gas)}$	51.7	46.1	55.0
$P_{valve(out)}$	2.07	2.03	2.04
$T_{boil(CM)}$	130	127	128
q_{HX}	213	1939	2339

Table 3 Results from a valve opening of 10%

	41-00001	41-00004	41-00002/3
Opening	0.15	0.15	0.15
$\dot{m}_{CM}$	6.97	6.74	8.93
$\dot{m}_{gas}$	36.58	35.50	44.19
$T_{CM(in)}$	15	15	15
$T_{CM(out)}$	27.617	135.291	122.667
$T_{gas(in)}$	27.62	135.3	123.1
$T_{gas(out)}$	23.3	100.5	94.8
$T_{HH(gas)}$	51.7	46.1	55.0
$P_{valve(out)}$	2.07	2.03	2.04
$T_{boil(CM)}$	130	127	128
q_{HX}	326	3010	3566

Table 4 Results from a valve opening of 15%

	41-00001	41-00004	41-00002/3
Opening	0.2	0.2	0.2
$\dot{m}_{CM}$	9.82	9.91	12.01
$\dot{m}_{gas}$	36.58	35.50	44.19
$T_{CM(in)}$	15	15	15
$T_{CM(out)}$	27.540	134.854	120.741
$T_{gas(in)}$	27.62	135.3	123.1
$T_{gas(out)}$	21.5	84.3	85.7
$T_{HH(gas)}$	51.7	46.1	55.0
$P_{valve(out)}$	2.07	2.03	2.04
$T_{boil(CM)}$	130	127	128
q_{HX}	457	4412	4711

Table 5 Results from a valve opening of 20%

The cooling medium outlet temperature was given with three decimals to be able to see the trends. The cooling medium outlet temperature of the dehydration suction cooler at a valve opening of 10% was rounded to 135.3^{o}C by Excel.

5.4 Results for fail in position

Table 6 shows the heat exchangers inlet and outlet gas and cooling medium temperatures at normal operating conditions. It also shows the exchanged heat duty.

Table 7 shows how much the mass flow of gas needs to increase to reach a gas temperature equal to the high high temperature when the exchanged heat duty were equal to the one found in Table 6.

Table 8 shows how much the mass flow of gas through the dehydration suction cooler needs to decrease to reach the gas temperature which is equal to the temperature that gives hydrate formation at the given gas outlet pressure. The exchanged heat duty was kept equal to the one found in Table 6.

As the gas temperature cannot decrease below the cooling medium temperature of 15°C it can be seen from Figure18 that hydrate formation will never occur at the 2nd stage after-cooler outlet.

	41-00004	41-00002/3
Opening	0.57	0.75
$\dot{m}_{CM}$	61.21	91.17
$\dot{m}_{gas}$	35.50	44.19
$T_{CM(in)}$	15	15
$T_{CM(out)}$	57.7	53.1
$T_{gas(in)}$	135.3	123.1
$T_{gas(out)}$	23	32
$T_{HH(gas)}$	46.1	55.0
$T_{hydrate}$	16	0
$P_{gas(out)}$	42.8	148
q_{HX}	9700	12883

Table 6 Results of fail in position at normal valve operation

	41-00004	41-00002/3
Opening	0.57	0.75
$\dot{m}_{gas}$	44.7	59.1
$T_{gas(in)}$	135.3	123.1
$T_{gas(out)}$	46.1	55.0
$T_{HH(gas)}$	46.1	55.0
$P_{gas(out)}$	42.8	148
q_{HX}	9700	12883

Table 7 Results of fail in position when the mass flow of gas is increased

	41-00004
Opening	0.57
$\dot{m}_{gas}$	33.42
$T_{gas(in)}$	135.3
$T_{gas(out)}$	16
$T_{hydrate}$	16
$P_{gas(out)}$	42.8
q_{HX}	9700

Table 8 Results of fail in position when the mass flow of gas is decreased

6. Discussion

From the results of fail open it can be seen that if any of the three valves are 100% open, the supply pressure to the cooling medium system drops below 6.5bara. The consequence of this is that there will be a deficit of cooling medium to the other cooling medium consumers. This could lead to a process shutdown to avoid overheating of process equipment.

When it comes to hydrate formation of the fail open failure scenario, the results show that both the 1st stage recompression suction cooler- and the dehydration suction cooler gas outlet stream are close to having hydrate formation, as the black dots indicated in Figures 16 and 17 show. A small decrease in the gas flow rate or a change in the gas composition can give temperature and pressure conditions that results in hydrate formation. Low gas flow rates are very likely in case of low gas lift requirements.

It cannot be concluded that hydrate formation will occur, but as both the 1st stage recompression suction cooler- and the dehydration suction cooler outlet are close to having hydrate formation, it cannot be concluded that hydrate formation will not occur either. If hydrate formation is present, a hydrate plug can in worst case be entrained from the pipe wall and flow along with the gas to the compressor. As this can destroy the compressor it is required to be sure that hydrate formation is not possible before fail open can be an alternative as the failure position.

Another reason for not using fail open as the failure position is the fact that the mass flow of cooling medium going through the dehydration suction cooler in a fail open failure position is 86.07kg/s, which corresponds to 309 852kg/h. This is almost 50% more than 210 000kg/h, which is the maximum allowed cooling medium flow through the dehydration suction cooler to avoid vibrations. Fail open is therefore not an alternative failure position for the globe valve that controls the cooling medium flow through the dehydration suction cooler.

When it comes to fail close as the failure solution, the 1st stage recompression suction cooler is not affected. The reason is that a heat exchanger at Odin cools the gas to its required 1^{st} stage recompression suction cooler temperature. The recompression suction cooler and the 2^{nd} stage after-cooler are on the other hand affected by fail close as the failure position. The gas outlet temperatures will increase to above the high high temperature that will lead to a shutdown of the gas compressor train. As a consequence of this, the gas turbine stops producing power and the gas lift, gas injection and water injection at Loke stops. That the power production stops will not result in the whole oilfield being without electricity as it has an emergency power solution. A consequence of no gas to the gas injection, gas lift and water injection is however reduced oil production, which gives economic losses for the operator.

Another issue with fail close is the risk of boiling of the cooling medium. From the results it can be seen that boiling of cooling water seems to occur at the outlet of the globe valve that controls the dehydration suction cooler. The cooling medium at the outlet of the valve that controls the 2^{nd} stage after-cooler will also get close to boiling with fail close as the failure position. The consequence of boiling the cooling water is a sudden expansion of the cooling medium which again leads to a sudden pressure increase that can burst the rupture disk and force the cooling medium to closed drain. If the rupture disk bursts, the system cannot get up and running again before a new rupture disk has been installed. In addition to this the reason why the control valve failed must be found, and the problem needs to be fixed. Of course, the problem needs to be fixed, whatever the failure position is.

Fail close with travel stop as the failure position gave results that are very similar to the case of fail close, especially when it comes to the heat exchangers cooling medium outlet temperature. Boiling cooling water seems to occur at the outlet of the globe valve that controls the dehydration suction cooler also for fail close with travel lock as the failure solution. Even at a valve opening percentage of 20%, which is a higher travel stop opening than any of the valves can have, the cooling medium outlet temperatures were only marginally lower than for the failure position of fail close. The reason is that the mass flow of water is very small compared to the mass flow of gas. As a consequence the water will have a very big temperature change from the heat exchanger inlet to outlet compared to the gas. The dehydration suction cooler and 2^{nd}

stage after-cooler outlet temperatures were above the high high temperatures, even though they were noticeably lower compared to fail close.

The minimum openings of the dehydration suction cooler and the 2nd stage aftercooler in normal operation are as mentioned 10% and 17% respectively. This is equal the maximum travel stop opening. From Table 5 it can be seen that the gas temperatures are significantly higher than the high high temperature even at valve openings of 20%, which means that the gas flow rate must reduce significantly before a process shut down will be avoided for the failure position of fail close with travel stop.

How the actuator fails is what determines whether or not the control room will get an alarm that it has failed for the case of fail close with travel stop. If the electrical signal of the controller or the solenoid valve is lost, the control room will get an alarm. On the other hand the control room will not get an alarm if air supply pressure for some reason fails. A normal position to use when setting the travel stop is at the minimum opening the valve has in normal operation. A solution could be to set the travel stop 2-3% lower than the minimum opening and program so the control room gets an alarm if the valve operates below the minimum opening. This will ensure that the control room will get an alarm whatever the reason for the valve failure is.

Fail in position is the failure solution that from a process perspective is the most optimal if the gas flow is kept relatively constant and the actuator solution is working. Then the process will be kept much as it was before the valve failure, which will give the operator more time to fix the problem compared to the other failure positions. The temperatures and therefore also the gas flow must change significantly to get a gas temperature that is above the high high temperature, causing shutdown of the gas compressor train. On the other hand, the gas flow cannot change very much before hydrate formation will occur, as shown in Table 8. If one well stops producing or reduced gas lift may be enough to cause this. As for fail close with travel stop, the way in which the actuator fails is what determines whether or not the control room will get a message that the actuator has failed. If the actuator fails, the system can in theory run for days before it is discovered. A possible solution to this can be to program that the control room will get an alarm if the valve keeps the exact same position over a certain amount of time.

There are however some uncertainties related to the fail in position actuator solution. While fail open and fail close are well known and proved failure positions, fail in position is a relatively new solution. Because of this it is hard to get any feedback on how well the actuator solution works.

The actuator solution connected to fail in position introduces two more components compared to fail close and fail open; the solenoid valve and the snap acting relay. In addition to this, extra tubing between the filter regulator and the solenoid valve is needed. More components increase the chances for the actuator to fail.

If the electrical signal of the controller is lost the valve will lose its control as the controller will not know if more or less pressurized air is needed. If the actuator fails, the solenoid valve needs to respond instantly. If not the pressurized air from the controller will keep flowing through the snap acting relay and make the valve drift until the solenoid valve responds. This can result in much more or less cooling medium through the valve compared to what it would have been if the solenoid responded instantly. In worst case this can result in hydrate formation or a process shut down.

There is also the question whether the fail in position actuator solution are able to keep the valve in the position where it failed. Especially for the double acting actuator solution presented in Figure 7 this is uncertain, as pressurized air on both sides of the piston is the only thing that ensures that the valve remains in its failure position.

7. Conclusion and further work

7.1 Conclusion

The only conclusion that can be made without any doubt is that fail open is not the optimal failure solution. As a consequence of fail open, other cooling medium consumers will have a cooling medium deficit, which could lead to a process shut down. In addition to this, fail open could give hydrate formation, which in a worst case can damage the gas compressor downstream the heat exchangers. For lower gas rates the consequences of fail open as the failure position will be even bigger due to a reduced gas temperature, which increases the possibilities for hydrate formation.

Even though fail in position is the best solution from a process perspective, it will not be chosen as the most optimal failure solution. There are too many uncertainties connected to the actuator solution. In a few years more feedback will probably be available concerning the various fail in position actuator solutions. If they function well over time, fail in position should be considered very seriously in similar future valve installations. An important criterion, however, is that the control room will get an alarm whatever the failure scenario is. A possible solution suggested is to program that the control room gets a message if the valve has the exact same opening over a certain amount of time.

The results show that the failure positions of fail close and fail close with travel stop will both give a heat exchanger gas outlet temperature above the high high temperature, and also give boiling of the cooling water. As the failure position of fail close with travel stop does not offer any advantages compared to fail close, it is not considered the optimal failure position. Fail close is therefore the failure position that is recommended based on the analyses presented here. If a failure scenario occurs, there will be a process shut down if the problem is not solved very quickly, which is not likely. This will result in lost production for the operator, which means economic losses. The actuator solution is trustable, and no process equipment will get damaged if fail close is chosen as the failure position.

7.2 Further work

Further work that is needed:

- Get feedback on how fail in position works at other installations
- Investigate if there are other fail in position actuator solutions that could be considered
- Find a method to let the control room know when the actuator has failed whatever the reason for the failure is

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