

DYNAMIC ANALYSIS OF COMPRESSOR TRIPS IN THE SNØHVIT LNG REFRIGERANT CIRCUITS

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ABSTRACT

Compressors are key components in the refrigerant circuits of the Snøhvit LNG plant and contain large amounts of mechanical energy. Thus it is imperative that the control system is able to keep the compressor out of surge in case of driver trip. A dynamic process simulator describing the total LNG plant has been developed by Kongsberg Process Simulation and the simulator has been applied in the engineering phase for the design and process verification. The simulator has also been used to verify the robustness of the closed loop refrigerant circuits and to verify that the refrigerant compressors are sufficiently protected after a driver trip. The presented work demonstrates the value of dynamic simulations for verification of compressor protection systems. In addition it shows the importance of using correct data for polar inertia of the rotating equipment, as well as the opening and dead times of the anti-surge valves. It is recommended to include a sensitivity analysis of these parameters as part of plant verification studies.

INTRODUCTION

Compressors are key components in the refrigerant circuits of LNG plants. It is imperative that these compressors are properly protected against surge. Compressors in surge can result in a severe damage to machinery and disturb plant regularity.

One of the most serious incidents that can happen to a compressor system is a driver trip. Gas turbines and electrical motors are the most common types of compressor driver and a trip of the driver here is more serious than a process trip. This is because in a process trip the shutdown of the driver itself can be delayed until the surge protection valves have been opened.

Important factors controlling the run down of the compressor after a trip are opening times of the anti-surge valves, piping layout, inlet and discharge volumes as well as the polar inertia of the rotating equipment including the compressor, gear and driver. It is essential to be able to validate that the compressor is properly protected during process upsets including the case of a driver trip.

The use of dynamic models for analysing compressor transient behaviour has been addressed by several authors in various scientific journals and conference papers and has also been demonstrated by the supplier industry. Patel [1] discusses dynamic simulators as an important tool in the design and operation of compressor systems. A good overview of the scientific research within this area is given in [2].

Dynamic simulation has been used as a tool for studying and training in the oil and gas industry since late 1980s. StatoilHydro has now around 20 industrial simulators that are customized to offshore oil and gas production as well as onshore gas plants and refineries. The trend has shifted from having two different simulators, one for training and one for engineering, to one common simulator for both training and engineering studies. The last is the case for the Snøhvit LNG plant where Kongsberg Process Simulation [3] has developed a multi-purpose simulator. The simulator is in use today for operator training and process analysis and has been applied in the work presented here.

The Snøhvit LNG plant is a highly integrated process and accordingly, the simulator is a complex dynamic model. The dynamic simulator from Kongsberg Process Simulation includes the whole Snøhvit process plant and all main process equipment from subsea facilities to LNG off-loading [4]. In the engineering phase of the plant this simulator was applied to analyse and verify the operation of the refrigerant circuits and compressors. This paper presents some of the results.

Related work has been performed by others using independent compressor models. A study of transients during emergency shutdown (ESD) showed the importance of the placement of the recycle valve and signal delay time to avoid surge during compressor run-down [5]. Other studies [6] and [7] have reported the experience from the analysis of compressors and driver behaviour during run-down for Troll Kollsnes gas treatment plant compressors. Morini [2] developed a one-dimensional dynamic compressor model based on first principles for simulation of transient behaviour. This was later the basis for the study of a complete compressor station [8]. Dynamic simulations of the Snøhvit compressors using independent models are presented in [9].

In this paper, a total plant simulator has been utilized in the analysis of the refrigerant compressors. In the case of one compressor trip the effect on other process components can be studied. The compressor models are based on ASME Performance Test Code. These procedures are designed for testing compressors, but they are also well suited as a basis for a simulation model. Moreover, the ability of the Snøhvit compressor protection system to keep the run-down out of surge after a driver trip is studied using real plant data.

SNØHVIT LNG PLANT

The Snøhvit LNG plant was brought on stream in September 2007. It is the first base-load LNG production and exporting facility in Europe [10]. Located near the city of Hammerfest at 71°N, it is also the world's most northerly base-load LNG plant. Gas production will be 6.9 billion cu m/year (bcmy), from which 4.3 million tonnes/year (tpy) of LNG will be produced.

The well stream from the offshore fields Snøhvit, Albatross and Askeladden is transported in a 143 km pipeline to the LNG plant. The stream first passes through the slug catcher, in which liquid slugs are buffered and an initial separation of the three phases - natural gas, condensate, and a water/monoethylene glycol (MEG) mixture-takes place.

Natural gas passes to the inlet facilities, where the remaining liquids, CO₂, water, and mercury are removed from the gas. Some gas is removed from the flow for use as fuel gas. The CO₂ is compressed and piped back to the Snøhvit field for injection into the reservoir.

The lighter gas passes through a cascade of three cooling stages, each with its own composition of refrigerants: Precooling to -50° C, Liquefaction to -80° C and Subcooling to -155° C. The temperature of the liquid gas is then reduced to -163° C by passage through an expansion turbine, which reduces the pressure of the gas. Finally the gas is piped to one of the two storage tanks. A schematic overview is shown in Fig. 1.

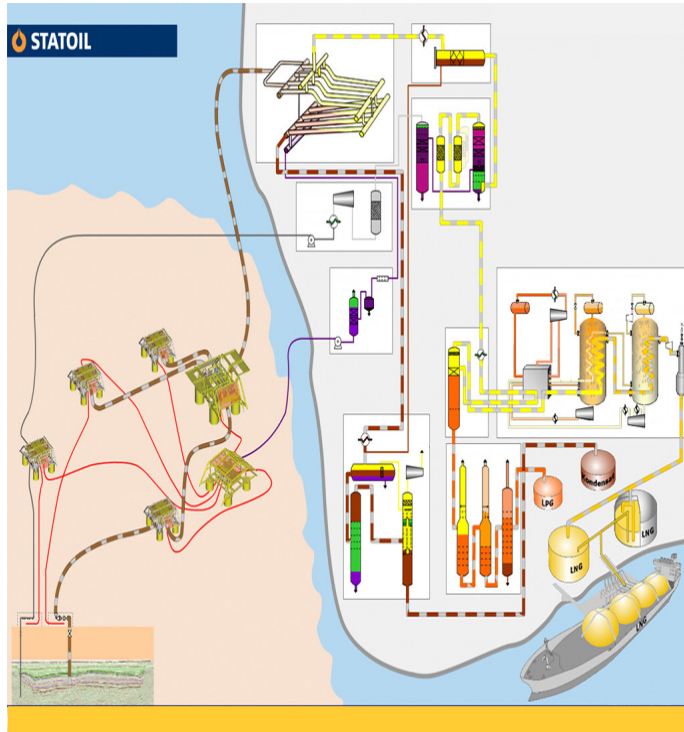


Figure 1. The Snøhvit LNG plant

The compressors are the key components in the refrigerant circuits. The Snøhvit compressors are run by electric variable-speed motors, this is the first use of all-electric

drive in an LNG plant. The Precooling cycle has a 65 MW side stream compressor train, Liquefaction a 32 MW inline compressor train, and Subcooling a 65 MW tandem casing compressor train.

This implies the need for efficient and robust anti-surge control systems regarding i) safety of plant personnel, ii) damage to machinery in case of failure and iii) process regularity. It is therefore essential to be able to validate the ability of the compressor protection system to keep the compressor run-down out of surge in the case of a driver trip. The robustness of large refrigerant circuits is not easily validated in practice. Steady-state models are often developed during the design phase but these cannot be used to predict dynamic transients or to validate the system robustness during start-up or shutdown.

SIMULATION MODEL

Kongsberg Process Simulation (previously Fantoft Process Technologies) has developed a dynamic process simulator for the Snøhvit LNG plant, describing the total value chain from the multiphase upstream flow to the loading of LNG tankers. This simulator has been used in the whole engineering phase and is also being applied for operator training. The simulator was built using the D-SPICE dynamic simulation tool.

The total LNG plant was constructed as a so-called distributed model. The model is split into smaller independent models, called sub-models. These sub-models can be solved simultaneously, but using different CPUs. D-SPICE has a built-in technology that coordinates the integration of sub-models. Cameron [11] present this technology in detail while Skjerven [4] describe the total Snøhvit model. Each sub-model can be used as a stand-alone simulator. In that case it must be implicitly assumed that the interactions with other sub-models are negligible or irrelevant or boundary conditions can be defined.

In the presented work the refrigerant section of the plant is in focus. This section corresponds to one of the ten sub-models in the total model. A schematic overview of the section is shown in Fig. 2.

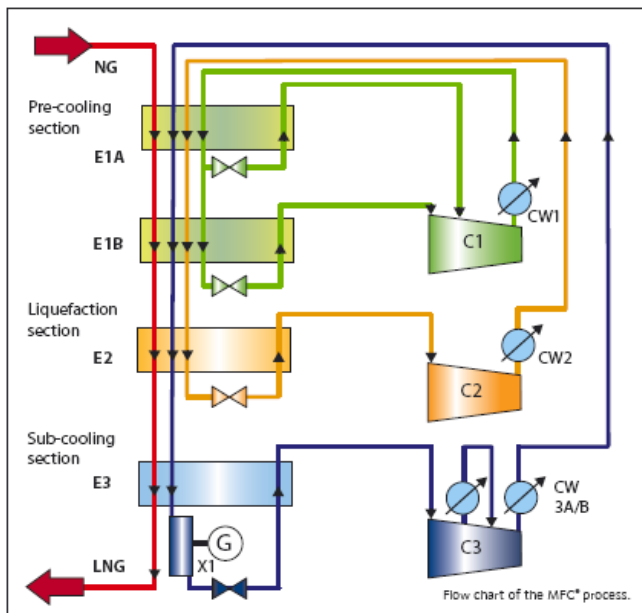


Figure 2. Schematic overview of the refrigerant circuits

The sub-model is more complicated than shown in the schematic overview. It consists of 1311 modules and is the second largest sub-model in the plant.

Process description

The refrigeration duty for production of the liquefied natural gas is provided by three closed loop mixed refrigerant cycles: A pre-cooling cycle, a liquefaction cycle and a sub-cooling cycle. Separate compressors driven by electric motors are installed for each cycle.

The liquefaction cycle has a single stage compressor. The other two cycles have a two-stage compressors running on single shafts.

Thermodynamic Model

The thermodynamic model chosen for the LNG as well as for the refrigerants was InfoChem's advanced version of the Soave Redlich Kwong equation of state (SRK-EOS) [12]. This version implements the so-called Peneloux density correction and a similar correction for vapour pressure.

D-SPICE

D-SPICE is a modular dynamic simulation tool programmed using an object-oriented programming language. A process model is built up using smaller building blocks, called modules. The flow of information between the modules is configured using input and output variables.

Modularity

The modules in D-SPICE usually correspond to process units or controller units. Examples of such modules are compressor, valve, vessel, pump and PID controller. This

level of modularity enables the user to work with a model in an intuitive way.

Some of the modules in D-SPICE depart from this principle as they represent only a part of a process unit. Examples of such modules are column tray and heat exchanger stream. This level of modularity gives better flexibility and allows detailed model constructions.

Numerical Strategy

There are two fundamentally different ways to solve a modular-based simulation model. One is the so-called simultaneous modular approach, where all the equations are solved simultaneously in a common equation solver. The other way is the sequential modular approach where the modules integrate individually over the time step. D-SPICE uses a technique that combines these two methods. The equations that relate pressure and flow are solved in a common equation solver, called pressure-flow network solver. All other equations are solved independently within each module.

Pressure-Flow Network. A process plant is modelled as a lumped system. Vessels, manifolds and pipe sections are treated as volumes. Between the volumes there are lumped flow restrictions.

Thermodynamic Properties and Phase Equilibrium

D-SPICE uses a local thermodynamic model in a tabular form to speed up the calculations. The thermodynamic property tables are built based on a thermodynamic property package called Multiflash delivered by InfoChem Ltd. A large collection of thermodynamic models are available in this package, among them are the Soave Redlich Kwong and the Peng-Robinson equation of state.

The data tables are created off-line over a specified range of pressure and temperature for a specified composition. The result is stored in specific files. Partial molar properties for each component are saved separately for the gas and liquid phase.

The algorithms that are used for look-up in the tabulated data are in fact simplified flash algorithms, so-called K-factor flashes, which essentially solve the Rachford-Rice equation [13] using K-values looked up from the table.

The Compressor Section Module

This module simulates a compressor section (centrifugal or axial). If a compressor has side streams or inter cooling it should be modelled by using two or more compressor sections together with a pressure node and (if needed) a heat exchanger between them. The mathematical model is based on the procedures described in ASME PTC 10 [14]. These procedures are designed for testing compressors, but they are also well suited as a basis for a simulation model. The compressor section is thus simulated using performance curves in the form of polytropic work coefficient and polytropic efficiency versus speed and flow coefficient.

Performance Curves. Performance curves are entered as a set of points that give polytropic head and polytropic efficiency as functions of suction volume flow conditions for n different rotor speeds.

$$H_p = F_i(Q_s) \quad i = 1, 2 \dots n \quad (1)$$

$$\eta_p = G_i(Q_s) \quad i = 1, 2 \dots n \quad (2)$$

where

H_p = polytropic head, J/kg

η_p = polytropic efficiency, dimensionless

Q_s = Volumetric flow at suction conditions, m³/s

These raw data points are converted to dimensionless groups and cubic spline equations are fitted to the resultant data. In this way, an accurate representation of compressor performance can be given at all operating points.

$$\mu_p = f_i(\phi) \quad i = 1, 2 \dots n \quad (3)$$

$$\eta_p = g_i(\phi) \quad i = 1, 2 \dots n \quad (4)$$

where

μ_p = polytropic work coefficient = H_p / N^2

ϕ = flow coefficient = Q_s / N

N = Rotor speed

The use of dimensionless groups makes the mathematical model less dependent on the gas composition and inlet conditions.

The polytropic work coefficient and the flow coefficient as defined above are not fully dimensionless. The size of the compressor (diameter) would be required to make the groups completely dimensionless, but it is taken out since it is always a constant. Since the variables only are used to compare states of the same machine the diameter would always cancel out in the end, and it would therefore only give extra work for the user to provide this parameter.

ASME Methods. If the gas composition or inlet conditions differ greatly from those that the performance curves represent, the module can be configured to compensate for it by considering the volume ratio as the fourth dimensionless group. This is done by a similar approach to that described in [14]. If this method is chosen the module will perform a polytropic head calculation for the reference gas, $H_{p,r}$. This is the gas composition and inlet conditions the performance curves represent. The speed for performance map look-up is then chosen as:

$$N_r = N \sqrt{\frac{H_{p,r}}{H_p}} \quad (5)$$

The method depends on performance data in the form of polytropic head and polytropic efficiency as function of suction volume flow for various speeds. A single speed

curve is not sufficient, since all the extra information used by this method lies in how the performance changes with speed. This method also depends on information about the gas composition and inlet conditions that the performance curves represent.

Surge and Stonewall. The surge limit line and stonewall limit line are set using spline fits in a similar way between the endpoints of the performance curves. If the surge region is defined, a qualitative representation of oscillating flow is modelled.

Assumptions and simplifications.

- When the interpolation parameter (for example speed) is outside the region where data are given, the nearest curve is used and fan laws are applied.
- When the operating point is to the right of the performance curves (Stonewall), the performance curves are linearly extrapolated.
- When the operating point is to the left of the performance curve (Surge), the flow is calculated by using a simple valve equation, but linearly transformed, so that the minimum point on this parabola is at a user-defined position. If there is positive flow the polytropic efficiency is calculated as the value at the surge limit.

Surging Flow Calculation. The surge flow behaviour is configured using three parameters: A surging flow factor, an admittance factor and a maximum flow gradient. When the polytropic head is higher than the surge limit, the flow is calculated by using a simple valve equation. If the surging flow factor is 100 % the end point of this valve equation will coincide with the start point on the performance curve, i.e. the surge limit. If the surging flow factor is 0 % then the valve equation is shifted to the left, so that the end point is at flow = 0. The flow is calculated from the excess head, i.e. how much extra head you have compared to the head on the surge limit.

$$Q_s = f_Q Q_s^* - Av \sqrt{H_p - H_p^*} \frac{\rho_d}{\rho_s} \quad (6)$$

where

Q_s = calculated flow when surging, m³/s

Q_s^* = volumetric suction flow at the surge limit, m³/s

f_Q = surging flow factor

Av = admittance factor

H_p^* = polytropic head at the surge limit, J/kg

ρ_d = discharge density, kg/m³

ρ_s = suction density, kg/m³

If the surging flow factor is set less than 100 % there will be a discontinuity in the flow equation. When the compressor is surging the flow calculations will therefore be unstable. Setting the surging flow factor to 100 % will ensure stable flow calculations.

Maximum Flow Gradient. If the maximum flow gradient is set, the surging flow admittance will be limited by this factor. Hence, if the surging flow is 100 % and the admittance factor is large, the surging flow will be given by the maximum flow gradient only. The surging flow characteristic will then be a straight line as indicated by the red line in Fig. 3.

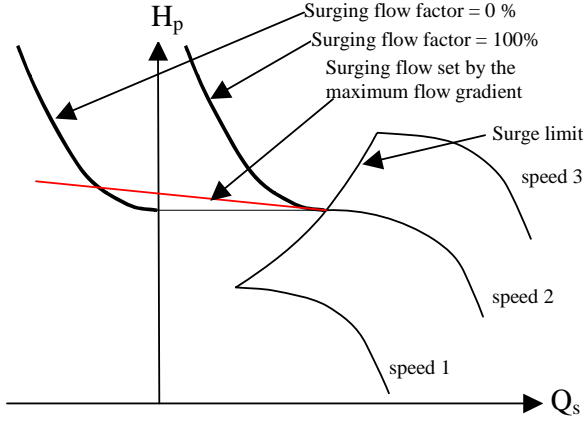


Figure 3. Configuring surge flow calculation

Calculation of Polytropic Head. The user can select one of three methods for calculating the polytropic head: Suction gas properties, Average suction/discharge properties, and HS (Mollier) integration. The HS-integration method is the most accurate method, but it depends on high quality thermodynamic property calculations. If a table lookup method is used and there is a long distance between the pressure/temperature points, it may be a better choice to use the Average suction/discharge properties method. If Suction gas properties or Average suction/discharge properties is chosen, the polytropic head equation is used:

$$H_p = \frac{n}{n-1} f p_i v_i \left(\left(\frac{p_d}{p_i} \right)^{\frac{n-1}{n}} - 1 \right) \quad (7)$$

where

- p_i = inlet pressure, Pa
- p_d = discharge pressure, Pa
- v_i = inlet specific volume, m^3/kg
- n = polytropic volume exponent, dimensionless
- f = polytropic work factor, dimensionless

The polytropic work factor, f , is calculated based on isentropic change of state:

$$f = \frac{h'_d - h_i}{\left(\frac{n_s}{n_s - 1} \right) (p_d v'_d - p_i v_i)} \quad (8)$$

where

- h_i = inlet enthalpy, J/kg
- h'_d = isentropic discharge enthalpy, J/kg
- n_s = isentropic volume exponent, dimensionless
- v'_d = isentropic discharge specific volume, m^3/kg

For a real gas the isentropic volume exponent is not the same as the heat capacity ratio C_p/C_v , and it is not constant along the compression path. The isentropic volume exponent is calculated in the same way as the polytropic volume exponent, giving an average value:

$$n_s = \frac{\ln \frac{p_d}{p_i}}{\ln \frac{v_i}{v'_d}} \quad (9)$$

The purpose of the polytropic work factor is to compensate for having a varying volume exponent. It is calculated using an isentropic analysis and then applying the assumption that the varying n affects H_p just as the varying n_s affects the isentropic head.

Average Suction/Discharge Properties. If Average suction/discharge properties is chosen then the polytropic volume exponent is calculated as:

$$n = \frac{\ln \frac{p_d}{p_i}}{\ln \frac{v_i}{v_d}} \quad (10)$$

Suction Gas Properties. If Suction gas properties is chosen then the polytropic volume exponent is calculated by using the Schultz method [15]:

$$n = \frac{1}{Y - m(1 + X)} \quad (11)$$

where

$$Y = \frac{p}{v} \left(\frac{\partial v}{\partial p} \right)_{T,i}$$

$$X = \frac{T}{v} \left(\frac{\partial v}{\partial T} \right)_{p,i} - 1$$

$$m = \left(\frac{ZR}{C_p} \right)_i \left(\frac{1}{\eta_p} + X \right)$$

and

- T = temperature, K
- Z = compressibility factor for gas law ($pv=ZRT$)
- R = gas constant, J/(kg K)
- C_p = constant pressure specific heat capacity, J/(kg K)

HS (Mollier) Integration. If HS-(Mollier) integration is chosen the head is calculated by integrating with constant polytropic efficiency from the inlet to the outlet conditions increasing the pressure in small steps and using entropy and enthalpy calculations on the gas in each step.

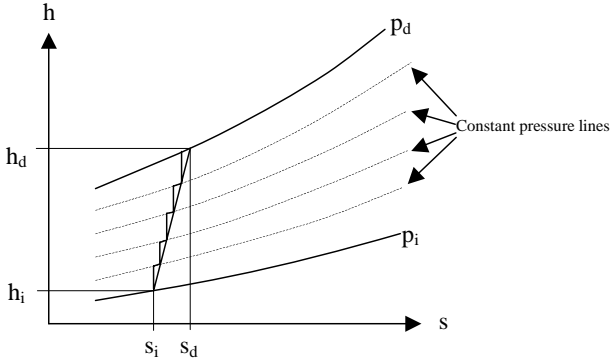


Figure 4. Head calculation using HS integration

The enthalpy increment in each step is calculated by first calculating the isentropic enthalpy increment and then divide by the “step isentropic efficiency”, η_{Δ} . The step isentropic efficiency is calculated based on the polytropic efficiency:

$$\eta_{\Delta} = \frac{\pi_{\Delta}^{\frac{n_s-1}{n_s}} - 1}{\pi_{\Delta}^{n_s \eta_p} - 1} \quad (12)$$

where

- π_{Δ} = pressure ratio over each step
- n_s = isentropic volume exponent

Power Consumption. The power consumption is calculated based on the mass flow and the actual head.

$$P = wH = w \frac{H_p}{\eta_p} \quad (13)$$

where

- P = power consumption, W
- w = mass flow rate, kg/s
- H = actual head, J/kg

The dynamic response of the shaft speed due to compressor power requirement changes is simulated in another module that interacts with the compressor.

Other Relevant D-SPICE Modules

Pressure Node. This module simulates a pipe volume such as a manifold or a header. The module is normally part of the pressure-flow network system. Such systems are built up by configuring flow elements, such as compressors and valves, between volumes and vessels. The model includes both heat and material balances for the fluid, and a heat balance for the surrounding wall. Heat transfer from the

fluid to the wall, and from the wall to the ambient can be configured.

Pipe Flow. This module calculates the flow through sections of pipe work. The flow is calculated based on frictional pressure drop and hydrostatic pressure contributions due to elevation differences.

Heat Exchanger Stream. This module simulates the behaviour of a process stream inside a heat exchanger. Two or more instances of this module can be used to configure ordinary two-stream heat exchangers as well as multi stream heat exchangers.

The module has two main parts: the fluid and the wall surrounding it. The fluid exchanges heat with the wall. In addition, it is possible to connect external surfaces (walls) that exchange heat with the fluid. Furthermore, it is possible to configure direct heat transfer between the local wall and external walls.

Valve. This module uses standard flow correlations with due regard taken for flowing medium conditions. Each valve is simulated using the information available on valve data sheets such as design CV, valve characteristic and actuator type. An empirical correlation is used to simulate choked flow. If available, the valve stem stroke times to open and close the valve are used. When these data are not available, stroke times are estimated based on service and valve size. There are three models available for the dynamic response of the stem: second order, first order and ramp.

Machine Driver. This module simulates an electric motor. It is used as a drive for compressors and pumps. The rotor inertias for all driven machines are treated by this module. Gear ratios between the machines can be specified. The motor power is calculated by using a performance curve of torque versus speed. The synchronous speed can be controlled via input control signal. The relationship between electric voltage and motor torque is modelled by scaling the torque by the square of the voltage fraction.

Snøhvit refrigerant circuits in D-SPICE

The three refrigerant circuits have been modelled in one sub-model in D-SPICE. This model represents the whole refrigerant process including all LNG heat exchangers. Figure 5 shows as an example, the liquefaction section as it is implemented D-SPICE. A suction drum, compressor and after cooler constitute the circuit. An anti-surge controller (SCV) manipulates the recycle downstream the after cooler to the suction drum. A hot gas bypass (SPV) is provided for surge protection.

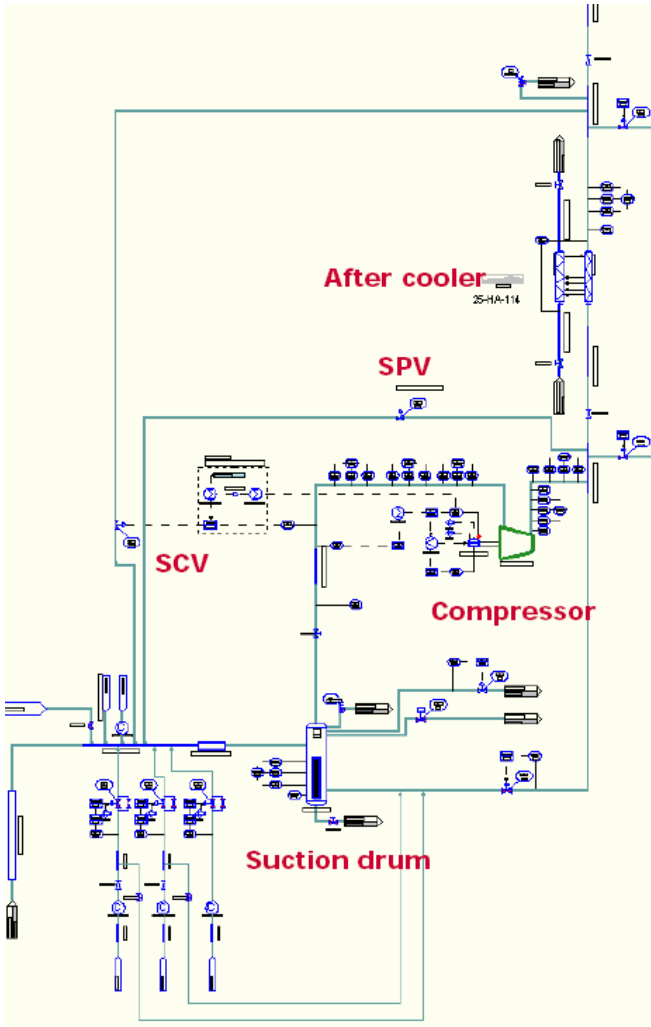


Figure 5. Liquefaction section in D-SPICE

Figure 6 shows the performance map for the liquefaction compressor as it is implemented in D-SPICE. The points are measured data from Factory Acceptance Tests.

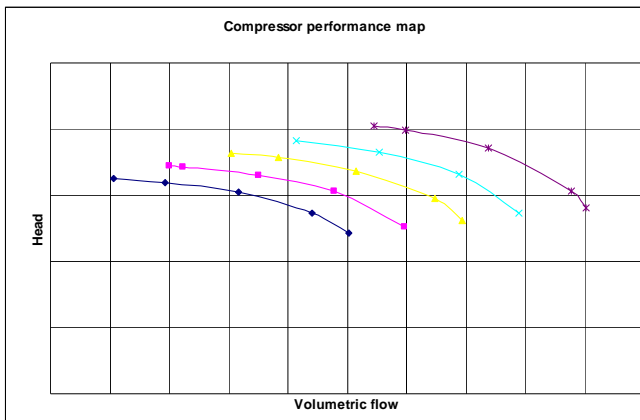


Figure 6. Performance map of liquefaction compressor implemented in D-SPICE

The same curves are repeated in the Fig. 7. Now plant operation data from site tests are included (red dots) and measured surge points (triangles). The match is very good and the discrepancies can be explained by site conditions.

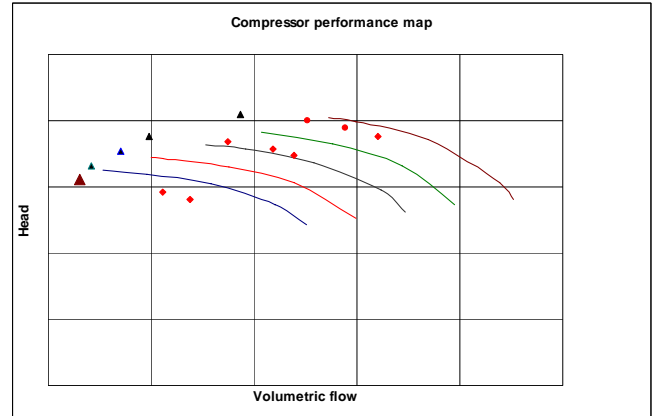


Figure 7. Performance map with points from compressor tests at site

SENSITIVITY ANALYSIS

This section presents some results from the sensitivity analysis on the refrigerant compressors. Simulations were performed to analyse the Snøhvit refrigerant circuits and the ability of the compressor protection system to keep the run-down out of surge after a driver trip. This was done by inducing a driver trip from normal operation and studying the run-down the first four seconds after the trip. This is the most critical phase of the run-down, since after four seconds most of the energy has been removed from the system.

Sensitivity to parameters such as motor moment of inertia and anti-surge valve characteristics was studied. Once the data from the factory acceptance tests (FAT) of anti-surge valves were available, simulations were also performed using these values.

Time delay and dead time on the opening of the anti-surge valves are essential parameters and the importance is shown in the simulations.

Moment of inertias

The compressor run-down was simulated using different moments of inertia. Figure 8 shows the run-down with a fifty per cent reduction in the motor moment. Simulations show that the compressor runs down in the surge zone. Figure 9 shows the correct run-down using the total moment of inertia given in the data sheet for the motor. This illustrates the importance of implementing correct values in the simulation model.

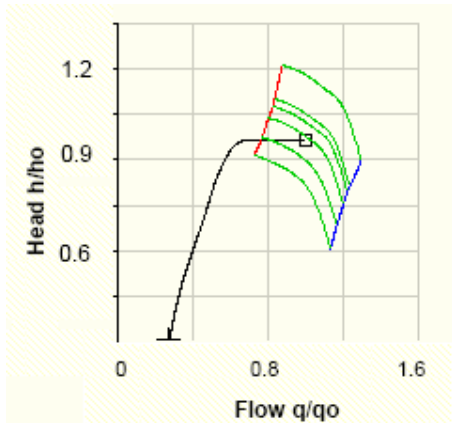


Figure 8. Low moment of inertia on compressor motor

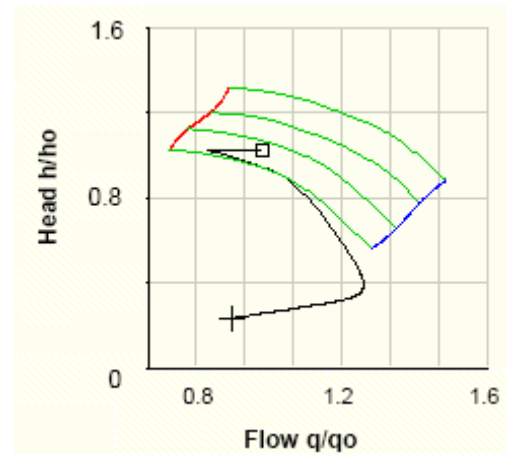


Figure 11. Compressor run-down using longer opening time of the anti-surge valves

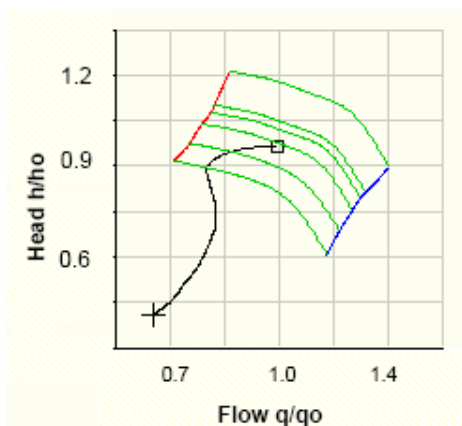


Figure 9. High moment of inertia on compressor motor

Anti-surge valve characteristics

The sensitivity to anti-surge valve opening times and dead time was analysed. The results are presented in the following.

Opening times of the anti-surge valves. Figure 10 shows a compressor run-down using opening time of 1.25 s based on valve factory acceptance tests. Figure 11 shows the run down using a guessed value of 2.3 s for comparison.

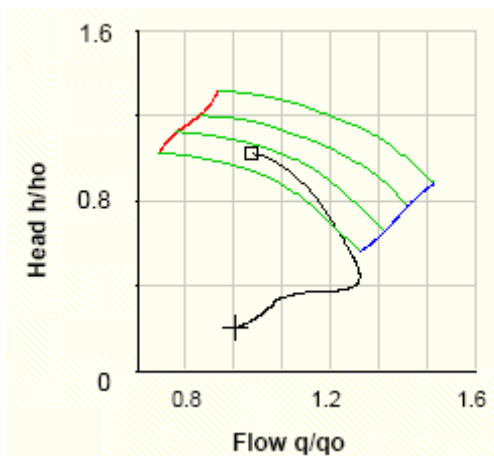


Figure 10. Compressor run-down using FAT opening times of the anti-surge valves

Dead time. Figure 12 show the compressor run-down using a dead time on the opening of the anti-surge control valves of 200 ms. Figure 13 shows the run-down with no dead time on the opening of the anti-surge valves.

The simulations results show that opening time and dead-time of the anti-surge valves have a significant effect on the run-down. It is important to use valve data from factory acceptance test when simulating and evaluating the robustness of the compressor system. Sensitivities to these values should also be simulated since valve opening time and dead time may increase due to lifetime wear.

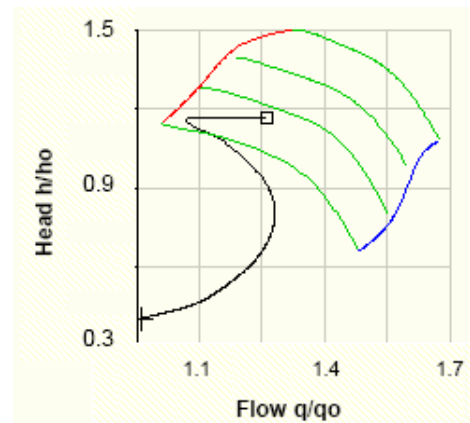


Figure 12. Dead time equal to 200 ms

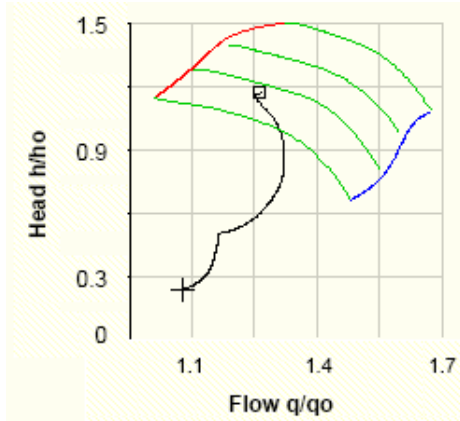


Figure 13. No dead time

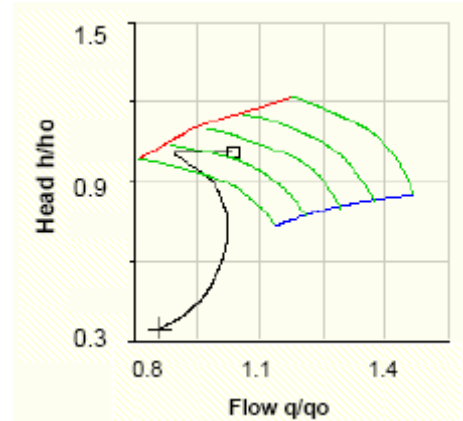


Figure 15. Compressor run-down, time step equal to 0.01 s

Moreover, the instrumented safety system must be designed so that time delay and dead time are as small as possible.

Time step

The sensitivities of the discretization time used in the dynamic simulations are studied in this section. Simulations were performed using varying time steps. Figure 14 shows run-down using a time step equal to 0.1s. The transient is not well documented in the simulation. Figure 15 shows the run-down using a time step equal to 0.01 s. The transient is well documented. Decreasing the time step did not give additional information, see Fig. 16. Nothing significant was obtained by reducing the time step further. Knowledge on this issue saves time and contributes to a more efficient analysis.

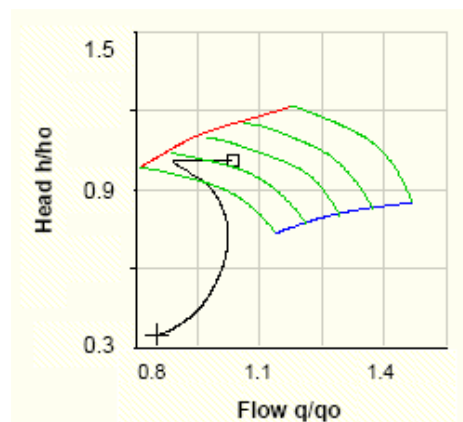


Figure 16. Compressor run-down, time step equal to 0.001 s

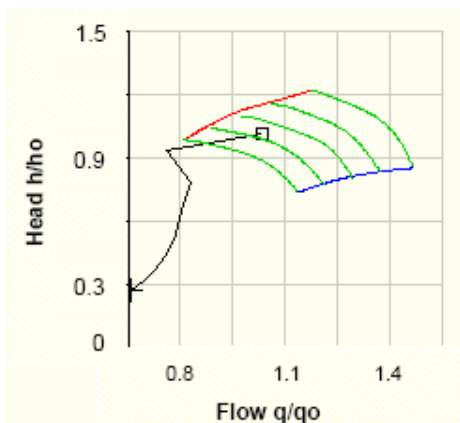


Figure 14. Compressor run-down, time step equal to 0.1 s

CONCLUSIONS

This paper gives an introduction to the Snøhvit LNG plant and the dynamic process model applied in the engineering phase for design and verification purposes. The total plant model has been utilized in the analysis of the Snøhvit refrigerant compressors. The advantage of a total process model is that in case of one compressor trip the effect on other process components can be studied.

The compressor models are based on the procedures described in ASME PTC 10. These procedures are designed for testing compressors, but they are also well suited as a basis for a simulation model. The model has been verified against plant operation data from site tests, measured at surge points. The match is very good and the discrepancies can be explained by site conditions.

A sensitivity analysis was carried out to evaluate the ability of the compressor protection system to keep the run-down out of surge after trip. One general conclusion is that the instrumented safety system must be designed so that time delay and dead time are as small as possible. Moreover, this work shows the value of dynamic simulations for the verification of compressor protection systems. The importance of using correct data for the polar inertia of the rotating equipment is demonstrated, as well as the opening and dead time of the anti-surge valves. It is recommended to

include a sensitivity analysis of these parameters as part the of general plant verification studies.

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