Statoil’s 4.3 million tpy Snøhvit LNG production plant, near the town of Hammerfest in Norway, is the first of its kind installed in Europe and the world’s northernmost LNG plant (Figure 1). The heart of the process plant, weighing some 33,000 t, excluding the cryogenic cold box but including the barge substructure, was prefabricated in the port of Cadiz in southern Spain, approximately 5000 km from where it had to be installed. Harsh arctic weather conditions and lack of local infrastructure at the installation site determined this solution.

Space limitation constraints had therefore high priority in the concept and layout of the process plant, which utilises electric motor driven refrigerant compressor trains and is based...
The remote location and the choice of electric motor driven compressor trains required a self sufficient power generation facility, which was provided by five LM6000 gas turbine generators supplied by GE Oil & Gas. These generators can operate in island mode or be connected to the land electric network providing a maximum import of 50 MW. The main electrical consumers on the plant are the three refrigerant compressors supplied by GE Oil & Gas, these being the:

- Precooling cycle, a 65 MW side stream compressor train
- Liquefaction cycle, a 32 MW inline compressor train
- Subcooling cycle, a 65 MW tandem casings compressor train (Table 1).

These rated power figures are a record themselves, the variable speed motors being the most powerful ever built in the world until this project started.

The concurrent cost, efficiency and productivity targets pushed towards a ‘single equipment design strategy’, particularly as regards the core process, consisting of the three mentioned cascade refrigeration cycles. The behaviour and the dynamic interaction of each of these compressor trains within the process plant hardware (pipework, heat exchangers, surge protection and control valves) and control systems (sequencing, control and safety), during all normal and abnormal operating scenarios, required careful analysis to match schedule and challenging targets in terms of safety, environmental compatibility, lifetime costs and time to market.

Statoil and Linde recognised early on in the project development the need to understand the process dynamics in their every facet, particularly as regards the new, electrically driven, rotating machinery. They also recognised that conventional ‘dynamic’ process simulators could not provide the insights to some very critical, fast changing phenomena related to mechanical dynamics, fluid flow dynamics and process thermodynamics which have mutual influence on compressor train behaviour, particularly during short transient system changes. The compact plant design and consequent low piping volumes also exacerbates the speed and depth of transient values. The overall situation required that the simulation contractor had a suitable simulation tool and full access and control of the software kernel code, in order to provide adequate modifications to the models, as long as new fidelity requirements were added during project evolution. Therefore Statoil, Linde and Compression Machinery Consultants Ltd selected Systems & Advanced Technologies Engineering S.r.l. (SATE) for the performance of the dynamic simulation study (DSS) and from then on GE Oil & Gas took the leading role in the execution of the DSS. SATE utilised its proprietary modelling environment COMPSYS™, which implements new solutions, advancing the state of the art in compression systems simulation, as summarised in this two part article.

The DSS of the refrigeration compressors started in November 2002, during the plant detailed design phase as an ongoing plant development tool, and proceeded by various targeted phases, ending with the final check of the startup and operational logic sequences of the compressors and with the preparation of the ‘road map’ for the commissioning activities.

**Why DSS was necessary and what it provided**

Early investigation of compressor surge upon ESD

Phase 1 of the DSS focused particularly on the basic design of the piping and compressor surge prevention and protection system under emergency shutdowns (ESD) caused by unplanned electrical outages. The purpose of this focus was to confirm or suggest modifications for pipe routing and location of non-return and control valves. Various options of antisurge line arrangements, valve size, opening laws and dead time were simulated. These simulations all led to the conclusion that a general increase of previously defined valve size was necessary. Phase 1 also made it apparent that hot gas bypass lines, previously avoided to attempt to simplify the control system, were now essential to minimise the events from which compressor surge could occur.

**Definition of recycle system**

Phase 2 DSS refined and extended the conclusions of Phase 1, including the definition and completing the implementation into the model of the surge control and surge protection controller functionality, to allow dynamic simulations of operational conditions under antisurge recycle and performance feedback control. The analysis enabled early development of startup sequences, and evaluated various ESD and voltage dip scenarios to assess sensitivity on the surge protection valves (SPV) and surge control valves (SCV) timing, their flow coefficient characteristic (opening laws) and the non return valves (NRV) closing characteristics. The DSS revealed that high $p\sqrt{v}$ peak values could occur in equipment such as suction drums and aftercoolers. This phase consequently enabled rearrangement and finalisation of the piping design and routing, releasing the specification of material and major piping items including surge control and protection valve sizes and non-return valve size.

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**Table 1. Main data of the LNG refrigeration compressors**

<table>
<thead>
<tr>
<th>Main compressors data</th>
<th>Precooling train</th>
<th>Liquefaction train</th>
<th>Subcooling train</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet volume flow (m³/s)</td>
<td>36</td>
<td>42</td>
<td>42</td>
</tr>
<tr>
<td>Inlet nozzle diameter (mm)</td>
<td>1400</td>
<td>1400</td>
<td>1400</td>
</tr>
<tr>
<td>Number of casings</td>
<td>1</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>Total number of impellers</td>
<td>4</td>
<td>6</td>
<td>13</td>
</tr>
<tr>
<td>Total polytropic head (kJ/kg)</td>
<td>135</td>
<td>170</td>
<td>320</td>
</tr>
<tr>
<td>Driver rating (MW)</td>
<td>65</td>
<td>32</td>
<td>65</td>
</tr>
<tr>
<td>Normal operating speed</td>
<td>3300 - 3800 rpm</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 1. Melkoya island, site of the Snøhvit LNG project (photo courtesy Eiliv Leren/Statoil).
Process scenario investigations and HAZOP cases

Having established the basic piping system design and component sizing in Phase 2, the Phase 3 conducted a broader review of operating scenarios suggested by process and HAZOP reviews. As a non-exhaustive example, this new runs programme had the following objectives:

- Redefine and verify the size of the SPV and SCV and the relevant piping under a broader set of ESD conditions; power loss from both the normal operating point and from the maximum continuous speed and the discharge pressure reached on saturated cold startup.
- Verify the startup capability of the trains and the driver torque margin on both warm and cold initial conditions.
- Confirm the shaft speed, power, discharge temperature and suction pressure at which the rated discharge pressure could be achieved with the compressors in full recycle on the surge control line (SCL).
- Understand the highest allowable restart pressure that could be tolerated to produce the rated discharge pressure when operating on the surge control line, whilst not exceeding the maximum shaft torque, the maximum discharge temperature, or the maximum continuous speed (MCS). Establish the suction pressure pull down at this operating point.
- Confirm that the starting inlet temperatures assumed for the cases considered are consistent with process and ESD settle out values.
- Ensure the load balancing between the two stages of the precooler and the subcooling compressors during startup, which foresee optional long speed holding at the borders of the operating speed range, particularly in the available intervals between the torsional and lateral critical speeds.

The above programme did not merely allow the final verification of the lines size, layout and components, but, even more important, provided a better view of the safe operational envelope of the compressors under normal and offdesign conditions, and during startup sequences.

Dynamic compressors maps and realism enhancement

Phase 4 was a further step forward in that it provided the verification of the compressors’ behaviour with various gas mixtures, identified by process development for commissioning activities, once again at different startup pressures and temperatures. During this phase it was essential to exploit one of the newest features of the COMPSYS™ simulator, i.e. the possibility to have the compressor map change during the same simulation run, in response to greatly deviating suction conditions from those upon which the basic performance maps are provided (Figures 2 and 3).

This simulator feature provides unprecedented fidelity in the simulation of transients characterised by large thermodynamic variations at the compressor inlet. This occurs, for example, with LNG refrigeration cycles, when a compressor is switched from full recycle to cold flowing conditions or from the cold flowing to the warm recycle conditions, e.g. on surge control intervention or ESD.

To obtain high fidelity the estimated compressor maps are still used as an input to the model, but they are referred to a specific suction condition (gas composition, pressure and temperature). COMPSYS™ then calculates the speed of sound in the gas at the inlet by means of accurate expressions for real gases. This speed of sound is then used to correct the compressor performance map in a manner determined jointly with GE Oil & Gas.

An enhanced model of the surge control system, both for the antisurge and performance control functionalities, was integrated into the models. This involved an additional flow measurement in the startup valve (SUV) line for the subcooling compressor, as well as a more sophisticated speed control for the two process stages of the precooling compressor, featuring the capability for smoothly switching the speed control between the medium pressure inlet stream, the low pressure inlet stream, or the discharge pressure.

FAT maps and operations flowchart testing

Phase 5 commenced upon completion of the compressors full load factory acceptance tests (FAT) when GE Oil & Gas issued the ‘as tested’ compressor maps. These maps were implemented into the model, thus approaching further the expected real plant behaviour. The scope of Phase 5 was at this stage focused to finalise and check...
the operating procedures and verify thresholds and control parameters to be set, at least as default values, for the real startup. Moreover the following detailed checks were defined:

- Detailed check of the previously established process startup sequences, loading, unloading and normal shut down transients.
- Finalise the speed ramp values for different sections of the start sequence in conjunction with restrictions on electrical power loading rates and avoidance of inter-harmonic disturbances in the plant electrical network.
- Retune the control system parameters, to avoid previously observed oscillations after transients, particularly at the end of the startup sequence.
- Verify the possibility of increasing the opening time of the SCVs and of the SUV (for the subcooling compressor) on ESD events, to limit the peak flow through the intercooler, aftercooler and suction drums.
- Check the effects and evaluate possible counteractions against supply voltage dips, at various dip durations.

The results of this phase gave useful indications about the definition of compressor startup and operations logic, e.g. a new SCVs and SUV management law during the trains acceleration. Furthermore, some potential dynamic instability to be managed by proper controller tuning was identified. Some of these arose simply as a consequence of gas mixture changes.

One of the most challenging goals set forth for the procedures definition was that a unique set of control and threshold parameters had to be compatible with all possible gas conditions during operations. Despite the compensations provided by some of the antisurge control algorithms, it must be remembered that they are based on physics approximations due to the impossibility of measuring some of the gas characteristics online (e.g. the gas compressibility, the specific heat ratios and the molar mass). The in-field tuning of the various controller parameters is an important and costly activity, performed to match the actual behaviour of the compressors, by empirically compensating for such inaccuracies and unknowns.

The Snøhvit project required a more straightforward and definitive in-field tuning plan compared to what would normally be developed, in order to manage a tight schedule and to be prepared for the expected dynamic response of the system on the first startup. Last but not least, the requirement to have the widest possible operating range at each speed, which means having confidence in stable and surge free operations even close to the map limits.

Shortening the path towards a correct controls procedure and parameters settings for full recycle and other operational scenarios, again with diverse gas mixtures, was therefore a fundamental goal for this DSS phase that was fully achieved. As a result of the accurate thermodynamics and gas dynamics of the COMPSYS™ model, the effects of the approximations of the inline algorithms used by the controllers could be detected, explained and highlighted in relation to the realistic model of the plant. In particular Linde could refine the gas mixture and mass inventory definition for the startup, based on the acceptable pressure (SUP) and temperature (SUT) identified by DSS.

A road map to commissioning activities and tests

The final DSS phase, startup simulation (SUS), addressed further operational details, specific to the commissioning tests of the compressors, in order to:

- Develop a strategy and a flow coefficient (Cv) characterisation for startup trims in surge control valves during compressor operation for initial process circuit cleanliness checking and follow-on operations using mixed refrigerant gas having a range of compositions.
- Establish a baseline for the operating conditions at the compressors and in the process system when operating with temporary strainers at the compressor inlets. The objective here being to utilise the lowest possible system pressure/gas inventory, and not to create too low a piping temperature, so reducing the time for strainer inspection and/or cleaning.
- Investigate how to maximise the compressor recycle system gas density and gas velocities to prove the system cleanliness prior to replacing the fine mesh in the strainers. The objective here being to utilise the lowest possible system pressure/gas inventory, and not to create too low a piping temperature, so reducing the time for strainer inspection and/or cleaning.
- Establish a surge control line that will satisfy operation on the rated gas composition and a lean gas composition which could be experienced during startups and
restarts following a depressurised compressor shut-down.

- Establish a compressor operating point in the lower speed range at low pressure ratio for warming up the recycle system prior to intrusive maintenance activities.
- Observe the dynamic response of the compressor and system to ESD during cleanliness checking activities and operation with compressor inlet strainers in place.
- Define a clear ‘road map’ of operational test paths and spot points for all the three compressors as a guide during the real startup, with the expected values of the main process signals.
- Define and anticipate any compressor surge scenarios during commissioning trip tests.

Prior to making the new runs SATE made an accurate calculation of the pressure loss coefficients of the temporary strainers, with a fine mesh on a perforated screen, both clean and fouled, in order to better evaluate the suction pressure changes during fast compressor accelerations and surge, and to establish safe conditions and avoid a potential mesh basket collapse.

The use of the compressors dynamic simulator was most convenient in the preparation of the above mentioned ‘road map’ of startup spot points. Defining them by ‘hand’ or even an iterative computer aided sequence of ‘static’ calculations would have been very demanding or would have come at the expense of accuracy. This is because flow rates through the recycle valves, pressures and temperatures in the various volumes, compressors maps, and consequently their own flow rates, are strongly interdependent and dynamic. Moreover, the instantaneous compressor speed is significantly different from the corrected one, increasing the difficulty. Instead, running a relatively small set of simulation runs allowed a fast and consistent collection of check point data.

The first results obtained in this final DSS phase were the basis for the issue of a revised engineering manual by the compressor vendor (GE Oil & Gas) in cooperation with the antisurge and performance controller supplier (CCC). The DSS results produced target log tables, and operating point paths plot overlapped onto the compressor maps for the following scenarios:

- Startup steps and valves manual override tests with two main gas mixtures for each compressor.
- Compressor maps confirmation, since FATs could not be performed with the real process gas (Figure 4).
- Trip test scenarios, to be safely performed to check the actual capability of SPVs and SCVs.

The DSS results were finally implemented into the Dynamic Commissioning Procedure (DCP), issued by Linde and Compression Machinery Consultants Ltd (UK). This DCP was used as the lead document, defining the recommended activities necessary for the efficient and safe commissioning of the compressor systems.

**Main features of the simulation models**

Three dynamic models for the precooling, liquefaction and subcooling compressors were implemented during the DSS described above, providing upgrades at each phase when needed to keep pace with the new information being available and the evolving, progressively more demanding, DSS requirements.

These models included the thermal and flow dynamics of heat exchangers, condensate separation drums, strainers and valves under the action of the various control system loops, particularly the antisurge and performance controllers. The main features of the modelling approach have been described in earlier articles. These articles provide an insight into criteria recommended for some specific analyses, such as the advanced methods for sizing SCVs and SPVs. Part two

![Figure 6](image6.png)  
**Figure 6.** Comparison between the pressure ratios obtained in the startup tests (solid lines) and the simulation values (dotted) at the guaranteed point (95% of maximum continuous speed (MCS)) and at the MCS. Pressure ratio versus volume flow rate, both normalised to rated values.

![Figure 7](image7.png)  
**Figure 7.** Comparison between the suction pressures obtained in the startup tests (solid lines) and the simulation values (dotted) at the guaranteed point (95% of MCS) and at the MCS. Pressure versus mass flow rate (mfr), both normalised to rated values.

![Figure 8](image8.png)  
**Figure 8.** Comparison between the suction temperature obtained in the startup tests (solid lines) and the simulation values (dotted) at the guaranteed point (95% of MCS) and at the MCS. mfr normalised to rated value.
of this article will highlight the innovations exploited for the Snøhvit project.

**Comparison with actual compressor startup data**

At the time of writing this article the startup of some of the Snøhvit compressors is ongoing. However a first indication of the accuracy of the model used for this DSS comes from the data recorded for the liquefaction compressor circuit (Figure 5) in full recycle mode (part two of this article will discuss the comparison results also for the two other compressors). The results of the comparison are remarkably good, particularly considering that the compressor FAT maps were derived from tests made with gas mixture components greatly different from those used in the commissioning test. In Figure 4 the mapped target points and the actual ones are plotted in an absolute coordinates plane together with the main and the extreme maps obtained with the highest and lowest expected temperatures. The real compressor spot points, represented by squares, lay slightly below the target pattern, represented by the cross points corresponding to the SUS simulations, particularly in the region close to the surge limit. The discrepancy however is negligible considering the concurrent perfect match obtained for the pressure ratio versus volume flow rate curves (Figure 6).

The suction pressures obtained are well overlapped by the simulations, with differences smaller than 5% for the minimum and 1% for the maximum recycled mass flow rates in the range explored (Figure 7). The differences in suction temperatures are in the range of 2 - 4 °C, following a remarkably matched trend as function of flow rate (Figure 8).

It is important to note that while the extensive FAT was performed in compliance with the ASME PCT10 code with respect to the rated process mixture and inlet conditions, the commissioning process conditions were not part of the FAT scope. The main purpose of compressor mapping during the site’s first startup was to verify and compare the results gained from the dynamic simulation studies.

The observed slight deviations can be due to several factors:

- The gas mixture composition slightly differed from those simulated for the test preparation.
- The measuring tolerances related to the installed instrumentation are larger compared with the observed discrepancy.
- While the initial pressure before the plant startup in full recycle was the same assumed in the simulations, the actual startup temperature was approximately 5 °C lower (10 °C instead of 15 °C, for which a revised simulation was not run).

**Conclusion**

Compressor DSS covered the full project life of the Snøhvit LNG plant, from basic design, through to detail engineering and final testing. They played an important role during the design and startup preparation and execution of the refrigeration compressors, particularly with regards to the verification of the interactive dynamic behaviour of these large compressors and variable speed electric drivers with the dry gas part of the plant process.

One of the most important technical aspects that arose during the simulation of the Snøhvit LNG refrigeration compressors is that the train and the short term process dynamics (at milliseconds to seconds scale) with electric motors is fundamentally different from that of gas turbine driven compressors, due to the much faster torque rise and decay transients (e.g. on ESD, power loss and voltage dips) and the interaction with the valves opening/closing transients and flow inertia phenomena.

The simulation results matched the real measurement values within the accuracy allowed by the instrumentation and within the plant engineering and operation requirements, as already proven by the initial startup tests in full recycle. The deviations observed can be fully justified by physical condition differences.

From the plant control perspective the DSS results proved very effective in:

- Securing a confident plant piping and machinery layout in step with project schedules.
- Development of compressor train start sequences and addressing complex issues.
- Understanding, gaining confidence and optimising a smooth system behaviour on startup.
- Checking and refining the operational procedures and their parameters.
- Identifying suitable points for the commissioning tests.
- Saving time for on-site tuning of controllers.
- Saving time in starting the machines and making trip tests.

One key factor for the successful application of DSS to this project was the close and effective cooperation among the various parties involved (end client, process and rotating machinery experts, compressor vendor and simulation contractor teams) during all the DSS phases and the full control and flexibility of the COMPSYS™ modelling environment and software made available by the simulation contractor. This allowed the full utilisation of the increasing information available during the project, keeping pace with the evolving design requirements and questions arising during the plant engineering and machinery procurement.

The use of purely commercial software simulators, as often proposed in dynamic simulation services, could not allow a similar ongoing model and scope development and in-depth analysis of the phenomena studied herein, as discussed in-depth in other previous articles.8

**References**


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