APPLICATION OF DYNAMIC SIMULATION IN THE DESIGN, OPERATION, AND TROUBLESHOOTING OF COMPRESSOR SYSTEMS

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ABSTRACT

Dynamic simulation with rigorous mathematical models allows for evaluation of compressor performance under nonsteady-state conditions. It has become a powerful tool in the design of compressor antisurge protection, evaluation of operation envelope, driver size selection, field support, and troubleshooting. Recent developments in the simulator technology make it possible to develop large-scale dynamic simulation models and use them in the entire project life-cycle.

This paper discusses the basic scope of dynamic simulation, typical application areas, and several case studies. The case studies are based on a wide range of applications, including conceptual feasibility study, detailed engineering, equipment specification, precommissioning, and field support. These case studies cover different compressor lineups, such as axial and centrifugal compressors and parallel operations. One example shows how a dynamic model was used to identify critical design issues at the early stage of engineering phase to avoid design or field problems at a later date. Other examples show how dynamic simulation was used to evaluate design alternatives and ultimately improve overall compressor operation and protection.

INTRODUCTION

The specification of a compressor system during the design phase is often based on a set of predicted operating points from steady-state heat and material balance. However, a compressor system can experience various transient operations when it is in service, including startup, shutdown, turndown and equipment failure. These operations are dynamic in nature, so a steady-state model may not be adequate to address issues associated with transient operations, such as startup capability, compressor surge protection, and stability of operation during turndown. In the past these issues were addressed by experience. As the industry progresses and plant scale continues to grow, there are compelling needs for dynamic simulation to provide a more accurate and consistent analysis to address these issues. This is particularly important for the specification of recycle loop configuration for compressor protection because underdesign can lead to inadequate flow (surge) while overdesign can lead to excess flow (stonewall).

Dynamic simulation is a tool to simulate and study process phenomena and equipment behavior that are typically characterized
by dynamics rather than steady-state. It is a critical link between design and safe operation. When applied to a compressor system, dynamic simulation allows for a detailed understanding of the actual operating characteristics of the compressor over the full range of operation. Another area of interest is plant expansion where dynamic simulation can be used to identify bottlenecks in the compressor system by simulating the performance of the existing compressors after plant throughput changes.

Before the 90s, many dynamic models were programmed in FORTRAN, which required considerable maintenance. There were also limitations on the numerical methods available and the ability to link the model to a thermophysical database. Therefore, the models were generally limited to flowsheets with a relatively small number of equipment units. In addition, because the model components were developed in-house and generally considered to be proprietary, the use of dynamic simulation in the industry was rather infrequent at that time. In the early 90s, simulators such as SPEEDUP and OTISS became readily available, which provided a platform to solve the basic dynamic equations and build unit operation blocks in a more efficient, consistent way. Using these simulators, case studies were done for complex units (Cassata, et al., 1993). Since then, it has become a common practice in the industry to use dynamic simulation to analyze large-scale, complex systems where multiple compressors are closely integrated with other processing units. Examples are liquefied natural gas (LNG) refrigeration compressors and offshore gas reinjection trains. Recent advancements in simulator technology provide the users with off-the-shelf model components for common process equipment, rigorous thermophysical database, ability to transfer interfaces. All these advancements have contributed to the wider use of dynamic simulation in recent years and dynamic simulation becoming integral to the engineering work process (Omori, et al., 2001), field support, and other stages of the plant life cycle (Valappil, et al., 2005). Table 1 lists some of the previous applications of dynamic simulation in the chemical process industry.

**Table 1. Some of the Applications of Dynamic Simulation in the Industry.**

<table>
<thead>
<tr>
<th>Facility</th>
<th>Application</th>
</tr>
</thead>
</table>
| Refining | • Reactor depressurization  
• Operation and control of distillation columns  
• Utility – steam and fuel gas: control system study and design verification  
• Relief and flare load calculation |
| LNG | • Refrigeration compressor system  
• Liquefied natural gas compressor: design verification and evaluation of operating procedure to minimize flaring  
• Power generation and electrical system: stability during transient operations in the plant  
• Utility – steam, fuel gas, heating and cooling medium: control system study and design verification  
• Relief and flare load calculation |
| Ammonia | • Reactor control system study  
• Utility – steam, fuel gas and cooling water: control system study and design verifications  
• Relief and flare load calculation |
| Offshore | • Topside compressor systems for stable parallel operation  
• Design optimization of topside piping and flare load calculation  
• Flowline analysis (flow assurance): to evaluate slugging and separator size  
• Utility – power generation, heating and cooling medium: control system study |
| Ethylene | • Process gas compressor: surge protection and stable operation  
• Ethyleneacker: control scheme study and evaluation of operating procedure  
• Utility – steam, fuel gas and cooling water: control system study and design verification  
• Relief and flare load calculation |

**BASIC EQUATIONS**

The core of dynamic simulation is a mathematical model that simulates the detailed transient behavior of equipment based on the heat and material balance and the performance characteristics of each equipment (Botros, et al., 1991). There are important distinctions between steady-state and dynamic modeling, as summarized below:

- In a steady-state model, the pressure and flow in the flowsheet are estimated based on steady-state and material balance. Therefore, in steady-state, the total mass inflow to a unit will be equal to the total mass outflow from the unit.
- In a dynamic model, the pressure of a unit is estimated based on accumulation. The flow between units is estimated based on pressure-flow relationship. For example, in dynamics, the total outflow from a drum can be smaller than the total inflow, in which case the pressure of the drum starts to rise. This occurs during a compressor trip when the discharge flow is recycled back to the suction and pressurizes the suction drum.

**APPLICATION OF DYNAMIC SIMULATION IN COMPRESSOR PROTECTION**

For a compressor, the operating characteristics are often shown in the form of a map of operating envelopes. Figure 1 is a typical compressor map with defined operating boundaries. The compressor manufacturer recommends a minimum flow, i.e., surge limit line (SLL) on the operating map, and generally performs a surge test at the supplier shop and in some instances retest at site to verify the surge limit line. If the compressor suction flow falls under the SLL, surge can occur, which is characterized by fast flow reversals that can be detrimental to compressor components. Therefore, surge prevention is one of the key aspects of compressor protection (White, 1972). Providing a recycle valve is a common measure to maintain adequate flow, and there are general requirements on the recycle valves in terms of capacity and stroke rate (Wilson and Sheldon, 2006). For large-scale compression systems, it is now imperative to use dynamic simulation to size the recycle valves based on various dynamic scenarios, such as startup, feed variation, shutdown, and shutdown, to ensure that sizing criteria cover the most severe operating condition rather than by experience.

![Compressor Head vs Flow Map](Figure 1. Typical Compressor Operating Map. For Safe Operation, the Operating Points Should Be Between the Surge and Stonewall Lines.)

In addition to the surge limit line, the compressor manufacturer often recommends a high-flow limit on the right-hand side of the operating map, as shown in the previous Figure 1. Excessive flow across the compressor, referred to as stonewall, can cause vibration and fatigue failure that can damage the entire compressor over
time. Again, dynamic simulation can be used to predict the operating point trajectory under various transient scenarios and evaluate measures that can reduce the risk of stonewall. For long-term operations, the most effective means of avoiding stonewall is by fine-tuning the operating conditions to reduce the excess inflow to the compressor. For short-term operations, for instance after fail-open of a recycle valve, high-flow alarm and trip are often used as protection against stonewall.

Another important application of dynamic simulation is to determine the design parameters of a compressor for stable and robust operation. One of the key parameters is head-rise-to-surge. Notice that the compressor performance curves in the previous Figure 1 become nearly flat as the SLL is approached. If the compressor curve is flat, i.e., a small head-rise-to-surge, even a relatively small disturbance can drive the compressor quickly toward surge when the operating points are close to the SLL. Dynamic simulation can be used to analyze the sensitivity of compressor behavior with respect to the shape of operating curves and determine the margin required between the operating point and the SLL. This is discussed in more detail in one of the case studies in this paper. Generally speaking, for a large-scale centrifugal compressor that is used in the LNG refrigeration system, a 6 to 10 percent head-rise-to-surge is needed for robust operation, which is also achievable based on previous site testing results. Note that if the compressor has multiple design guarantee points, it is possible to have a smaller head-rise-to-surge for some of the design points, in which case the control system must be more robust to provide adequate protection.

SIMULATION MODEL

Figure 2 shows a typical single-stage compressor line-up that includes a suction drum, an aftercooler, and a set of recycle valves. Figure 3 shows a screenshot of dynamic model for the same flowsheet. The simulation flowsheet, graph (showing compressor speed in this example), and model components can be seen from the user interface.

Table 2 lists the key input data for a dynamic model of the compression system. Notice that the list includes components such as pipe segments, location of check valves, and instrument lags in measuring the process conditions. These components may not be important in a steady-state model since they do not affect the overall heat and material balance, but are critical to the dynamic behavior of the overall process.

Table 2. Key Input Data of a Compressor Dynamic Simulation Model.

<table>
<thead>
<tr>
<th>Item</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor curves</td>
<td>Head and efficiency vs inlet volumetric flow for each type of gas over the entire speed range, with the surge and stonewall limit identified on the curves</td>
</tr>
<tr>
<td>Driver</td>
<td>Inertia and startup curves (torque vs speed)</td>
</tr>
<tr>
<td>Exchanger</td>
<td>Cooling or heating medium conditions (e.g., ambient temperature for air coolers), heat transfer coefficients and heat transfer area</td>
</tr>
<tr>
<td>Pipe segment</td>
<td>Dimension and equivalent length, and location of instrument taps</td>
</tr>
<tr>
<td>Vessel</td>
<td>Dimension, and location of nozzles and instrument taps</td>
</tr>
<tr>
<td>Control valve</td>
<td>Capacity (Cv), characteristics, stroke rate and delay in the actuator response</td>
</tr>
<tr>
<td>Check valve</td>
<td>Size, location and type (e.g., slam or non-slam)</td>
</tr>
<tr>
<td>Controller</td>
<td>Set point, and control algorithm (e.g., PID)</td>
</tr>
<tr>
<td>Transmitter</td>
<td>Range, sampling interval and lag</td>
</tr>
</tbody>
</table>

Once it is developed, the dynamic model should be validated before it is used for analysis. There are two aspects to validation: steady-state and transient. General guidelines on the model acceptance criteria are provided in ANSI/ISA-S77.20 (1993). For steady-state, the model prediction should be within 2 percent of the measuring instrument range for critical parameters such as flow and pressure, and for transient operations, the overall system transient characteristics’ time should be within 20 percent of the plant responses. Industrial experience shows that steady-state design data can be very closely matched with accurate dynamic models (Dukle and Narayanan, 2003). In general, steady-state validation is a requirement for all the dynamic simulation studies to verify the model input data and the starting points of dynamic analysis.

Validation under transient conditions is less frequent since it generally requires considerable efforts in gathering data from the plant; however the authors’ experience shows that accurate dynamic models can also match good, reconciled plant data very closely. For example, in a recent study, a dynamic model was developed for a three-stage propane compressor that has been in operation for several years (Figure 4). A startup case was performed with the model, and the results closely matched the plant data that were recorded at five-second intervals (Figure 5). In this example, the facility was to be expanded, and the dynamic model was used for analyzing the proposed design modifications of the compressor circuit. Validating these models under the steady-state and dynamic conditions has provided further confidence on the validity of the model results.

Figure 2. Equipment Units in a Compressor System. The Compressor Had a Recycle Valve CV101 to Maintain Adequate Flow During Normal Operation and a Hot Gas Bypass Valve XV101 for Additional Protection Against Surge on Coastdown.

Figure 3. Snapshot of a Compressor Simulation Model and the User Interface.

Figure 4. The Startup Circuit of a Three-Stage Compressor. The Compressor Was Started up in Total Recycle with All Three Valves Fully Open.
STUDY APPROACH

Case Definition and Input Data

The simulation cases to be performed for a specific project depend on the project development stage, available input data, and project-specific issues. Generally speaking, for a conceptual study at the early phase of a project, dynamic simulation is used to address issues such as driver size and selection (Hori and Konishi, 2004) and overall process configuration (e.g., single versus parallel trains). As the project progresses to the engineering phase, additional cases are required to size recycle valves and recycle coolers, determine startup power requirements, and evaluate robustness of the control system. Finally, for field support and troubleshooting, the cases are defined based on field events, often to evaluate potential system modifications before they are implemented at site and/or diagnose specific plant problems.

Although the basic layout of the simulation flowsheet may remain the same, the input data should be updated when more definitive design and operating information becomes available as the project progresses. Table 3 provides a summary of the input data and typical simulation cases for different stages of the project.

Table 3. Compressor Simulation at Different Stages of Project Life-Cycle.

<table>
<thead>
<tr>
<th>Project stage</th>
<th>Available input data</th>
<th>Typical simulation cases</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conceptual study</td>
<td>Estimated compressor curves, preliminary equipment layout plan, and design data from similar plants</td>
<td>Variation of simulation conditions, full-pressure data, shutdown, and startup of one of the parallel trains</td>
</tr>
<tr>
<td>Engineering</td>
<td>Estimated compressor curves based on proven operation, piping isometric drawing, as-designed equipment data sheet, input from compressor and control system suppliers</td>
<td>Start-up with different gas (normal gas, air, nitrogen, defrost gas, etc.), trip, rundown, and normal shutdown, equipment failure</td>
</tr>
<tr>
<td>Commissioning and startup</td>
<td>Shop-tested compressor performance curves, as-built equipment data sheet, as-built control configuration, preliminary controller tuning parameters, and compressor operating procedure</td>
<td>Initial start-up with different gas (normal gas, air, nitrogen, defrost gas, etc.), variation of process conditions for control system check-out and tuning</td>
</tr>
<tr>
<td>Field support</td>
<td>Site-tested compressor performance curves and surge map, actual plant operating conditions, as-installed equipment performance test data, controller tuning parameters, and field modifications</td>
<td>Cases to be defined based on field request and prior incidents. To be used for field support, it is imperative that the dynamic model must be updated with as-installed equipment performance data</td>
</tr>
</tbody>
</table>

Initial Conditions

The starting conditions of dynamic simulation should be based on the equipment design conditions or preferably the actual performance data if they can be obtained. The initial conditions should cover a full range of operation with different combinations of maximum/minimum plant throughput, cold/hot temperature, and heaviest/lightest gas composition.

CASE STUDIES

Five case studies are presented in this paper. They are selected from a wide range of applications where dynamic simulation has proved to be a useful tool. The case studies are organized based on project life-cycle, starting from conceptual feasibility study, through detailed engineering, and finally to field troubleshooting. Table 4 is a list of these case studies.

Table 4. List of Case Studies.

<table>
<thead>
<tr>
<th>Case study</th>
<th>Project stage</th>
<th>Compressor configuration</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Conceptual feasibility study</td>
<td>Parallel trains with multiple compressors and a separate drive for each train</td>
<td>Feasibility of parallel compressor trains was evaluated, and critical design issues such as deviation tolerances were identified</td>
</tr>
<tr>
<td>2</td>
<td>Engineering</td>
<td>Multiple-stage centrifugal compressor</td>
<td>Dynamic simulation was used to specify the compressor protection system, including the number of recycle loops and the size of recycle valves. Simulation cases included startup, shutdown, and equipment failure</td>
</tr>
<tr>
<td>3</td>
<td>Equipment specifications</td>
<td>Multiple-stage propane compressor with helper/starter motor</td>
<td>Compressor driver size and selection were analyzed with dynamic simulation. Issues associated with full-pressure restart were discussed</td>
</tr>
<tr>
<td>4</td>
<td>Pre-commissioning</td>
<td>Parallel compressors with a high compression ratio</td>
<td>Simulation cases were performed to identify risks of compressor operation and evaluate measures to improve the overall control system</td>
</tr>
<tr>
<td>5</td>
<td>Field support</td>
<td>Multiple compressors in series, but on separate drivers</td>
<td>With the help of dynamic simulation, modifications were made to the isolated control system to reduce the response time of recycle valves and improve the overall compressor protection</td>
</tr>
</tbody>
</table>

Several case studies made reference to a helper/starter motor. Figure 6 shows a typical compressor line-up where a helper/starter motor is used. In this example, two compressors are driven by a Frame 7 gas turbine. Because the standard Frame 7 gas turbine has limited starting capacity (breakaway torque) for applications such as for the compressor train, an electric motor or similar driver is used for startup during the acceleration phase. The motor can also be used during normal operation to provide additional power for the compressors when required.

![Figure 6. A Typical Compressor Lineup with Gas Turbine Driver and Helper/Starter Motor.](image)

It is important to note that the results and conclusions from each case study were based on the design and operation data for a specific application. They should not be considered as general design criteria for other applications.

CASE STUDY I — FEASIBILITY STUDY OF PARALLEL REFRIGERATION COMPRESSORS

In a recent study, parallel compressor trains were proposed for a large-scale refrigeration compression system. Due to the significant amount of capital investment involved, a feasibility study was performed at an early stage of the project to confirm the feasibility of the proposed design and identify critical design issues that should be addressed on a priority basis.

Figure 7 is a schematic diagram of the parallel compressor trains. Each train consisted of two compressors in series, and was driven by a Frame 7 gas turbine and a helper/starter motor. One of the concerns with parallel operation was the interaction between the two trains, particularly the ability to start and maintain stable operation when the plant throughput changed or when one of the trains tripped or restarted.
Figure 7. Parallel Refrigeration Compressor Trains. Each Train Consisted of a Single-Stage and a Two-Stage Centrifugal Compressor, and Was Driven by a Gas Turbine and a Helper/Starter Motor.

Overview of Model

A dynamic simulation model was developed for the process. The compressor performance curves were based on the single-train compressors that were in operation and of similar size. Although the two trains were duplicates, it was expected that there would be differences in the actual performance between the trains. Therefore, deviations were imposed on the performance curves of the individual trains in the simulation model.

Deviations in Individual Performance Curves and the Effects on Parallel Operation

Simulation results showed that because the two trains shared the same suction and discharge, the overall head (i.e., pressure rise) across each train was more or less equal, but the flow distribution between the two trains was different due to the deviations in performance curves (Figure 8). If the curve of compressor B (shown as curve B1) was close to that of compressor A, the difference in load between the trains would be small, and both compressors would operate at the same distance from the surge limit. On the other hand, if the performance curve for compressor B had a lower head and a smaller head-rise-to-surge (shown as curve B2), compressor B would operate closer to the surge limit.

Based on these results, the proposed process configuration was feasible if the deviations between the two compressor trains were small. The authors analyzed the performance data for several facilities currently in operation, and found that the differences in the head curves between duplicate compressors were generally less than 1 to 2 percent. Therefore, it was concluded that the proposed parallel-train configuration was feasible based on the experience with duplicate compressors.

When the project progresses beyond the conceptual phase, more in-depth discussions are expected with the compressor manufacturer on the compressor specifications, particularly deviation tolerance and expected head-rise-to-surge. The simulation model will then be updated for design verification.

Operation Flexibility

Another objective of this study was to determine whether it was possible to keep one compressor train running after the other train was tripped. Simulations showed that without corrective actions, the remaining train could be stalled and eventually tripped due to the driver overload. Figure 9 shows that after compressor A was tripped (curve A), the load was initially shifted to compressor B (curve B). This was caused by the increase of compressor suction pressure when the forward flow to Train A stopped after the trip. As a result, compressor B was not able to stay online when its gas turbine driver could not meet the higher power demand and subsequently tripped on low speed. One proposed solution based on simulation results was to install a quick closing valve at the suction of each compressor train, so when a train was tripped the isolation valve to that train quickly closed to prevent overloading the remaining train. Since fast-closing isolation valves of that size have been in service for years, the proposed solution was considered to be feasible.
After a Compressor Was Tripped (Curve “A”), the Remaining Compressor Could Experience a Higher Load (Curve “B”).

**Driver Configuration**

The final driver selection for the project was two Frame-7 gas turbines. Other configurations were considered, as shown in Figure 10. Alternate 1 is using two variable speed drivers. Alternate 2 is the final selection as shown in detail in the previous Figure 7, with each compressor driven by its designated Frame 7 gas turbine driver with very narrow speed variation. Alternate 3 is with both compressors driven by a common Frame 7 gas turbine driver with narrow speed margin.

For all three configurations discussed previously, a fast closing valve is required for each suction and side stream to isolate a tripped compressor from the remaining string. These valves also give field engineers a means to test and adjust the compressor operating conditions after installation.

**Overall**

This case study showed a successful use of dynamic models to evaluate the feasibility of a conceptual design. In addition, critical design issues were identified at an early stage of the project, so they could be addressed promptly as the project progressed. In this case study, the dynamic simulation was completed over a short period of time and simulation results were available for review in a timely manner. This would not have been possible in the 80s and early 90s when simulation was more resource-intensive.

**CASE STUDY 2—LNG PLANT COMPRESSORS OPERATING PROCEDURE AND ANTISURGE PROTECTION**

This case study is a dynamic study performed during the detailed engineering phase of an LNG project. The focus of the study was a four-stage centrifugal compressor as the main part of the propane refrigeration compression system (shown in Figure 12). The compressor train was driven by a Frame-7 gas turbine and a helper/starter motor. The objective of the study was to verify the compressor protection system, including recycle loop configuration, recycle valve size, control system responsiveness, and implementation of operational procedures.
Figure 12. Configuration of Antisurge Protection System. Each Stage Had an Antisurge Valve for Protection During Normal Operation, and a Hot Gas Bypass for Protection Against Coastdown Surge. The LP Stage Had an Additional Bypass Valve for Defrost Gas Startup.

Case Definitions

Below show the scenarios studied to cover all critical operational aspects:
- Compressor train initial startup and restart after trip
- Compressor trip or emergency shut down
- Failure of equipment such as loss of cooling, trip of motor, and line blockages

For each scenario, different starting conditions were used based on combinations of maximum/normal/minimum plant throughput and cold/average/hot ambient temperature. This ensured that the most critical operating case was identified. In all, over 50 simulation cases were performed as part of the study.

Train Startup

One of the design requirements was to use different gas media other than the normal gas during the initial compressor startup. Data provided by the compressor and recycle valve manufacturers showed that the compressor performance curves and recycle valve capacity varied considerably as a result of the change of gas composition. To adequately size the recycle system for all startup gases, a number of simulation cases were performed as shown below:
- Study Cases for Train Startup—
  - Variation of startup gas—propane, nitrogen, and defrost gas (defrost gas is mainly methane with a small quantity of nitrogen and ethane)
    - Single recycle valve for each stage
    - Parallel valve for the first low pressure (LP) stage
    - Varying valve sizes

Figure 13 shows the simulation results for variation of startup gas. When the compressor was started up with propane, the recycle valve for the LP stage required a capacity (Cv) of 2000. However, when the compressor was started up with defrost gas with a molecular weight of 16, the recycle valve for the LP stage required a Cv of 5000.

Based on the simulation results and inputs from recycle valve and control system suppliers, a parallel bypass valve for the low pressure stage was added and final valve sizes were determined accordingly. The final configuration was shown in the previous Figure 12.

Emergency Shutdown

Compressor trip in the event of an emergency is a critical scenario that requires a series of instant actions to protect the compressor from surge during coastdown. Equipment placement, recycle line take-off and tie-in points, recycle valve size and responsiveness, and control logics are all dynamic variables that determine the effectiveness of the antisurge system. The dynamic model is required to precisely model the system variables as well as the operating sequences. For this study, in the case of emergency shutdown, a trip signal would initiate the following actions/responses:
- Trip of helper motor
- Trip of gas turbine, with the decay of turbine output as a function of fuel valve closure rate
- Energizing the actuators for the recycle valves and then opening of the recycle valves
- Closure of isolation valves at the compressor suction and discharge based on the valve stroke time

From the study, due to the constraints on equipment placement, there was a large system volume in the compressor discharge. It was found that the original configuration with single recycle loop from the compressor recycle cooler was unable to depressure the compressor discharge quickly enough to prevent coastdown surge. Based on the dynamic study results and consultation with the compressor and control system suppliers, hot gas recycle valves were added. These hot gas bypass valves were connected directly to the compressor discharge line and provided an additional recycle loop to depressure the compressor discharge quickly. The final configuration was shown in the previous Figure 12.

Surge Protection from Equipment Failure

After the configuration of antisurge circuit was determined based on startup and shutdown requirements, additional simulation cases were performed to evaluate the effectiveness of the protection scheme for other transient events, such as loss of air coolers and blockage of one of the lines connected to the compressor. The simulation results were reviewed by the compressor and control system suppliers, and it was concluded that the combination of cold and hot recycle valves as proposed was adequate for the transient cases identified.
Site Observations

The plant has been in operation for over two years. Site feedback indicated that the compressor protection scheme as determined based on the dynamic study results was effective in protecting the compressor for the full range of operation.

CASE STUDY 3—RESTART OF COMPRESSORS

Restarting a compressor in a safe, efficient way is important in minimizing the downtime of a facility. There are issues specific to compressor restart, such as power and torque capability of the starter motor and permissible system restart pressure. A typical restart curve is shown in Figure 14. The top curve is the torque generated by the motor, whereas the bottom curve is the load torque from the compressor train during the acceleration phase. The motor torque must be greater than the load torque with some margins; otherwise the motor may stall before the final speed is reached. Notice that the closest approach between the curves is often at an intermediate speed. To confirm the ability to restart the compressor, dynamic simulation is often used to predict the load torque curve for the entire acceleration phase.

![Figure 14. Startup Curves of Compressor with Starter Motor. The Circle Shows the Closest Approach Between the Two Curves at an Intermediate Speed.](image1)

Power and Startup Pressure

In this case study, the restart of a propane refrigeration compressor was analyzed. The configuration of the compressor system was shown in the previous Figure 12. As required, a starter motor was used for restart. Simulations showed that the power requirement of the motor was a function of the initial system pressure, and a motor of greater than 55 MW would be needed to restart from the settle-out pressure of 10 bara (Figure 15).

![Figure 15. Increase of Power Requirement with Higher Startup Pressure. The Curve Shows the Motor Power as a Function of Startup Pressure Based on Simulation Results. To Restart the Compressor Without Depressurizing (Full-Pressure Restart) Required a Starter Motor of Greater than 55 MW.](image2)

Economic Considerations

For similar propane compressors currently in operation, the motor size is less than 20 MW, so the system must be depressured prior to the restart. Depressurization results in a loss of refrigerant that is of high value and limited supply, as well as an extended shutdown of the system. The results of this case study show that by increasing the motor size to over 55 MW, it is possible to restart the compressor without depressurization. Economic analysis should be done, case-by-case, to determine whether the increase in unit run-time and reduction in loss of refrigerant can offset the cost of a larger motor and the upgrade of power generation. For this case study, it was concluded that the large motor size could be justified based on the overall cost matrix.

This case study demonstrated that dynamic simulation was able to provide valuable information to address issues associated with the compressor restart and assist in the specifications of compressor driver. It is important to note that these simulation results were based on design information specific to this project and should be verified on a case-by-case basis for other systems.

Other Issues in Compressor Restart

Besides power requirement, another important issue is the ability to restart the compressor without surging during the entire acceleration phase. The previous Case Study 2 has shown the use of dynamic model to size the recycle valve for startup. Where practical, the simulation should be extended to cover the entire restart procedure, including initial acceleration, commissioning of antisurge control, pressurization, establishing forward flow, and final compressor loading.

CASE STUDY 4—CONTROL SYSTEM CHECK-OUT BEFORE COMMISSIONING

This case study was based on a recent project where an offshore oil platform was constructed and commissioned. Figure 16 is a schematic diagram of the processing units on the platform, with details on the gas reinjection compressor trains. The incoming fluid to the platform was a mixture of gas, oil, and water from different production wells. The gas was separated from oil and water in two parallel separators, and then compressed from 10 bara to 200 bara by two parallel reinjection compressor trains. Each train had three compression stages and was driven by a variable-speed gas turbine. The high-pressure gas was eventually reinjected back to the wells to maintain reservoir pressure.

![Figure 16. Offshore Gas Reinjection Compressor Trains. All Instrumentations (PI, TI and FI) Shown on the Graph Are Connected to the Antisurge and Capacity Controllers. The Overall System Included Antisurge Control, Capacity Control, Load Sharing, and Pressure Override.](image3)

For this facility, a stable pressure at the separators was important in minimizing the slugging in the subsea flowlines between production wells and platform. Slugging is a phenomenon characterized by alternating plugs of liquid and gas bubbles, which is a common occurrence in multiphase pipelines and often causes operational problems such as pressure surge in the process system. In the control system that was originally proposed in the design phase, the separator pressure was maintained by varying the speed of the gas turbines. For instance, if the output from the production wells went down and caused a reduction in the separator pressure, the gas turbines would slow down and the compressor trains would operate at a low speed until the separator pressure recovered.
Potential Risk of Unstable Operation

The contractor that built the facility was required to demonstrate stable, uninterrupted operation for a sustained period of time during the commissioning phase. The contractor considered the reinjection compressors to be the critical element in achieving uninterrupted operation, because:

- A trip of reinjection compressor was common based on past experience, and
- A shutdown of the compressor trains would automatically stop the production from the platform.

Further analysis identified several risk factors in maintaining stable operation of compressor trains, as listed below:

- The operating conditions differed from the design conditions during the commissioning. The facility would be operated at about 50 percent of the design throughput since all the production wells were not yet connected. It was not clear whether the parallel compressor trains would operate in a stable manner at a low throughput.
- The gas flow to the platform could be unstable during the commissioning, since the wells were being tested and the number of wells that were connected to the platform could change at any given time. It was not clear whether the parallel compressor trains would operate in a stable manner at low throughput.
- The gas compositions to the two parallel trains were different due to the alignment of wells and the asymmetry in pipe routing on the platform. This presented additional challenges in parallel compressor operation.

Enhancement of Control Scheme

To evaluate these risk factors and identify improvement to the control system that would mitigate the risks, the contractor performed a dynamic simulation for the entire platform, with emphasis on the operation and protection of the reinjection compressor trains. One of the simulation cases was a shutdown of half of the production wells. Figure 17 shows that, based on the original control scheme where the pressure was maintained by varying the compressor speed, the separator pressure could drop to 8 bara after a sudden reduction in gas throughput. The pressure of 8 bara would result in a low-pressure alarm, which presented a risk of compressor trip on low-pressure. The results were reviewed by the supplier of the control system, and with their collaboration a pressure override control (POC) was added to the control system. The POC would open the recycle valves of the first compressor stage when the separator pressure started to decrease. As confirmed by simulation results shown in Figure 17, the POC minimized the pressure fluctuation in the separator and reduced the risk of compressor trip on low-pressure.

Load Sharing

Dynamic simulation was also used to fine-tune the load sharing scheme. Simulation showed that after a reduction of gas throughput, the compressor trains would operate in partial recycle. Without load sharing, it was possible that the opening of recycle valves differed considerably between the parallel trains due to asymmetry in piping and gas feed. This can be seen in Figure 18: Without load sharing, the recycle valve on Train A was almost fully open whereas the one for Train B was almost closed. As a result, Train A would be vulnerable to surge if the gas throughput was further reduced. Figure 18 also showed that if load sharing was used, the opening of recycle valves of the two trains would be about equal, so both recycle valves had additional capacity to handle further disturbance. The effect of load sharing can also be observed in Figure 19 where the compressor operating points predicted by the simulation model were plotted. Both compressors initially operated near the normal speed. After a reduction of throughput, both compressors moved to a lower speed. Without load sharing, the operating points between the two parallel compressors were far apart, with compressor A near the operating boundary. On the other hand, if load sharing was used, the operating points of the two trains were closer, and both trains had room to accommodate further reduction in the gas throughput. The simulation results were given to the control system supplier for their use in configuring the load sharing scheme.

Figure 18. Improving Compressor Protection with Loading Sharing. After a Reduction in Gas Feed, Both Compressors Were Operated in Recycle. Without Load Sharing, the Recycle Valve for Compressor A Was Close to Full Open, So the Compressor Became Vulnerable to Surge If There Was Further Reduction in Gas Feed.

Figure 19. Improvement of Parallel Compressor Operation with Load Sharing. With Load Sharing, the Operating Points of Both Compressors Were Away from the Operating Boundaries, Which Provided Additional Operation Flexibility.

Pretuning Controllers

In addition to the control system for the compressor trains, dynamic simulation was used in pretuning the controllers in other units to minimize the risk of unstable operation. Simulation results showed that pretuning the controllers for the separators reduced the feed fluctuation to the gas compressors and helped achieve stable, uninterrupted operation.
With the assistance of dynamic simulation, potential risk factors were evaluated prior to the commissioning and solutions were developed to mitigate risk. The facility was successfully started up, and turned over to the owner shortly after the startup.

CASE STUDY 5—FINE-TUNING COMPRESSOR ANTISURGE CONTROL

Recently, dynamic simulation was used to address issues associated with compressor services on separate line-up and drivers. Figure 20 is a schematic diagram of the MR compressor train in a facility that was recently constructed. MR is a mixture of methane, ethane, propane, and nitrogen. The compression system included three MR compressors in series (K-001, K-002, and K-003). There were antisurge controllers and recycle valves for each compressor. One key feature in the configuration was that the LP MR and medium-pressure (MP) MR compressors were on one string with a gas turbine, but the high-pressure (HP) MR compressor K-003 was on another string with the propane compressor K-004 and another gas turbine. As a result, there were considerable interactions between the two rotating assemblies.

![Figure 20. Compressor Line-Up with the High-Pressure Compressor K-003 Tied to the Propane Compressor K-004. When K-004 was Tripped, the Recycle Valve for K-003 Was Tripped Fully Open, Which Resulted in a Reduction in Flow to K-001 and K-002 over a Short Period of Time.](image)

Field Observations

The design intent was to keep the LP and MP MR compressors (K-001 and K-002) in recycle if the HP MR compressor and propane compressors (K-003 and K-004) were shut down. However, shortly after the plant was started up, it was reported that when the propane compressor tripped, the LP and MP MR compressors quickly approached surge and were subsequently tripped on surge count. Further analysis of site data showed that the flow across the LP and MP MR compressors (K-001 and K-002) was reduced quickly when the recycle valve for the HP MR compressor (K-003) was tripped open. Due to the delays in the actuator response and valve stroke, the original surge detection and prevention measures were not adequate. Considering the compressors could be damaged by a small number of surge events, there was a compelling need for developing an improved control solution.

Modifications

In collaboration with the controller system supplier, a feed-forward (FFW) action was added to the antisurge configuration of the LP and MP MR compressors (K-001 and K-002). The feed-forward action would open the recycle valves of K-001 and K-002 on trip of the HP MR (K-003) and propane compressor (K-004). Simulation results show that the feed-forward action reduced the delays in the response of recycle valves, and the LP and MP MR compressors were able to operate in total recycle outside the SLL (Figure 21).

![Figure 21. Improvement of Compressor Protection with Feed-Forward (FFW) Action. With FFW, the Recycle Valves for K-001 and K-002 Started to Open as Soon as the Downstream Compressor Was Tripped, Which Reduced the Delays in Valve Response and Kept the Compressors out of Surge.](image)

This solution has been found to be effective in improving the compressor protection based on field testing, and will be considered for similar compressor lineups in the future.

SUMMARY

Compressors represent a significant capital investment and operating expense, and are an essential component in providing revenue streams of a plant. Therefore, compressor operation and protection are critical issues in all phases of the project life-cycle. Dynamic simulation has become a powerful tool for leveraging simulation technology to assist the design and operation of compressor systems. With collaborations with equipment and control system suppliers, a high-fidelity dynamic simulation model can be developed as a value-added tool, even at early phases of a project. As demonstrated in the case studies, the dynamic model allows for studying the performance of a compressor train under the full range of operation and eventually improving compressor efficiency, availability, and protection.

REFERENCES


BIBLIOGRAPHY


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