ASIA TURBOMACHINERY & PUMP SYMPOSIUM
SINGAPORE | 22 – 25 FEBRUARY 2016
MARINA BAY SANDS

**Tutorial: Transient Modeling and Analysis of Centrifugal Compressors**

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## ABSTRACT

Centrifugal compressors are subject to transient events, such as emergency shutdowns, which can cause energetic surge events during rapid shutdown transients. Many modeling tools are used to predict the behavior of compressor systems during fast transient events. Even more, centrifugal compressor dynamic modeling is a valuable assessment tool that can help improve the design of the compressor anti-surge system to prevent harsh conditions while the unit is coasting down. Modeling of centrifugal compressor transient events requires a detailed evaluation of many system variables to obtain accurate results. Additionally, many parameters should be included and analyzed to adjust the model and obtain acceptable predictions. Those parameters can include recycle valve characteristics, the controller and actuator, suction and discharge piping volumes, aftercoolers, and operating conditions. Other parameters, such as machine coast down speed and inertia and different control response times will determine if the compressor will reach its surge limit at a detrimental high head condition, avoid it or go through it at low acceptable energy conditions. Many designs use general guidelines to select and plan the compressor anti-surge system and its main components. For example, the recycle valve type, size and actuation time are key parameters during the shutdown of the compressor unit since they directly affect the amount of flow through the compressor and its reaction time. Other factors are also relevant and should be assessed in detail to assure a proper design to avoid such transient events as a sudden shutdown of a compressor from a full load or high head operating conditions resulting in damaging surge events.
In order to refine the simulation approach of the transients in centrifugal compressors, an extensive assessment has been conducted of the main parameters that will affect the modeling predictions. Parameters such as recycle valve size, type, actuation time, after-cooler volume, friction factors, unit speed coast down predictions, compressor map, isolation valve control, acting time, as well as placement of the discharge check valve influence the modeling predictions considerably. Therefore, parametric studies of some of those variables have helped to refine and adjust the modeling technique while improving the accuracy of the results. Moreover, computational predictions have been compared against high fidelity – high accuracy data collected in a full scale compressor system. Initial comparisons indicated reasonable results while adjustments in the technique and main assumptions improved the modeling predictions considerably.
A generic methodology to improve modeling predictions and main considerations are part of the analysis. In addition, a comparison of the modeling results and experimental data are presented and complemented with parametric studies of different variables. In general, this work should provide guidelines for advancing the modeling of centrifugal compressor transients as well as showing the application of a valuable tool for designing surge control systems for centrifugal compressors.

## NOMENCLATURE

| | | |
|---|---|---|
| A | Area | ($m^2$) |
| ASV | Anti-Surge Valve | (-) |
| Cv | Valve Coefficient | ($GPM/PSI^{0.5}$) |
| D | Diameter | (m) |
| ESD | Emergency Shutdown | (-) |
| H | Head | (kJ/kg) |
| J | Rotational Inertia | ($kg\text{-}m^4$) |

| | | |
|---|---|---|
| N | Rotational Speed | (RPM) |
| p | Pressure | (bar) |
| P | Power | (kW) |
| Q | Volumetric Flow Rate | ($m^3$/h) |
| SM | Surge Margin | (%) |
| TD | Turndown | (%) |
| T | Torque | (Nm) |
| t | Time | (s) |

## INTRODUCTION

Centrifugal compressors are required to rapidly shutdown or trip in the event of high vibration, loss of lube oil, or a similar alarm condition, and to avoid energetic surge events during these rapid shutdown transients. The recycle valve and its controller and actuator as well as the suction and discharge piping volumes, aftercoolers, operating conditions, other parameters, and various control response times are all characteristics that govern whether a compressor will reach its surge limit at a detrimental high-head condition or at a more satisfactory low-head condition. Although there are rules of thumb and other general guidelines for designing recycle valves and surge control systems, the only way to ensure the proper design of a sudden shutdown of a compressor during a full load or high head operating condition without resulting in a damaging surge event is to simulate the transient events that take place immediately after a trip. Detailed simulations are necessary to design surge control systems that prevents damaging high-energy surges. Centrifugal compressors may experience surges whenever they are suddenly tripped, however, controlling the head and total energy of the surge event by creating a sufficient recycle valve opening and a low enough head before the surge event occurs is the objective of transient simulations.

Currently, computational models are very common and useful for providing quick, reliable, and cost-effective solutions to real problems. In general, pipeline models include a lot of detailed and specific information about the actual system being modeled. Therefore, it is important that this data be accurate in order to ensure the predictive capability of the computational model. In addition, there are some key parameters and techniques that can help to increase the accuracy of the model predictions. A similar version of this tutorial was presented at the 2015 Middle East Turbomachinery Symposium (Garcia-Hernandez, A., and Bennett, J.A).

## SURGE

Surge is an unstable operating condition that occurs when the compressor can no longer maintain a desired pressure ratio (head) due to insufficient flow. Therefore, compressor maps are drawn with a surge line to indicate the minimum flow required for pressure ratios available during normal operating speeds. Under normal conditions, the compressor operates to the right of the surge line. However, as fluctuations in flow rate occur, or under startup / emergency shutdown, the operating point will move towards the surge line due to a reduced flow. If conditions are such that the operating point approaches the surge line, the impeller and diffuser begin to operate in stall and flow reversal occurs. The flow separation will eventually cause a decrease in the discharge pressure and flow from suction to discharge will resume (Brun and Nored (2007) and Garcia et al. (2009)).

The surge cycle will repeat itself unless control systems are installed or operational changes are made to bring the compressor out of the surge cycle. The surge cycle may result in a small or large flow reversal period depending on the discharge gas volume and the pressure ratio. Chronic surge is characterized by intermittent periods of small flow reversal that may not cause severe damage to the machine. Acute surge is more pronounced, usually due to a rapid transition across the surge line. Any surge event can cause severe damage to the thrust bearings, seals, and the impeller. The extent of the damage due to surge occurrence is somewhat a result of the compressor design (Brun and Nored (2007) and Garcia et al. (2009), McKee and Garcia (2007)).

Design of the anti-surge control system involves several considerations such as selection and engineering of the piping and valves, together with the selection and the placement of instruments. Moreover, those parameters affect the performance of the control system and machine. If this is not done properly prior to the final design or engineering of the compression system it could result in several issues even during normal operation. Reengineering or equipment repairs could be more costly and time consuming than the initial design (Brun and Nored (2007) and Kurz (2009)). A typical configuration for a recycle system is outlined in Figure 1.

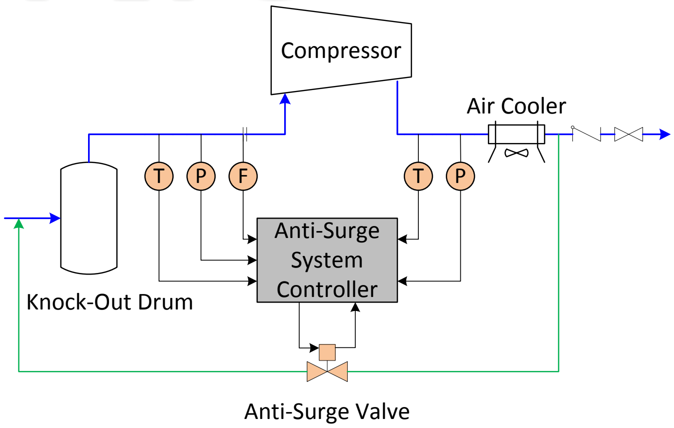


**Figure 1. Anti-Surge System Schematic**

At a minimum the anti-surge system should include pressure and temperature sensors in the suction and discharge of the compressor, a flow metering device in the inlet piping, a recycle or anti-surge valve, an aftercooler, and a check isolation valve. The measured parameters are continually monitored and used by the control system to determine whether or not the compressor is operating and it then provides feedback to the machine and recycle valve to maintain the unit in a safe condition. A surge or safety margin limit is usually used to define how close the compressor is operating from its surge line. The surge margin (SM) is defined by:

$$SM = \frac{Q_{op} - Q_{surge}}{Q_{op}} \Big|_{N=const} \qquad (1)$$

Similar to surge margin, an alternative measure of the surge limit is turndown (TD) and is defined by:

$$TD = \frac{Q_{op} - Q_{surge}}{Q_{op}} \Big|_{H=const} \qquad (2)$$

The anti-surge control should be designed to react to normal operating fluctuations; process upset conditions, and harsh transient events such as an ESD. In addition, the control system includes normal start-up, shutdown, and load sharing sequences.

## TRANSIENT MODELING METHODOLOGY

In order to summarize the modeling methodology a flow diagram of the basic and most important steps is presented below in Figure 2. In general this methodology includes key points and general guidelines to accurately model the transient behavior of centrifugal compressor systems.

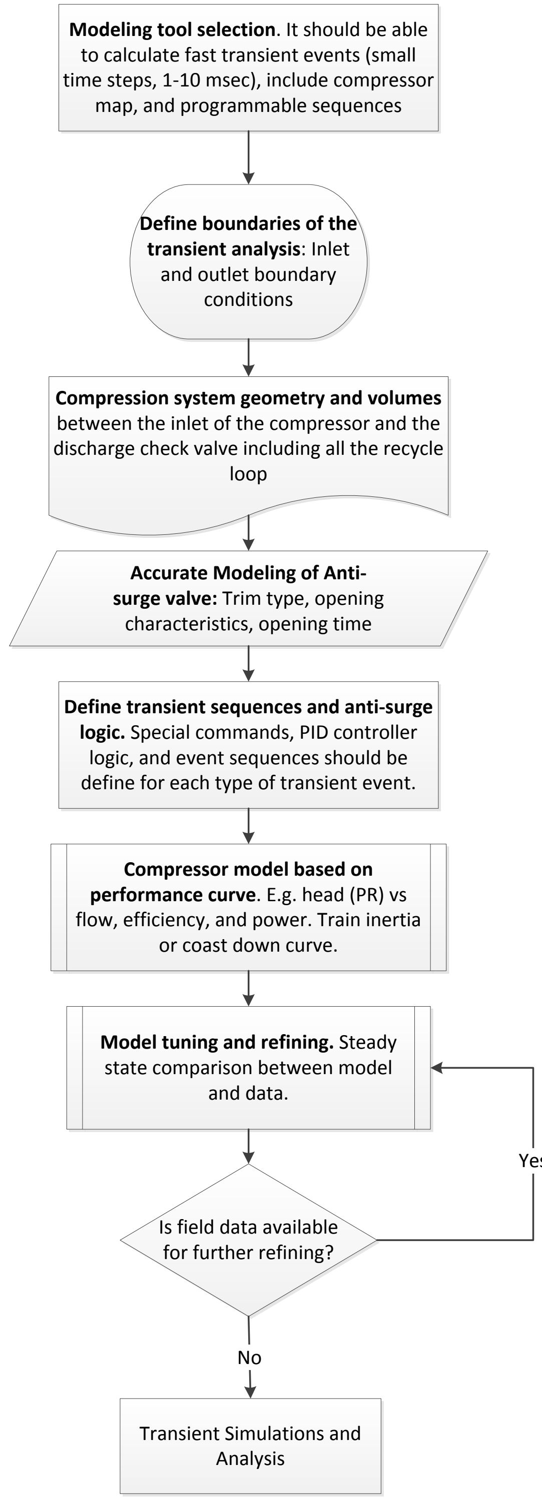


**Figure 2. Modeling Methodology**

## MODELING KEY PARAMETERS AND GENERAL GUIDELINES

For any simulation of the transient behavior of the system, the following features of the system must be known(Garcia et al. (2009), Morini et al. (2007) and White and Kurz (2006)):

- The geometry of the piping system, in particular the volumes of the piping and vessels (after-cooler and KO-drums)
- The opening and closing characteristics of the valves (flow coefficients and type of recycle and check valves)
- Recycle valve timing (opening and any signal delay)
- The inertia of the compressor train or speed coast down curve of the machine
- The thermal inertia of the driver
- The compressor performance map
- Control logic, special commands, and sequences
- Any known delays in the surge control system
- Boundary conditions/ expected flow conditions
- Compressor driver details and operating strategy
- Load sharing strategy

A large amount of detailed data is required in order to develop an accurate and complete model of a centrifugal compressor and its controls. The complete piping system with lengths, diameters, and all branches or connections are defined and input into the model builder. Details such as volumes, lengths, and heat transfer surface areas are provided for coolers, heaters, scrubbers, or filters. The rotational inertia of the compressor with its driver and the torque characteristic of the driver are needed so the rate of coast down or start up can be simulated.

## COAST DOWN SPEED

One of the key challenges in ESD simulations is to determine the correct deceleration of the train. The rate of deceleration is determined by the inertia of the train J (referenced to compressor speed (N) for geared trains), any residual power generated by the gas turbine ($P_{GT,resid}$), the friction losses in the system, and the power absorption of the compressor (Garcia et al. (2009) Morini et al. (2007)).

$$P = P_{compr} + P_{frict} - P_{GT,resid} = T \cdot N = -J \cdot N \cdot \frac{dN}{dt} \qquad (2)$$

$$\frac{dN}{dt} = \frac{P_{compr}(t) + P_{frict}(t) - P_{GT,resid}(t)}{-(2\pi)^2 \cdot J \cdot N(t)} \qquad (3)$$

Solving Eq. 2 for steady-state power and speed at the initial operating point and assuming a constant Q/N the integration of Eq. 3 yields the following expression for the speed decay:

$$N(t) = \frac{J\ N_o^{\ 3}}{J\ N_o^{\ 2} + P_o\, t} \qquad (4)$$

Where:
J: Inertia of the train
$N_o$: Initial speed
$P_o$: Initial shaft power (compressor +bearings friction losses)
t: Time
$N_{(t)}$: Speed versus time

Another method to simulate the speed decay of the compressor is to generate a ramp down function using the measured compressor speed coast down data, if available. A comparison of the measured and calculated coast down speed for a centrifugal compressor is presented in Figure 3. The main difference is the residual power generated by the gas turbine after the fuel valve is closed which is very difficult to determine. Also, bearing friction losses and the inertia of the train are affected by every single component of the train and sometimes this information is not available. However, in both cases, if the right considerations are taken, the calculations will match in the first few seconds of the coast down which are the most important for the dynamic event.

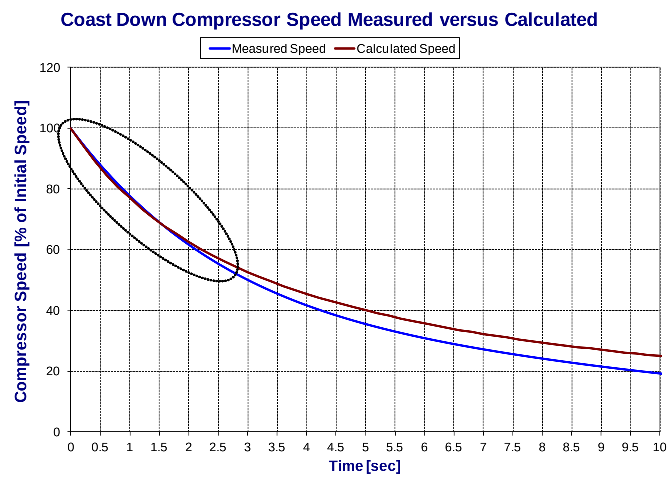


**Figure 3. Compressor Speed Coast Down Comparison**

## PIPING AND GEOMETRY

Piping and geometry changes such as reductions not only affect the volume considerations for the system; but also influence the impedance of the system which is the dominating factor for the initial pressure wave propagation when a transient occurs (Sparks (1983), Botros and Ganesan (2008)). The system's impedance can affect the initial dH/dQ behavior of the compressor until the system flow resistance starts dominating the movement of the gas flow (Sparks (1983), Botros and Ganesan (2008)). Thus, in general the piping layout should include the main piping volume and its acoustic length as well as any flow restriction elements.

Another important consideration is the after-cooler volume and friction factor since the reactance and resistance components of the impedance for this element could affect the initial pressure wave and also generate some attenuation of it. Typically, an equivalent or hydraulic diameter for the discharge after-cooler is calculated using the wetted perimeter concept. The total area of the after-cooler is calculated by multiplying the number of tubes by the area of each tube and then the hydraulic diameter is determined utilizing Equation (5). In addition, the total volume of the cooler is calculated by adding the volume of each tube. It was found that the volume and friction factor of the cooler affect the simulation results significantly.

$$D_h = \frac{4 * A_{tubes}}{U} = D_{tube} * \sqrt{N_{tubes}} \tag{5}$$

$D_h = Hydraulic\ diameter,\quad U = Wetted\ Perimeter,$

$D_{tube} = Tube\ diameter,\quad N_{tubes} = Number\ of\ tubes$

A comparison of the cooler friction factor is presented in Figure 4. In the first case a smooth pipe friction factor was used and then an experimental friction factor (approximately 20% higher) was included to observe its effect in the simulation results.

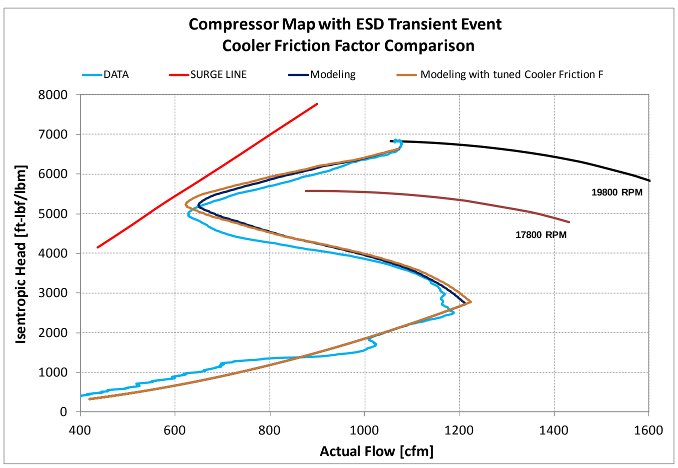


**Figure 4. Cooler Friction Factor Comparison**

A positive effect was obtained when the experimental value was implemented in the simulations and model predictions improved with respect to the experimental data. The experimental cooler friction factor affects the impedance of the discharge piping system. Therefore, less flow resistance is obtained on the discharge side of the compressor, the recycle loop system, which slightly increases the actual flow during the transient of the compressor. This example illustrates that a more refined friction factor for the cooler will improve the model prediction. However, field data for tuning friction factors is not always available; thus, it is important to ensure that the real cooler volume, and inlet and outlet effects are considered for the transient predictions.

## RECYCLE OR ANTI-SURGE VALVE

The dominant control method for reducing stored discharge pressure energy or head across the compressor is the recycle valve or unit bypass valve. The larger the recycle valve, the more flow that can be moved from the discharge side of the compressor to the suction side. The speed of the recycle or bypass valve opening is also important in rapidly reducing the discharge pressure and hence, the head and stored energy at the compressor. One important tradeoff in relation to recycle valves is that, in general, the larger the valve, the slower its opening time. There are certainly cases when a faster small valve is better than a larger slow valve but in all cases, the recycle valve must be large enough to handle the flow supplied by the compressor with a pressure differential sufficiently high enough to store energy once the recycle valve is open (McKee and Garcia (2007)).

In some compressor installations, it is necessary to have both a normal recycle valve line, which includes a cooler within the re-circulation loop and a hot gas bypass valve line, which can be opened to rapidly transfer gas back to the suction side of the compressor without cooling. A hot gas bypass allows the head across a compressor to be rapidly reduced. The primary advantage of transient flow modeling is that various options for recycle and bypass valve size, opening time, arrangement, and sequence can be simulated and compared. Parametric studies of the surge control valve size and opening time can yield optimum sizes for recycle valve applications. Figure 5 shows an example of the difference in the discharge pressure (related to head) at which a shutdown surge occurs based on the size (Cv) of the recycle valves.

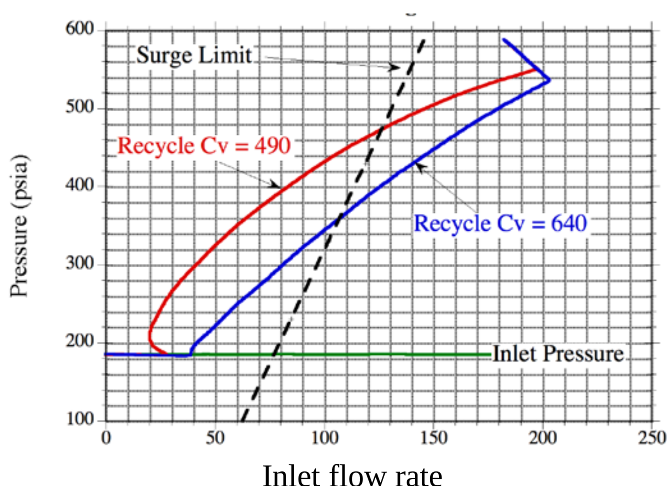


**Figure 5. Comparison of Two Different Recycle Valves during a Sudden Trip**

A sensitivity analysis provides a better understanding of system behavior when specific parameters are modified. Additionally, it helps to determine the effect of one variable while others remain unchanged. Figure 6 shows a comparison between test data and simulation data. The simulation was conducted with different assumptions made about the flow capacity of the recycle valve.

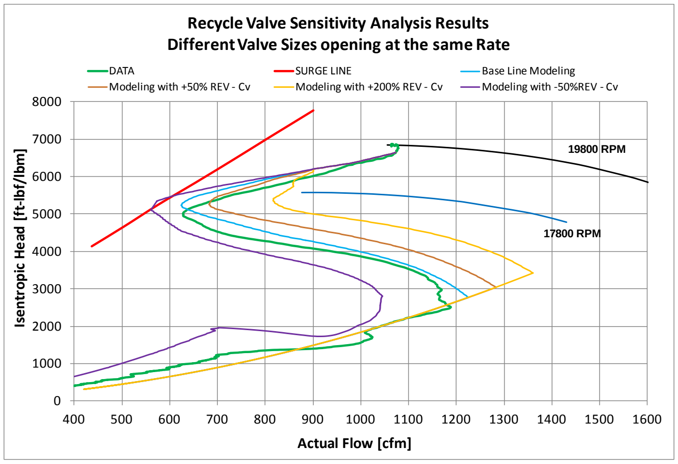


**Figure 6. Anti-Surge Valve Sensitivity Analysis Results**

The assumptions range from a recycle valve with 50% capacity of the valve actually installed, to 200% capacity. The baseline condition, which models the actual flow characteristics of the valve, shows a good correlation to the measured data. An ASV with half of the flow results in a minimum flow near the actual surge line. In this example, the system could benefit from a larger valve to reduce the likelihood of surge.

The opening trend of the anti-surge valve will affect the amount of mass recirculated early in the transient event; thus, based on the type of valve more or less flow can be diverged to the suction of the compressor for the same opening time. This effect is very critical for a system that required an initial boost in the suction flow or rapid reduction of the discharge pressure to avoid the surge condition. A vast variety of recycle valves are available on the market; however, it is critical to determine what type of opening curve is required for each specific system application. For example, a quick opening valve could work well in a single-stage machine that tends to work near its surge margin while the same valve would not work for a multi-stage machine since it will withdraw excessive flow from the suction side of the second stage early in the transient event; thus, it could cause the second stage to surge more rapidly. Normally, there are three types of recycle valves installed in compression systems. Those valves are linear acting, quick opening, and equal percentage. Figure 7 presents a comparison of a quick opening and a linear acting valve with the same opening time and maximum flow coefficient.

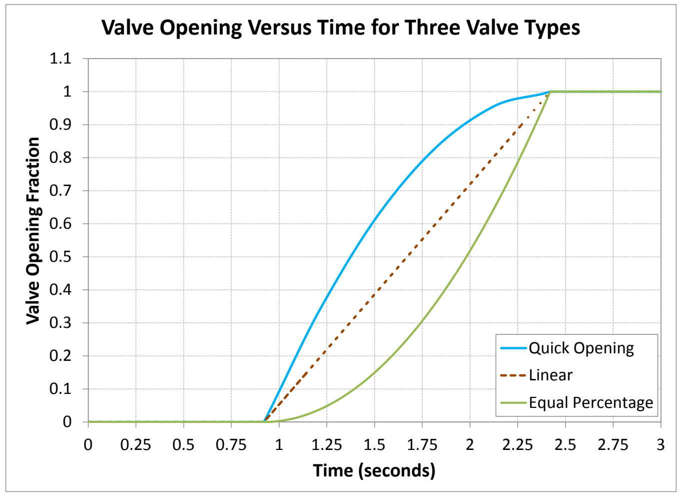


**Figure 7. Valve Opening Comparison**

Figure 8 presents a case where a quick opening valve was used to reduce the possibility of surging the compressor. In this case the initial system design used a linear acting valve and several vibration issues were encountered during the trip of the machine. The dynamic analysis indicated that the unit was getting very close to its surge condition; thus, an initial recommendation was made to modify the trim assembly of the recycle valve to make it quick opening since no modifications in the actuator and control logic (opening time) were possible. As shown in Figure 8 the linear acting valve will bring the unit near its surge condition while the quick opening provided an additional 4.2% in suction flow during the ESD; thus, the surge margin was sufficient to reduce the surge occurrence. The quick opening behavior provided a faster reduction in the discharge pressure as well; thus, the discharge check valve closed a little earlier eliminating the possibility of flow reversal.

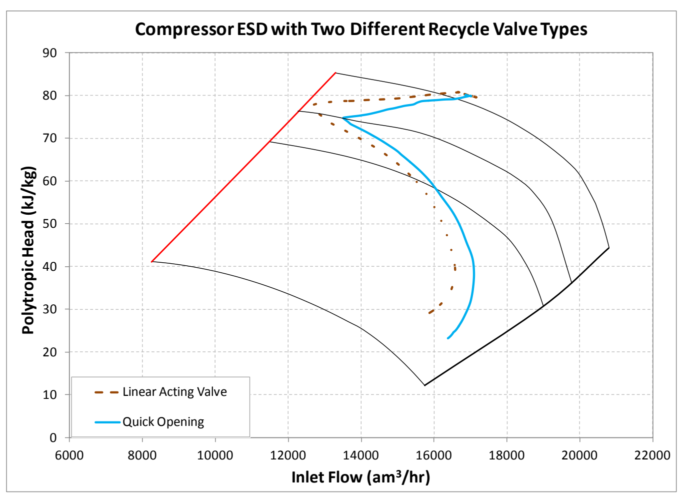


**Figure 8. Anti-Surge Valve Sensitivity Analysis Results**

Other important elements that may be present in the compression system are the vent valves which are used to release energy in a very quick manner. Most vent valves are fairly small and can be opened rapidly. Given the high pressure ratio between a discharge pressure and a vent or flare header, the flow through the vent valves is usually sonically choked and mass flow rate is directly related to line pressure. Therefore, vent valves are most effective when they are first opened and as the line pressure decreases, the flow rate through the valve decreases substantially. It is not a simple calculation process to properly size vent valves to limit surge energy following a trip. However, the optimum size and timing for a vent valve, if one is used, can be determined by modeling the entire compressor system and the surge control actions. An example of different discharge vent valve release times is presented in Figure 9. Therefore, incorporating the correct timing and flow coefficient for the vent valves is essential for obtaining accurate predictions in the dynamic analysis.

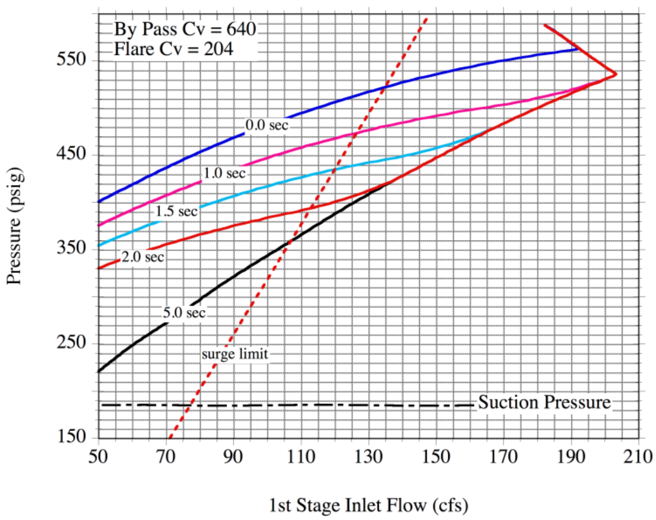


**Figure 9. Vent Valve Sensitivity Analysis Results**

## COMPRESSOR MAP

Compressor maps are generated from theoretical calculations or performance test data with the main assumption that the machine is operated in steady state at all times (Morini et al. (2007), Morini et al. (2009)). Compressor units are exposed to harsh transient conditions during ESD and other process upset conditions; thus, there is an uncertainty regarding the transient predictions. However, it has been documented that steady state data is suitable for use in simulations of fast transients while keeping in consideration the impact on the apparent compressor efficiency and the absorbed power due to the heat transfer effect (Blieske et al. (2010)).

It has also been found through real data comparison that steady state compressor maps are a good approach for predicting transient conditions (Garcia et al. (2009), Moore et al. (2009)). The accuracy of the compressor map will represent the most important parameter in defining the transient predictions since actual performance conditions of the machine can be shifted from the theoretical calculations; thus, an initial tuning of the compressor map to match the real conditions is highly recommended prior to commencing the transient evaluations. The ideal situation would be to obtain field performance data to build and refine the computational compressor model. An example of this is presented in Figure 10. In this case the theoretical predictions matched quite well with the test data; therefore, no tuning was required. In addition, the differences in head and flow from this comparison can be used to calculate uncertainty values which can be used to report the accuracy of transient results and ensure that any narrow margin is within the acceptable limits.

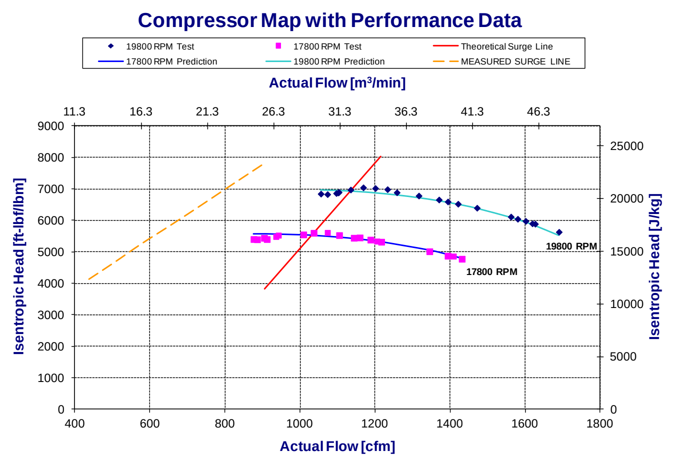


**Figure 10. Compressor Map Theoretical Predictions and Performance Test Comparison**

## CONTROL SEQUENCES

Dynamic simulations of centrifugal compressors must follow a detailed transient sequence which is designed to activate different elements in a specific order; thus, the control system (anti-surge logic) protects the unit from a surge condition. The anti-surge control logic function and parameters should be included in the model logic. The order of events in an ESD are typically started by tripping the unit; opening the recycle valve; closing safety isolation valves; and opening the blowdown valves while different delays may occur between each action which are very critical as well. Even small changes in the different actions' delays may significantly affect the predictions of the system. A comparison of different control sequence delays and actions is presented in Figure 11. In this case, an incorrect delay in the opening of the recycle valve could lead to an inaccurate surge prediction while the real system is responding adequately. In some cases it is difficult to obtain the exact timing of events; thus, simulations should investigate the sensitivity of various parameters of the anti-surge control system such as the recycle valve opening time, the delay of the opening of recycle and blowdown valves as well as coast down delay.
Typical anti-surge system control parameters built into a

dynamic simulation are ESD sequences, surge margin, surge control line, and boost line. In addition, more details about the PID controllers and sensors (gains, error bands, integration and derivative parameters, etc.) may be included in the simulation; however, that requires a much higher level of effort.

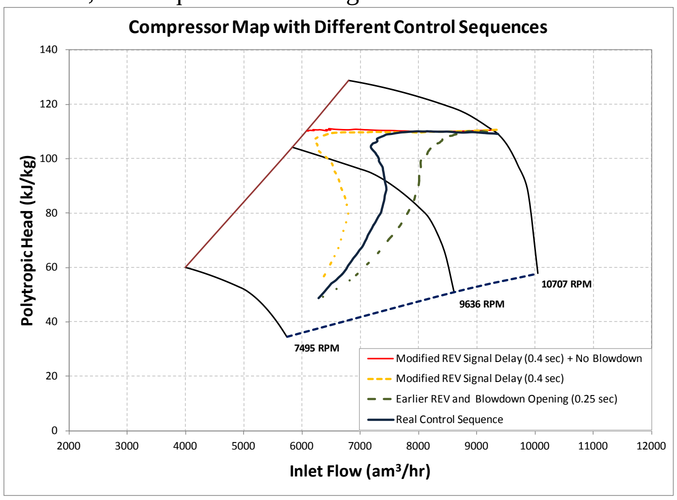


**Figure 11. Compressor Map with Different Sequences**

## CASE STUDY

A three-stage centrifugal compressor train barrel type is driven by a gas turbine; thus, all the stages are connected through a common shaft. Mechanical couplings are installed between the GT driver and the multi-stage compressor. Each stage has its own dedicated anti-surge loop and cooling system. The trains are fed with natural gas from a processing facility and the design flow and compression ratios are different for each stage. Moreover, some stages are fed through more than one stream at a time. The compressor trains have low pressure (LP), medium pressure (MP) and high pressure (HP) stages and each one of them includes a suction scrubber, cooler, recycle loop, and blowdown line. In addition, each train is fitted with an integrated turbine compressor control panel. Figure 12 presents a typical diagram of the compression trains and its main components.

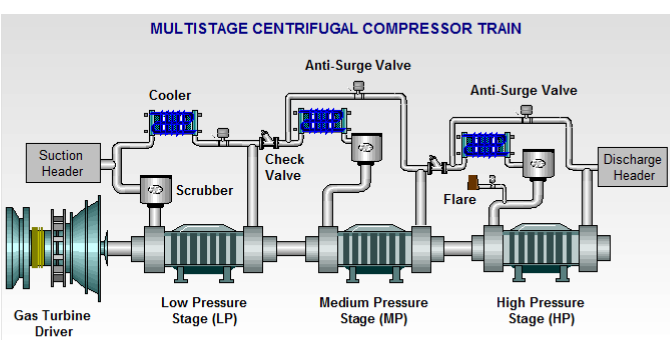


**Figure 12. Multi-stage Compressor Train Diagram**

Three compressor trains are used to deliver natural gas to a pipeline system from a processing facility. Thus, they are key components of the production process; moreover, any unexpected shutdown or failure can originate upset conditions that will affect the process and pipeline deliverability. In the past, those units have presented different types of mechanical and control failures. Therefore, a detailed analysis of the system was required to mitigate the process and control issues. Thus, the three multi-stage compressor trains were dynamically analyzed to determine if their anti-surge logic and controls were operating properly. In addition, it was very critical to understand and deeply evaluate the interaction of the different stages with their surrounding stages and components, since minor changes in one stage will affect the transient behavior of the neighboring stages.

Figure 13 shows the surge margin of the three stages of the compressor during the ESD. In this figure, it is clear that both the second and third stages surged. The ESD is shown to have little effect on the operating point of the first stage. This is most likely due to two factors: 1) the recycle valve of the first stage was already opened to a fraction of 31% at the beginning of the ESD event, and 2) the first stage recycle valve is the first valve that opens during an ESD event and has the quickest opening time. All of these factors reduce the effect of the compressor shutdown on the surging of the first stage. The first stage had a minimum surge margin of 18.4% during the transient simulation, which is considered acceptable.

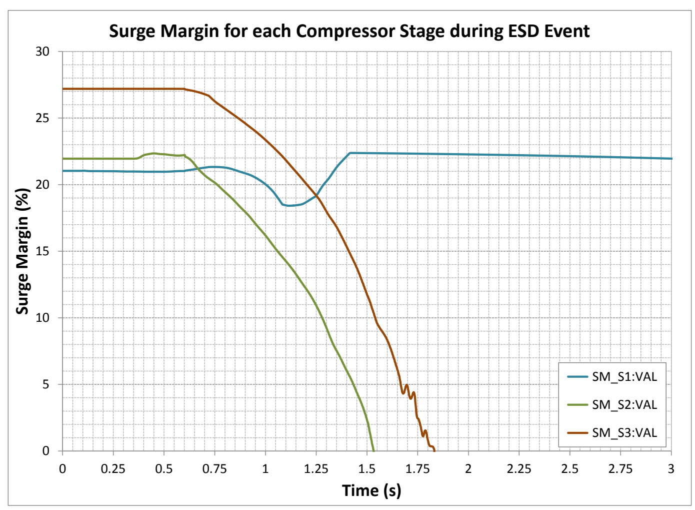


**Figure 13. Surge Margin Values during an ESD Event for the Existing Configuration**

The second and third compressor stages experienced a surge event during the simulated ESD for the existing anti-surge configuration. Those two stages presented the slowest opening time and the longest signal delay. Thus, they did not have sufficient flow to recover from the rapid speed reduction that originated during the ESD. Moreover, as soon as the recycle loop of the first stage started drawing more flow it affected the dH/dQ behavior of the second and third stages, since the dQ component decreased rapidly. Thus, the compressors move to the surge condition quickly.

Contrary to the first stage, the flow and head in the second and third stages reduced rapidly bringing the compressors to a surge condition. It was observed that the anti-surge valves of both units were not able to divert sufficient flow to the suction of the compressor stage during the transient. Moreover, the second stage valve has the slowest travel time and the third stage valve has the longest delay time. Thus, those conditions affect the behavior of the system, since the high pressure energy is maintained in the discharge of both units for a longer period of time during the coast-down of the unit. Therefore, the lack of enough flow in the suction side and the high energy present at the discharge (head) during the transient event cause a harsh surge condition in both compressor stages, which are not acceptable even for a small period of time.

Therefore, the current anti-surge design was not sufficient to protect the compressor train when it experienced an ESD event from a normal operating condition. Thus, several emergency shutdown simulations were completed on the model of the multi-stage compressor and piping systems. These were simulated at the provided operating conditions. The same starting operating conditions were used for all simulations. The simulations investigated the sensitivity of various parameters in the surge control systems, such as recycle valve opening times, recycle valve sizes, delay of opening of recycle valves, and the effects of the opening time of the blowdown valve. Thus, the anti-surge system control parameters were built into the simulation as well as the ESD sequences and coast down speed of the compressor. The values of control parameters such as Surge Control Line (SCL) and Boost Line (BL) were based on the existing compressor configuration.

Results from the different parametric studies indicated the most critical scenarios to be analyzed for the entire train, as well as the best direction for any proposed change or modification. Therefore, modified sequences and alternatives were simulated and analyzed for the compressor trains. The first modified sequence involved changing valve delay and opening times on the second and third stages to avoid surge; all stages avoided surge with the proposed timing. The surge avoidance on the second and third stages was marginal. A second modification in the anti-surge logic introduced a coast-down delay of the compressor unit; thus, all stages avoided surge when the compressor shutdown was delayed by 1.15 seconds. The surge avoidance on the second stage was marginal. Another option evaluated was to implement hot bypasses in some of the stages. In this option, the original valve timing was used and additional recycle valves were added in parallel to the recycle valves on the second and third stages (referred to as hot bypass valves). Several cases were presented where surge was avoided. However, in the majority of these cases the surge avoidance was considered marginal. One case was presented where the surge margin was considered acceptable. This included the use of a 6 and 3-inch "quick" opening valve on the second and third stages, respectively.

A final option blended two of the previously evaluated options in one; a compressor coast-down delay and hot bypass. This sequence took advantage of both the coast down delay and a hot bypass valve on the second stage. Surge was avoided with a coast down delay of 0.5 seconds and a second stage hot bypass valve with a flow coefficient (Cv) equivalent to a 4-inch valve. The surge with this delay and valve was considered marginal. The surge avoidance was found to be better with a coast down delay of 0.75 seconds and a second stage hot bypass valve of 4-inch as presented in Figure 14.

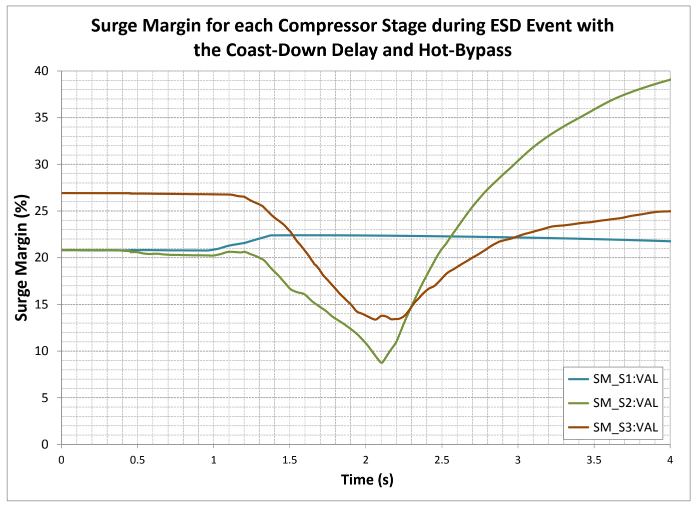


**Figure 14. Coast down Delay and Hot Bypass Option Results**

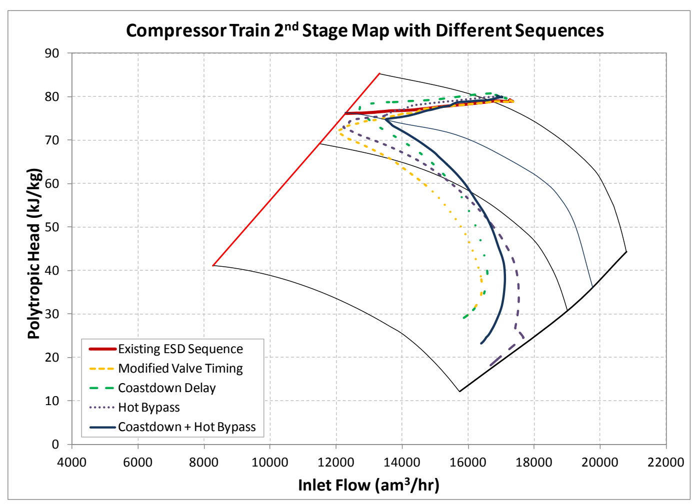


**Figure 15. 2nd Stage Compressor Map with Deceleration Paths for Different Sequences**

Figure 15 and Figure 16 show each of the five sequences discussed above on the compressor maps for the second and third stages. Based on the results of these studies, it is recommended that either the hot bypass sequence (use of 6 and 3-inch quick opening valves with 1.2 and 1.5 seconds on second and third stages, respectively) or a combination of a coast down delay and hot bypass on the second stage compressor be implemented. In addition, it was observed that the implementation of the hot bypass will affect the neighbor stages, since it will produce a quick drop of the discharge pressure of one stage as well as the suction pressure of the following stage as observed in Figure 15 where the coast down delay and the hot bypass combined effect is in the middle. In this case, the hot bypass option of the 2nd stage is diminishing the positive effect of the coast down delay.

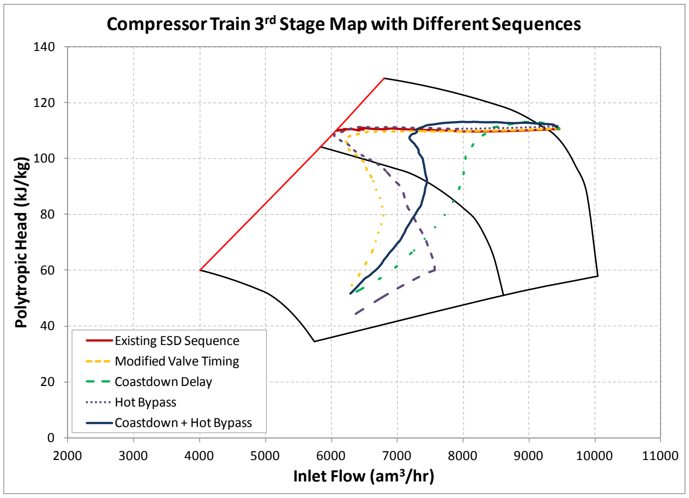


**Figure 16. 3rd Stage Compressor Map with Deceleration Paths for Different Sequences**

## CONCLUSIONS

This tutorial paper has reviewed and presented some of the most relevant parameters which should be considered when a dynamic simulation of a centrifugal compressor is performed. As a summary, five rules of thumbs are provided:

- **Rule 1- Piping Volume**: The geometry of the piping system, in particular the volumes of the piping and vessels (after-cooler and KO-drums) play an important role in predicting the impedance of the system which affects the initial dH/dQ behavior of the compressor. Special attention should be focused on the piping/vessel volumes between the suction valve and the discharge line up to the check valve as well as the recycle line.

- **Rule 2 - Recycle and Check Valves**: A precise definition of the recycle valve type and flow characteristic behavior is a requirement. The opening and closing characteristics of the valves (flow coefficients and type of recycle and check valves). Recycle valve timing (opening and any signal delay). In addition, the placement of the discharge check valve is critical for the transient predictions. Another important detail is to determine if the check valve is damped or undamped, for its closing time.

- **Rule 3 – Compressor Coast down:** An accurate prediction of the speed coast down of the machine is critical in the first few seconds of the transient analysis. For this purpose, the inertia of the compressor train or field data can be used to generate a precise coast down curve.

- **Rule 4 - The compressor performance map:** The accuracy of the compressor map will represent the most important parameter in defining the transient predictions since actual performance conditions of the machine could be shifted from the theoretical calculations. A refined compressor map and field data for validation is ideal; however, it is not always available. A comparison of the computational model and the available data will provide an excellent source for defining the model uncertainty.

- **Rule 5 - Control logic, special commands, and sequences:** Sequence of events and their timing are very important for simulating a transient event. In a compressor shutdown the first action after the machine trips (fuel is cut-off) is to open the recycle valve and then start closing the isolation or safety valves, and open the blowdown, if applied; however, all this happens in a matter of seconds. Thus, the correct order of actions, their timing, and signal delays are necessary to obtain a good prediction of the transient event. In some instances the misrepresentation of a signal delay (milliseconds) could lead to a surge condition prediction.

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