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Gas Turbine Compressor System Design Using Dynamic Process Simulation

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ABSTRACT
Almost all petrochemical processes are modeled as steady state to establish the required sizes of pipe, vessels, compressors, pumps, and valves. A dynamic simulation study of the process compressor system design can now be done as part of the initial plant design activity. The system piping design and proper placement of the anti-surge valves (ASV) along with preliminary valve sizing can be evaluated. Additional control strategies may be evaluated for emergency shut down (ESD). This will confirm the system design to safely shutdown a compressor without surge. Many of the issues that may be uncovered during study execution can save time and expense before the initial commissioning of the process. For a plant start up, the driver capability, compressor loading, positions of valves, and control philosophy may be tested before the components are committed to hardware.

INTRODUCTION
New advances in computer computation speed and improvements in simulation programs now make it possible to provide simulation modeling as part of the initial plant/process design. The requirements to accurately model the major components of the simulation are discussed in this paper. The dynamic analysis of the process can be done up front because the programs have become less complex to set up and run. Skill and knowledge are still required to properly create a simulation model, interpret the data and recognize valid results. The lead times to do such an analysis have been dramatically reduced. The system modeling may start with some basic simple assumptions. This initial model can then grow, as more details become known. Additional components may be added to the model and evaluated as an integral part of the system. Issues of startup and shut down may point to system changes that would not readily be understood from the normal steady state process flow diagrams. The DYFLO program (1) by QMC Corporation was the simulation software package used in the evaluation of several actual case studies that this paper is based on.

NOMENCLATURE
ASV Anti-surge valve
Cv Valve flow capacity
ESD Emergency shut down
H Head
HAZ-Op Hazardous Operation Review
LNG Liquefied natural gas
N Rotating speed
NGL Natural gas liquids
Overload High flow region beyond the compressor map
Q Volume flow
Surge A point where the flow within the compressor becomes unstable.

Gas Turbine Compressor Applications
- Gas pipeline applications usually have relatively constant gas composition with variations in slow changing upstream and downstream pressure variations caused by the large volume of gas.
- Gas reinjection applications can have variations of gas composition with potential for rapid changes in flow because of process upsets. There also could be slugs of liquid that cause flow disturbances in the compression system. They usually are high pressure ratio applications that have discharge gas conditions greater than 5000 PSIG.
- Gas gathering applications collect untreated gas to make it suitable for transfer to transmission pipelines. This type of compression system can have variations in gas composition and system transients caused by dehydration, filtration and sweetening system upsets. They are somewhat similar to reinjection applications. Discharge pressures are usually under 3000 PSIG.
- Processing plant applications cover a broad spectrum that could include NGL removal, LNG facilities, and refinery & hydrocarbon processing plants. The plant transients include variations in gas composition along with upstream/downstream processing upsets.

Requirements for a dynamic study
As mentioned above, the dynamic simulation requires some components to be specified or approximated. The compressor flow map is needed for estimating the performance. The volumes at the inlet and the discharge of the compressor are critical to evaluate the dynamic response of the system. It is important to note that the volumes determine the mass

contained in the process system. The compressor is used to move mass from inlet to discharge. The time to move this mass is determined by the volumes and the given flow capability of the compressor. The time it takes to fill a volume determines the rate of increase in discharge pressure (head build up). A compressor operating at full speed may operate in overload until the pressure in the discharge volume builds up. During start up, the compressor will accelerate and move gas mass from inlet to discharge. Thus there can be a significant lag in the build up of discharge pressure. However, the compressor power will follow its overload curve for flow and head internally even though the external system head may be much lower.

The calculations for a system are based on mass distribution, pressure and temperature. The compressor is operating between two volumes at different pressures. The volume head is fixed and at a given speed the compressor flow is the resultant and not the driving element. To effect a change in the flow, the pressure and/or compressor speed must be modified. Therefore, the head and speed determine the flow for a dynamic simulation.

Piping Geometry and Equipment Layout

The location of components in a compressor installation is critical to a plant design. This should be reviewed during the initial plant design to help prevent operational problems. Some of the important points to consider are:

- Large volumes located between the compressor discharge flange and the recycle valve should be avoided or minimized.
- Cooling of the recycle gas is required for operation in continuous recycle.
- Liquid separation vessels should be located upstream of compressors and downstream of cooling elements in a compressor system.
- Separate and independent recycle loops should be provided for multiple sections of compression connected in series. Placement of check valves and recycle take off and return lines is critical to achieving separate loops of compression.
- Parallel compressor units must have check valves installed to isolate each unit.

The volume of the piping, vessels and coolers determines the system response to a change. The change may be to move from one operating point to another or a system trip of the compression train. The dynamic simulation determines the mass inventory of each component at a given time. As the mass inventory changes, the specific component pressure and temperature will change.

Turbine Modeling

For a process simulation the gas turbine is simply modeled as a power producing component along with its transient response capability to load change. The elements of the gas turbine are not modeled in detail as compressor, combustor and turbine.

The knowledge of gas turbine operational sequences along with other physical characteristics of the gas turbine compression system are the essential requirements for developing an accurate model for dynamic process simulation. These characteristics include the following:

- Rotating inertia of the gas turbine component mechanically connected to the load (i.e., power wheel)
- Control system response time to a transient condition including:
    1. fuel shutoff and control valve inherent time lag and response characteristics
    2. stored energy of the fuel between the last fuel shutoff valve and turbine combustor(s)
    3. Turbine unloading through variable geometry or blowoff

These all combine to define the power decay characteristics of the rotating equipment.

Some turbine manufacturers have test stand and or field data to accurately determine response characteristics. When these are not available, assumptions have to be made which may affect accuracy of results. The customary assumption of instantaneous loss of power production in response to a transient event can lead to an over design of components with associated commercial impact. Dynamic simulation can therefore provide a tool to gauge the validity of the design.

Intentional time delay of the turbine trip has been utilized successfully to prevent compressor surge in response to the rapid deceleration. See discussion below

Other gas turbine characteristics that need to be considered for upset conditions include maximum load acceptance and rejection rates. These define the acceleration and deceleration rates for safe and continuous equipment operation. Sometimes these maximum rates may not be satisfactory for connected equipment or stable process operation. A dynamic simulation will help determine the best operational rate of change characteristics required for a specific plant equipment design.

Gas turbine start up characteristics and sequences may also impose some additional limitations. These include turbine warm up at minimal speed and power producing capabilities for heat soak requirements. These are outside the normal operating envelope of compressors and could have special requirements.

Gas turbine restart with high compressor settle out pressure conditions may impose excessive load at low speed This should be considered in equipment and start up control sequence design to prevent an undesirable operating condition. Dynamic simulation can be used to determine safe operating parameters for plant and equipment start up.

Turbine Trip Delay

Some gas turbine compressor applications can and have used a trip delay feature to prevent compressor surge during an equipment shutdown event. This delay allows anti-surge valves and optional hot gas bypass valves time to respond prior to terminating the fuel source to the gas turbine. Compressor surge on trip can cause equipment damage and should be avoided or minimized as much as possible. The use of this control strategy requires the categorization of all the turbine-compressor equipment trips into two categories. These trips can be classified as either critical requiring immediate shutdown or non-critical where a delay of 1 to 3 seconds can be tolerated. The definition of a critical shutdown is any malfunction that could pose a threat to equipment, personnel or environment. A HAZ-OP study should be conducted with participation by various parties including the owner/operator, engineering contractor, equipment suppliers and possibly consultants. This

HAZ-Op study could be used to help categorize the trips. Some examples of critical type shutdowns are over-speed, low oil pressure and fire or gas detection. Some examples of non critical shutdowns are high bearing temperature, high process scrubber level or high process gas temperature. The results of this categorization are then incorporated into the control system logic. This control strategy would not totally eliminate compressor surge during trip but rather minimize the number of shutdown events that could cause a surge on trip. A careful evaluation should be made to determine if this control strategy could be utilized on a specific application.

Compressor Modeling

The modeling of a compressor may be as simple as using classical fan laws defined by the normalized parameters of head divided by the square of the rotational speed (H/N^2) as a function of flow divided by rotational speed (Q/N) and efficiency as a function of flow divided by rotational speed (Q/N). This approach works well for single compressor stages and some multistage arrangements. However, this can lead to significant error in determining surge depending on the number of compression stages in series in the compressor body and the resulting overall compressor performance map especially below design speed.

Predicted head and flow data for multiple stages in series are derived from individual stage performance. The combined performance prediction may be accurate over a very limited range of speed range. Extrapolating this data to a larger speed range can lead to errors of 25% at overload and surge. A more accurate compressor map must be generated at multiple speeds especially for start up and shutdown studies. During an ESD the shape of the surge curve is critical. Many systems model the surge limit as a linear relation between flow and differential pressure developed across the compressor. The surge curve for each compression system is unique and may not be linear in many applications. Modeling the system must try to be as accurate as possible. The same is required for overload operation. Many compressor systems are started in overload. As multiple sections of compression are part of many processes the accuracy of the curves used modeling become even more important.

The practice of normalizing the head flow curve from the Q/N vs. Head /N^2 relationship to represent the entire curve from low to high speed will lead to modeling errors. As shown in Figure 1, the performance prediction by the compressor manufacturer's extended performance curves does not match the extrapolated Q/N based on the normal design speed curve. As can be seen the Q/N speed curves have a much different shape as the speed decreases. The change in shape will cause erroneous results for the dynamic simulation. Figure 2 shows the overlay of the surge and overload lines. The Q/N predicted overload at the end of the curve will be greater than is actually available at lower speed based on the manufacturers data. The surge points will also not be estimated correctly and could lead to inaccurate conclusions. The conclusions drawn from a system that is not modeled accurately can lead to either over design or under design of the compression system components. Over design could restrict operating range, reduce efficiency, and provide poor control system response. Under design may lead to predicting safe operation in an area of the extrapolated compressor performance map that is actually in surge or

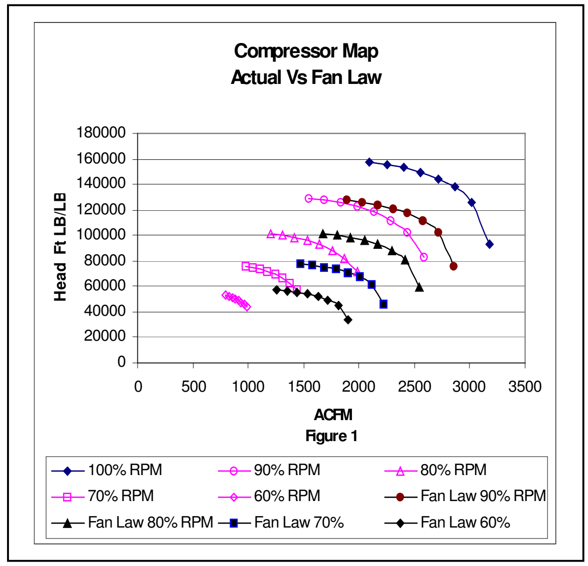

Figure 1 Comparison of fan laws to actual compressor flow Map

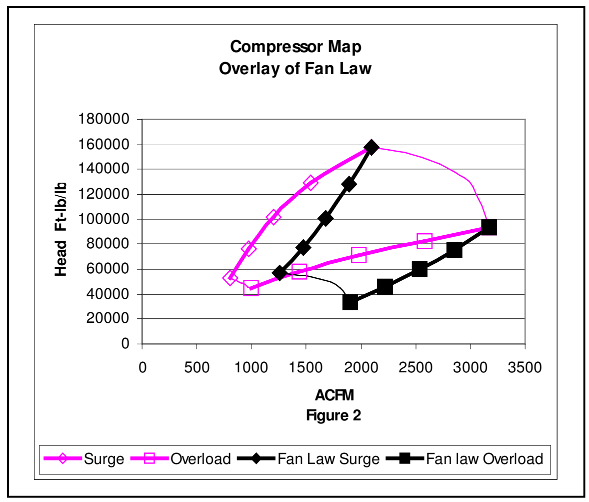

Figure 2 Comparison of fan law estimation to actual performance showing the shift of the overall compressor performance map

overload. The consequences of this could limit the equipment life and process operating range.

Continuous operation in overload is not recommended by some compressor manufactures (3). While not as detrimental as operating in surge, the mismatch between components can cause damage to the internal parts of the compressor.

Anti-Surge Valve Requirements

The proper protection of a compressor requires analysis of steady state online requirements and transient requirements of the valve capacity. Steady state requirements can be determined from compressor performance data. Traditional sizing uses an over sizing factor that multiplies surge flow by the sizing factor to determine a required Cv for the anti-surge valve. This method does not accurately predict specific compressor requirements for surge protection during a high-speed transient event such as an emergency shutdown

(ESD) of the compressor. These transient requirements are not as easily determined. The transient requirement must be able to handle the compressor flow along with rapid depressurization of the discharge volume to match the compressor head producing capability as it decelerates. Traditional valve sizing procedures may select a valve capacity that is too large to provide stable control when operating near the surge control line. Over sizing may allow the compressor to operate in overload if the valve goes to full open position during normal operation. Dynamic simulation is a predictive tool, which provides more accurate sizing to account for the transient behavior of the system. This may indicate the use of two valves that have different anti-surge protective functions. One valve would be used for traditional on line modulating anti-surge protection and the other valve would be used only for shutdown in a discrete open/close mode to handle the transient flow. Dynamic simulation may also identify other valve characteristics that can be used to optimize system response. These include valve trim characteristics and valve stroke time.

Start-up or on-line operation for extended periods on partial or full recycle may dictate a requirement for low noise trim on a modulating valve. Noise requirements versus cleanliness of gas may also necessitate special design considerations. High speed transients like a compressor ESD event are of short time duration and may not necessarily have the same noise requirement as required by continuous operation. If low noise trim is used extra consideration to system cleanliness must be required. Noise attenuation techniques used in valves can require small openings that act like a filter to collect debris from the piping system. Initial start up may warrant the use of a valve start up trim or trash screen installed upstream of the valve.

Some systems may require a hot gas bypass valve to assist in rapid depressurization of the compressor discharge volume to keep the compressor out of surge. This is accomplished by separating some of the hot discharge volume from the compression total discharge volume, i.e. installation of a bypass and check valve in the discharge piping following the compressor discharge flange. The size of the volume and the selection of the valve become a specifically tuned system, which can be optimized through dynamic simulation.

Plant Modeling and Multiple Unit Simulation

Typical gas turbine compressor applications can have multiple units operating in either a parallel or series configuration. Some applications allow switching operating configuration between series and parallel.

Transients caused by trip or start up of individual turbine compressor units can be utilized to determine effective control strategy along with determining overall plant response to upset conditions. The impact on upstream and downstream conditions external to the unit boundary will also be identified. Load sharing schemes can be analyzed for response to transients.

Tuning of control system response to imposed transient can be analyzed for providing optimized response.

Troubleshooting plant performance and operational problems can be analyzed to provide solutions that resolve or mitigate the effect of an undesirable operating condition.

Real time equipment performance evaluation can be embedded within a control system to fine tune control system response and load sharing which optimize throughput and plant efficiency.

The battery limits of a dynamic study must be clearly established to meet the objectives defined for the study. The specification of the known parameters and their expected variations need to be clearly spelled out to the simulation models. The model can be simplified by using lumped volume parameters to model piping and vessels. The model may also be complex to include modeling of 2-phase flow in the piping and specific equipment such as separators.

Evaluation of Settleout

Settleout of the system is usually determined by evaluating the mass and pressure for given volumes and assuming the mass contained in the total volume and estimating the pressure. The temperature does not solve as easily. It must be noted that the settle out of a system is a dynamic process. Gas is in circulation through the system as the compressor is slowing in speed. The rate of gas flow through the cooler in the system is decreasing. The temperature rise developed across the compressor is also decreasing. Since this change is occurring in a matter of seconds the cooler thermal controls do not normally react to match the rapid decrease in flow. The flow exiting the cooler will, therefore, approaches the coolant temperature. The average temperature at initial settleout will then be closer to the coolant temperature for the above reasons.

Example 1: ESD Prediction

An ESD requires the compressor to shut down with the power cut off. A single compressor process simulation model generated with the DYFLO program is shown in Figure 3. The elements of this model are essentially the compressor and inlet and discharge process components.

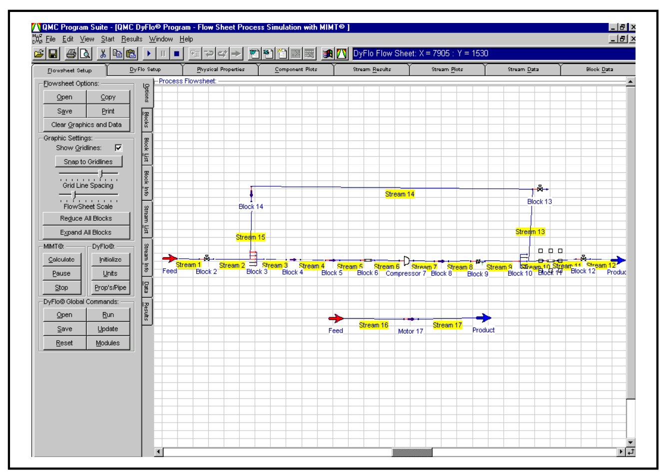

Figure 3 Example of DYFLO flow sheet model.

The simulation of an equipment train ESD event requires the depiction of the rotating components and the interaction with the total modeled system. The rotor speed decays as a function of the residual power absorption in the compressor, the inertia (WR^2) of the rotor system and the driver power decay. As the compressor decelerates, the pressure in the discharge volume and the pressure in the inlet volume determine the head. The rate at which the discharge pressure decreases and the head developed by the rotating rotor will determine if surge will occur. If the rotor developed head is less than the system volume head, the discharge mass attempts to back flow through the compressor creating the instability called surge.

The mass of gas contained in the discharge must be reduced so that the discharge pressure decays. The ASV is commanded open as fast as possible to reduce the discharge mass which will lower the discharge pressure. Figure 4 depicts the effects of an ESD event has on a compressor. Shown is the

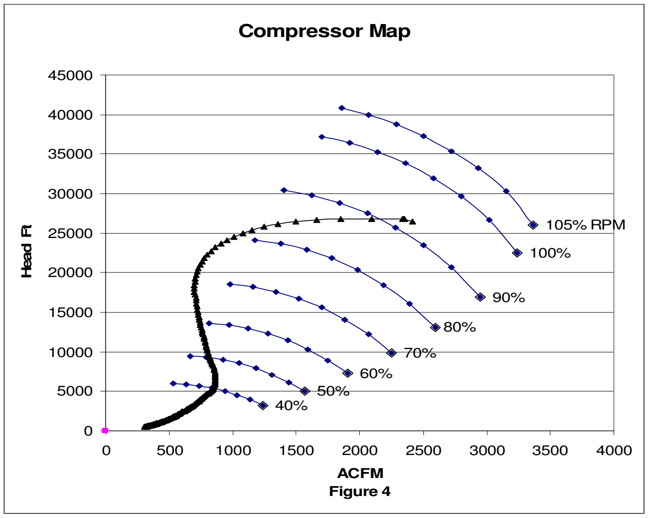

Figure 4 Compressor Flow Head map with operating path in response to an ESD event, surge is encountered.

path of the compressor operation as a function of volume flow versus head. This is superimposed on the compressor map with lines of constant speed. The surge limit is the left most point on each speed line. As can be seen the path of the deceleration crosses the surge limit line. Figure 5 shows the same flow data on a time line along with the surge flow limit line. The compressor flow drops below the surge limit line into surge in about one second and does not recover until six seconds into

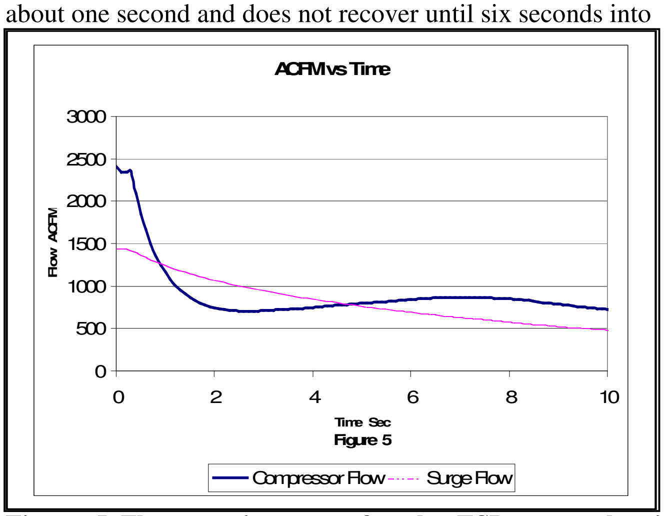

Figure 5 Flow vs. time map for the ESD event showing surge limit line and compressor operating path during ESD

the event. A larger anti-surge valve capacity will be required to prevent this excursion into surge. An 80% larger ASV was selected after a number of iterations. This resulted in the compressor staying out of the surge region during the deceleration. Figures 6 and 7 depict the same compressor system response with the larger valve. These are the results of an actual case study.

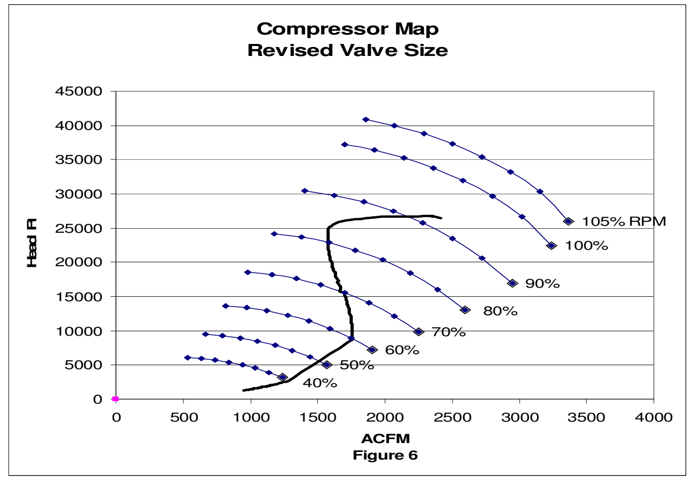

Figure 6 Compressor Flow Head map with operating path in response to an ESD event, no surge is encountered

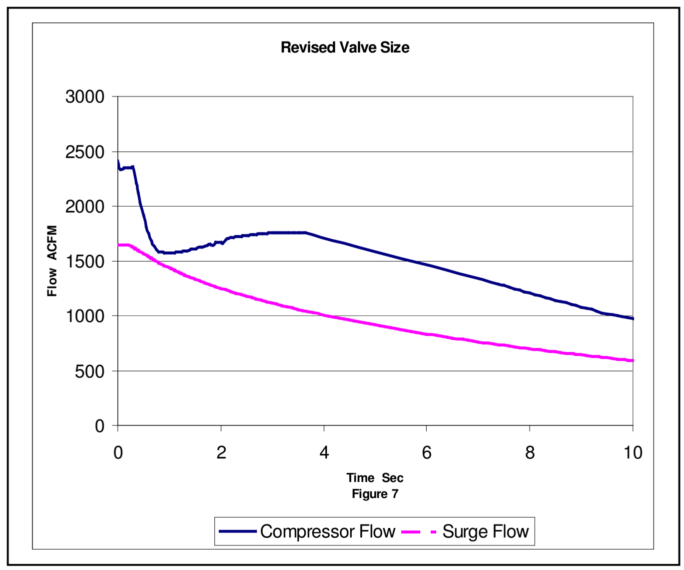

Figure 7 Flow vs. time map for the ESD event showing surge limit line and compressor operating path during ESD

The total deceleration of the rotating components is shown in Figure 8.

The estimation of the required size of the valve to avoid surge and prevent possible equipment damage during an ESD event can be accurately predicted though the use of dynamic simulation.

Example 2
Sudden discharge blockage with ASV system for pressure Control

A Sudden blockage of the discharge side of a compression system can be controlled by several methods. The anti-surge system could open the recycle valve when it senses a reduction of inlet flow. This causes the compressor operating point to move away from surge hence reducing discharge pressure. The control system can also be configured with a discharge pressure override control. This system will sense a

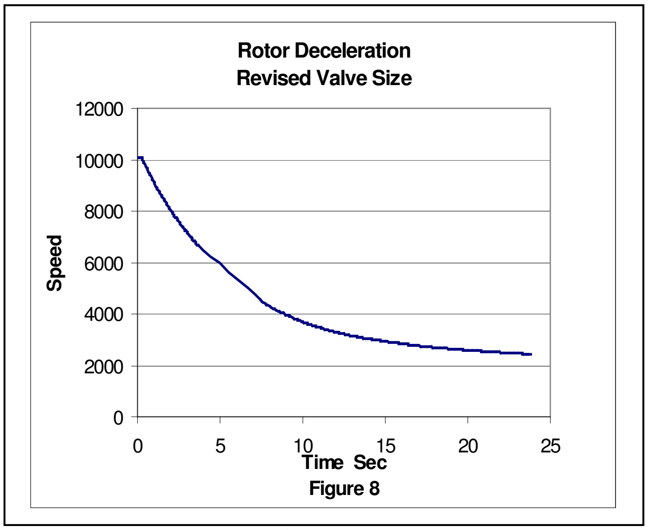

Figure 8 Rotor Speed vs. time response to an ESD.

rise in discharge pressure and open the recycle valve to control the discharge pressure. A variable speed gas turbine driver may also be used to control the discharge pressure by reducing the operating speed. Control system algorithms can be implemented to provide a coordinated response in multiple section compression systems. The control system algorithm can be a simple Proportional Integral Derivative control loop or more sophisticated to include control action based on rate of pressure change, adaptive gain or other parameters as deemed necessary.

A sudden blockage of the discharge system event is depicted in Figures 9 and 10. Figure 9 shows pressure versus time response for pressure control valve to operate. There is a

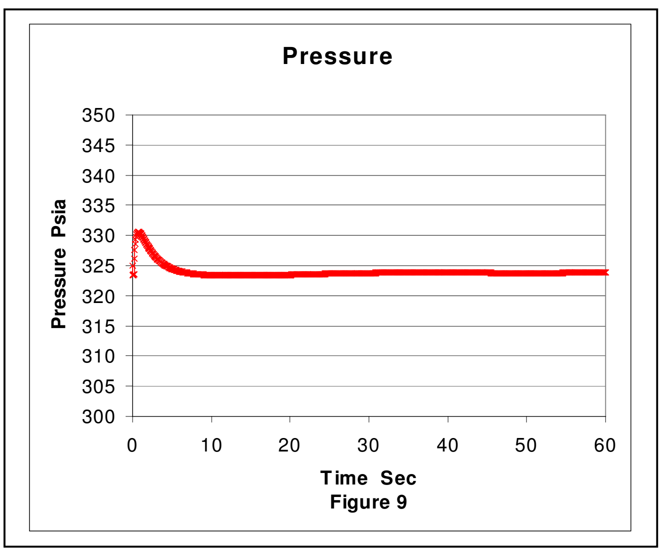

Figure 9 Pressure vs. time response to a discharge blockage

small increase in pressure and recovery to set point in 10 seconds. Figure 10 shows the reduction in flow to a minimum value to maintain the discharge pressure for this event. While the pressure recovery was quick (10 seconds) the time required for the flow to stabilize took longer due to the system volume.

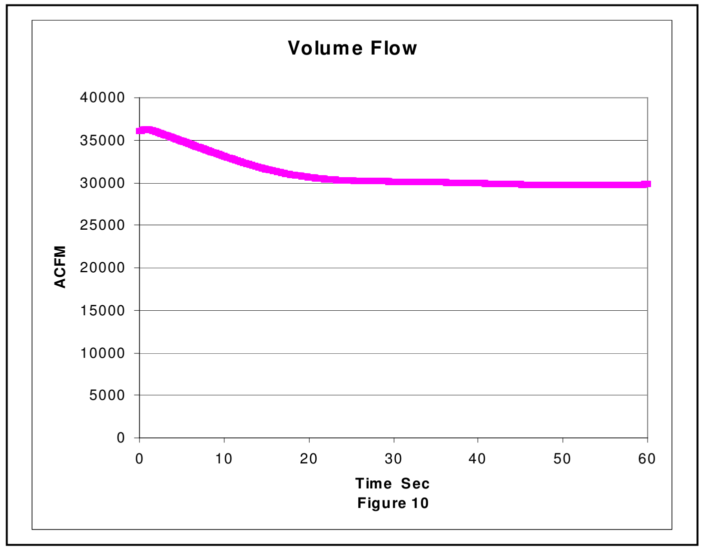

Figure 10 Flow vs. time response to a discharge blockage

Example 3 Sudden discharge blockage for relief valve sizing

A sudden blockage on the discharge of a compression system is an event that can cause the compressor operating point to move towards surge and result in a rise of the discharge pressure. If the rise in pressure is sufficiently fast or there is not proper pressure control then the discharge pressure could approach design limits where a pressure relief valve would operate to control the pressure. Proper sizing of the relief valve would prevent pressure from exceeding equipment design limits. An undersized relief valve will not protect the system to defined pressure limits. Over sizing of a relief valve may provide over pressure protection but could also cause rapid swings in pressure that may be detrimental to equipment and cause disturbances to the connected process external to the compressor. Dynamic Simulation allows verification of sizing to confirm peak pressures do not exceed design limitations. Figures 11 and 12 show an under sized relief valve response.

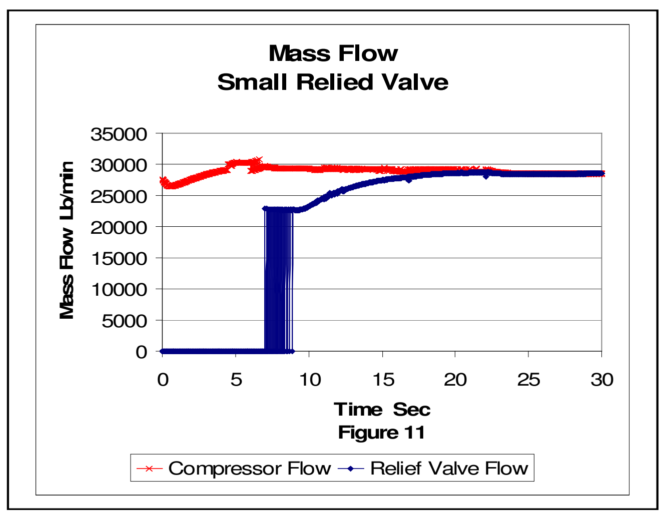

Figure 11 Flow vs. time for undersized relief valve

The valve goes through a number of open close cycles due to the volume of the compressor discharge and the flow supplied by the compressor. The flow through the relief valve, when open, may allow the pressure in the discharge volume to

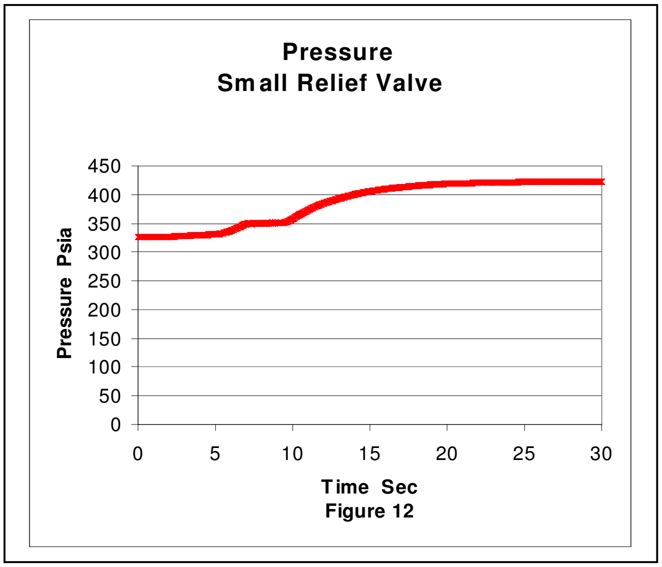

Figure 12 Pressure vs. time for an undersized relief valve

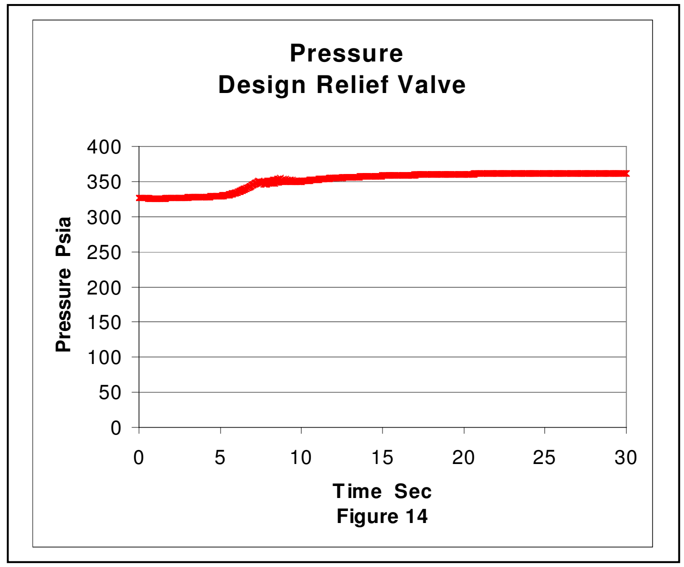

Figure 14 Pressure vs. time design relief valve.

decrease and thus the valve will close. This is purely a mass imbalance relationship. The control limit is 360 psia however the relief settle out was over 422 psia. The 422 psia was the pressure where the compressor flow and the relief valve flow matched. Figure 13 and 14 demonstrate the response of the properly sized relief valve. Once full open the valve allowed the pressure to reach 362 psia.

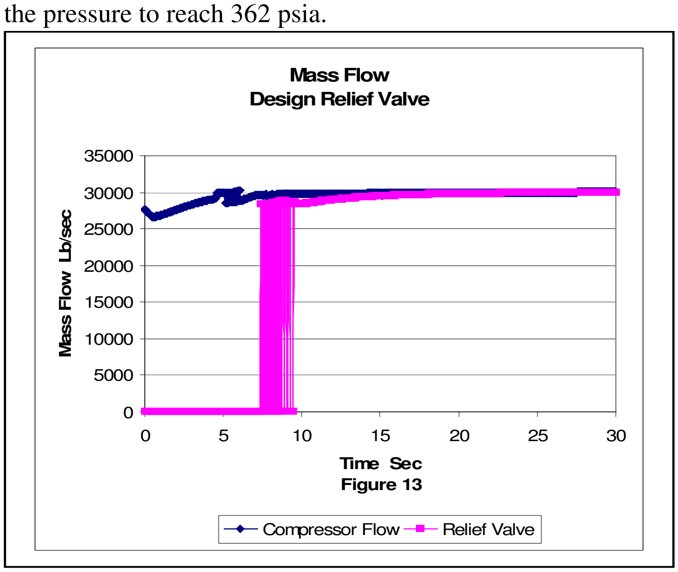

Figure 13 Flow vs. time design relief valve.

Conclusion

Dynamic process simulation is a valuable tool that can be used to validate process design to ensure trouble free operation from initial start up. It can also assist in the resolution of existing plant operational problems. This tool has evolved from a complex computer program, involving many hours to set up and run, to the point where it can now be effectively used as part of initial design of petrochemical processes.

Appendix

Simulation Requirements

1. Process and Instrument Diagram
2. Process Flow Diagram
3. Gas analysis
4. Volumes of piping , coolers, and vessels
5. Control valve design including Cv, and stroke time
6. Block valve design including Cv and stroke time
7. Compressor map for flow, pressure , temperature and speed
8. Design operating point for compressor system
9. Driver characteristics, power, speed, rate of change, startup speed torque data
10. Inertia of the rotating shaft system
11. Specific conditions at process boundaries
12. Definition of relief valves

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