

SURGE AVOIDANCE FOR COMPRESSOR SYSTEMS

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ABSTRACT

This tutorial explains the surge phenomenon as well as the precursors of surge and the damage that surge can cause to the equipment. Then, methods to keep the compressor system from surge, i.e., modern surge control systems, are discussed. This includes the necessary instruments, algorithms used, as well as the piping layout required. Sizing considerations for valves and other system components, in particular methods to correctly estimate allowable upstream pipe volumes, are described. Surge control systems for single body and multibody or multisection compressors will be explained, and attention will be given to the integration of the surge control system with other aspects of the station and unit control. In a look forward, new methods for surge detection and detection of system changes prior to surge are covered.

INTRODUCTION

Operation of centrifugal gas compressors can be defined by three operating parameters: speed, head, and flow. Centrifugal compressors have a maximum head that can be achieved at a given speed. At that peak head there is a corresponding flow. This is a stability limit. Operation of the compressor is stable provided the head is lower (less resistance in series with the compressor) and the flow is greater than these values. That is, the system is stable, as long as reductions in head result in increases in flow. Surge occurs when the peak head capability of a compressor is reached and flow is further reduced. Depending on the dynamic behavior of the compression system, system surge can occur at somewhat higher or, seldom, lower flows than the peak head capability. This is a particular issue in systems with low frequency pulsations (Kurz, et al., 2006). When the compressor can no longer meet the head imposed by the suction and discharge condition (which are imposed by the compression system), flow reverses.

When a compressor approaches its surge limit, some of its components (diffusers, impeller) may start to operate in stall. Stall occurs when the gas flow starts to separate from a flow surface (Figure 1). Changing the operating point of a compressor always involves a change in incidence angles for the aerodynamic components. Just as with an airfoil (Figure 1), increasing the incidence angle will eventually lead to stall. Stall in turbomachines often appears as rotating stall, when localized regions of separated flow move along the diffuser at speeds below the rotational speed of the impeller (Day, 1991). Surge is the ultimate result of system instability.

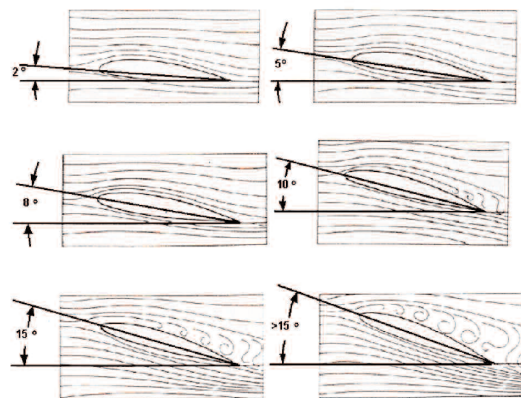


Figure 1. Progression of Stall. (NACA4412 airfoil at increasing angle of attack, based on data by Nakayama, 1988)

Attempting to increase the resistance in series with the compressor beyond that which drives the compressor to peak head will result in unstable operation. Flow will decrease and subsequently the head capability of the compressor will also decrease. As the head capability decreases, flow further decreases. Once the compressor can no longer meet the external head, flow reverses.

Surge is what happens after the stability limit of the compression system is passed.

Not only is this detrimental to meeting the process objectives, the resulting axial and radial movement of the rotor can cause damage, sometimes severe, to the compressor. Surge can be avoided by ensuring the flow through the compressor is not reduced below the flow at peak head.

The surge avoidance system prevents surge by modulating a surge control (bypass) valve around the compressor. A typical system consists of pressure and temperature transmitters on the compressor suction and discharge lines, a flow differential pressure transmitter across the compressor flowmeter, an algorithm in the control system, and a surge control valve with corresponding accessories.

A surge avoidance system determines the compressor operating point using the pressure, temperature, and flow data provided by the instrumentation. The system compares the compressor operating point to the compressor's surge limit. The difference between the operating point and the surge limit is the control error. A control algorithm (P+I+D) acts upon this difference, or "error," to develop a control signal to the recycle valve. When opened, a portion of the gas from the discharge side of the compressor is routed back to the suction side, and head across the compressor is prevented from increasing further. When the operating point reflects more flow than the required protection margin flow, the surge control valve moves toward the closed position and the compressor resumes normal operation.

There are five essentials for successful surge avoidance:

1. *A precise surge limit model*—It must predict the surge limit over the applicable range of gas conditions and characteristics.
2. *An appropriate control algorithm*—It must ensure surge avoidance without unnecessarily upsetting the process.
3. *The right instrumentation*—Instruments must be selected to meet the requirements for speed, range, and accuracy.
4. *Recycle valve correctly selected for the compressor*—Valves must fit the compressor. They must be capable of large and rapid, as well small and slow, changes in capacity.
5. *Recycle valve correctly selected for the system volumes*—The valve must be fast enough and large enough to ensure the surge limit is not reached during a shutdown. The piping system is the dominant factor in the overall system response. It must be analyzed and understood. Large volumes will preclude the implementation of a single valve surge avoidance system.

BACKGROUND

The first turbocompressors were manufactured at the turn of the 1900s. They were originally developed by steam turbine manufacturers and were widely used for ventilation purposes in deep shaft mining, in particular the coal industry. At that time, the method of producing an impeller relied upon fabrication. It would be decades before technology would allow highly efficient turbocompressors to be made. It was not until late in World War II that sufficient money was pumped into technology to allow the production of efficient high-speed compressors. In 1947 to 1948, Ingersoll-Rand and Clark designed the first centrifugal compressors for natural gas transmission. In September 1952, El Paso Natural Gas became the first company to use large gas-turbine-driven centrifugal compressors for natural gas transmission.

The pressure ratio across an early riveted impeller would have only been on the order of 1.2:1. This would have meant that to

reach a final working pressure of 7 barg, a turbocompressor needed as many as 10 or 11 stages. A single modern impeller can produce a pressure ratio as high as 8:1.

As the performance of compressors increased, so did the potential for damage from surge. Not only the damage to compressors needed to be avoided but avoiding upsets to the process became essential. To address these ever growing needs, surge avoidance systems evolved.

Early surge controls were pneumatic. They monitored the pressure differential (DP) across the compressor versus the DP across a flow measuring element through a balance beam that controlled the pneumatic signal to a recycle valve. With higher performance, higher stressed compressors, and more challenging applications, better surge controls were required.

Initially electronic surge controls were models of the pneumatic ones. They were faster, less complicated, more reliable, and required less maintenance than their pneumatic predecessors. With the advent of the microprocessor, surge avoidance systems became more algorithmically intense, surge limits were modeled via polynomials, and asymmetrical control schemes came into use. Modern surge controls ensure surge avoidance while maximizing the operating range of the compressor.

THE SURGE LIMIT MODEL

In order to avoid surge it must be known where the compressor will surge. The more accurately this is predicted, the more of the compressor's operating range that is available to the user (Figure 2). A compressor's operation is defined by three parameters: head, flow, and speed. The relationship between the compressor's operating point and surge can be defined by any two of the three.

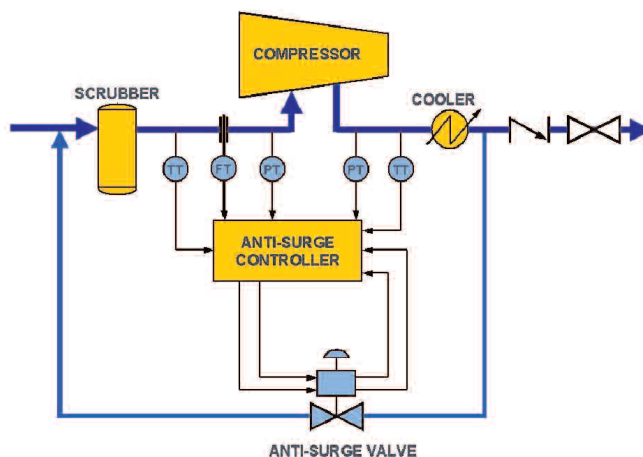


Figure 2. Surge Avoidance System Schematic.

The first two models on the left of Figure 3 involve speed. The speed of the compressor at an operating condition is strongly influenced by changes in gas composition, because the machine Mach number will change. The head versus flow relationship on the right provides a means for modeling the surge limit without being affected by gas conditions or characteristics. The parameters of the surge limit model on the right can be measured in terms of head across the compressor and head across the flowmeter.

$$H_p = K \cdot \frac{\left(\frac{P_d}{P_s}\right)^\sigma - 1}{\sigma} \cdot T \cdot SG \cdot Z \quad (1)$$

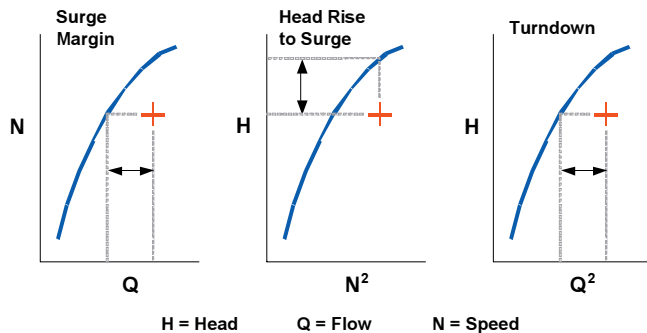


Figure 3. Surge Limit in Different Systems of Reference.

This is the basic equation for head. In the head across the compressor and the head across the flowmeter monitoring the flow through the compressor, there are common terms. These common terms (units, gas temperature, specific gravity, and compressibility) are equal in both equations and can be cancelled. This results in a simplified model that is referred to as reduced head versus reduced flow.

$$H_{REDUCED} = \frac{\left(\frac{P_D}{P_S}\right)^\sigma - 1}{\sigma} \quad (2)$$

THE CONTROL ALGORITHM

A surge avoidance control needs to be able to react appropriately to changes in power or the process. There are two very different situations that the system must respond to.

If the operating point slowly crosses the protection line, that is, at the same rate it has been moving left for the past several hours, opening of the recycle valve should be small and slow. The interdiction of the surge avoidance control should be unnoticeable. It should be as though the compressor had infinite turndown.

Conversely, if the operating point races across the compressor map, the recycle valve should begin opening before the operating point crosses the protection line. Reaction of the control should be aggressive to protect the compressor. In this case we are less concerned about the process, as it has already been impacted.

A sudden change in the system produces a control response. This is a standard control test. Ping it and see how it rings. Figure 4 reflects reactions of variously tuned controls. Low gains produce a slow response. A critically damped control produces an aggressive response but settles down quickly. If the gains are too high the system will oscillate.

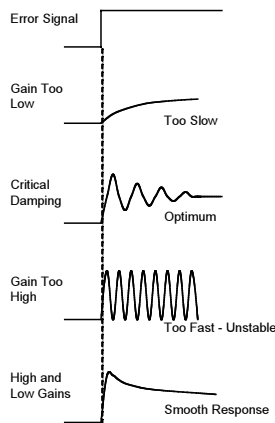


Figure 4. Reaction of a Control System to an Error Signal.

What does a surge avoidance system do most of the time? Hopefully nothing! Then, with very little margin it must act aggressively, probably requiring gains higher than could be maintained stable, to protect the compressor. To avoid instability the gains are reduced to close the valve. Once surge has been avoided, the control system should bring the process back online slowly and smoothly to avoid further upsets.

The need for extremely high gains is driven by the following: surge avoidance systems are normally built up out of commonly available process control plant components. As such these components are designed for ruggedness, reliability, and low maintenance. In general they are not focused on speed of data acquisition. Information about changing process conditions is often 1/10 of a second old. As will be seen in later sections significant advances in surge control valve action have been made recently. However, the response of the valve is typically the dominant lag in the system.

INSTRUMENTATION

To avoid surge, the control needs to know where the compressor is operating in relation to surge *in real time*. Again, how close the protection margin can be placed to surge depends on how accurately and how quickly the change in flow is reported to the control. Correctly selected instrumentation is essential. The system must have accurate measurements of the suction and discharge pressures and temperatures, and the rate of flow. Flow is the most important parameter as it will move the fastest and farthest as the surge limit is approached. Ideally, the flow transmitter should be an order of magnitude faster than the process. Unfortunately, compared to pressure and temperature transmitters, flow transmitters tend to be slow. Even the best surge avoidance control will allow a compressor to surge, if it is connected to a slow transmitter.

Flow-Measuring Devices

Most commercially available flow-measuring devices are accurate enough for surge avoidance; however, it is the transmitter that slows things down. A differential pressure transmitter's response time is inversely proportional to its range; thus, the stronger the signal, the faster the response.

Devices that develop high DP signals are desirable. Those with low signal levels tend to have low signal-to-noise ratios. Transmitters for low DP signal ranges typically have slow response times. Devices that create an abrupt restriction or expansion to the gas, such as orifices, cause turbulence and, subsequently, create noise.

It is preferable to place the flow-measuring device on the suction side of the compressor. Typically, variations in pressures, temperatures, and turbulence of the gas are less upstream of the compressor. Also, the device must be inside the innermost recycle loop (refer to Figure 2).

At a minimum, failure of the device will cause the compressor set to be shut down until the device can be replaced. If the failure results in pieces being ingested by the compressor, it can cause an expensive overhaul. For this reason, devices that are cantilevered into the gas stream are not recommended. Low cost flow-measuring devices do not always result in cost savings in the long run.

Low permanent pressure loss (PPL) devices are often recommended; however, their benefits may be marginal. The lost power cost impact of operating a device can be calculated. For example, a flowmeter developing a 100 inch H₂O signal and a 50 percent PPL flowing 100 mmscfd (50 lb/sec) is equivalent to about 20 hp.

As noted, strong signal devices are highly preferred. Pitot types (annubars and verabars) have a relatively low signal level, around 25 inches H₂O. In the middle are orifices and venturis with a moderate signal of around 100 inches H₂O. Compressor suction-to-eye provides a strong signal (around 700 inches H₂O) with the added benefit of not causing any additional pressure loss.

Suction-to-Eye Flow Measuring

Suction-to-eye uses the inlet shroud or inlet volute of the compressor as a flow-measuring device. This feature is now available on many compressors. The design requirements for the inlet volute and the flow measuring device have several things in common. Performance of the first stage impeller and the device is dependent on the uniform direction and velocity of the flow presented to it.

Critical to the operation of suction-to-eye flow measurement is the placement of the eye port. As the impeller approaches surge an area of recirculation begins to develop at the outer perimeter of the inlet to the impeller. If the eye port is placed too close to the impeller's outer perimeter the relationship of the DP to flow will be affected. Fortunately the meter factor (C') typically remains nearly the same for the same surge margin. Hence, selecting the meter factor at the desired surge protection margin will contribute to effective surge avoidance.

In a typical pipeline application (600 psi suction pressure) suction-to-eye will develop 25 psid (692 inches H2O). This is nearly seven times the differential of an orifice plate. Typically the signal to noise ratio is low and there is no additional permanent pressure loss. For surge avoidance the suction-to-eye method is strongly recommended.

Compressor Instrumentation

Optimal performance of any control system is dependent on the speed, accuracy, and resolution of the instrumented process conditions. To achieve optimal performance the instruments should have performance specifications an order of magnitude better than the requirements for the system. Typical gas compressor systems have a first-time constant of about one second; hence, no instrument should have a first-time constant of greater than 100 ms. The surge control system is expected to discriminate between single digit percentages of surge margin; hence, measurement of the process parameters should be accurate to 0.1 percent. The final control elements (recycle valves) probably can resolve 1 percent changes in their command signals; hence, the process variables should be resolved to at least 0.1 percent (10 bits) of their normal operating range. Over-ranging transmitters degrade resolution.

THE SURGE CONTROL VALVE

Characteristics

Earlier it was discussed how the control should react differently to gradual and rapid approaches to surge. Likewise, the valve must address these two very different requirements. For the gradual approach, it should behave like a small valve and produce smooth throttling. For the rapid approach case, it should act like a large fast valve to handle sudden major changes.

There are three general valve characteristics (Figure 5): quick opening, where most of the valve's capacity is reached early in its travel; linear, where capacity is equal to travel; and equal percentage, where most of the capacity is made available toward the end of the valve's travel. All three types of valve have been used in various configurations as recycle valves.

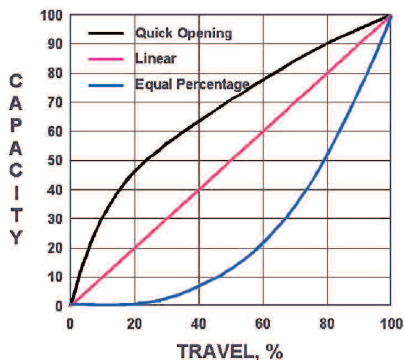


Figure 5. Valve Types.

Equal percentage valves, and in particular noise-attenuating ball valves, are recommended for surge avoidance systems with a single surge control valve. They perform like smaller valves when nearly closed and bigger valves when close to fully open. Figure 6 is a comparison of two types of equal percentage valve. For a given valve size, the noise-attenuating ball valve is often twice the cost of the globe valve, but it provides approximately three times the flow coefficient (Cv) or capacity. Also, it is more reliable as it is less susceptible to fouling and improper maintenance.

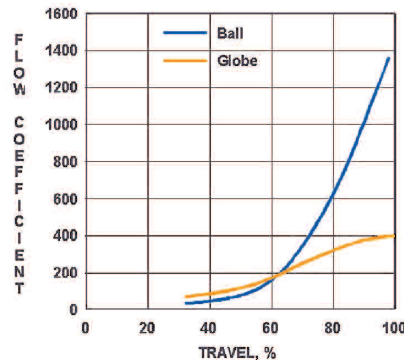


Figure 6. Ball and Globe Valves Compared.

Employing a valve with an equal percentage characteristic may provide the capacity needed to avoid surge during a shutdown while maintaining enough resolution at less than 50 percent capacity to provide good control at partial recycle. With an equal percentage characteristic the valve typically has greater resolution than a single linear valve selected to fit the compressor.

Multiple Valves

If the volumes on either side of the compressor are large, a multiple valve approach may be needed. If an integrated approach is used, the total valve capacity will be reduced.

Probably the most common is the hot and cold recycle configuration (Figure 7). Usually the cooled (outer) valve is modulating and the hot (inner) valve is a quick opening on-off type. Generally the two valves are sized independently. If the cooled valve has a solenoid, its capacity can be considered with that of the shutdown valve; subsequently the shutdown valve can be smaller.

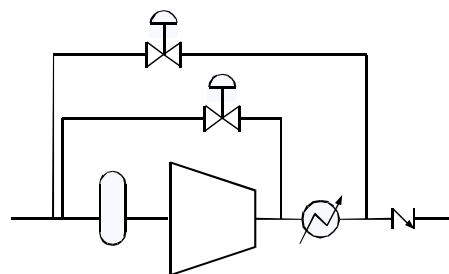


Figure 7. Hot and Cold Recycle Valve Arrangement.

An alternate to this configuration is having a second cooled valve in parallel with the first (Figure 8). This arrangement provides some measure of redundancy. In the control the two valves are operated in cascade. That is, they have different set points, say 9 percent and 10 percent surge margin. Under normal movements of the operating conditions only the 10 percent surge margin valve (primary valve) will open. If movement is fast enough to push the operating point down to 9 percent, the second valve (secondary valve) will open. If the primary valve becomes fouled and no longer positions properly, the control can place it in

the secondary position and the secondary becomes the primary valve. This change can be made without taking the compressor offline.

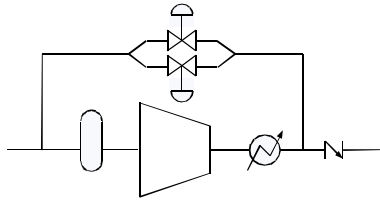


Figure 8. Parallel Recycle Valves.

The advantages of the two parallel valves do not come without a price. In normal operation 2 percent to 5 percent of the pressure rise across the compressor will be lost across the cooler. In the shutdown scenario the required flow through the cooler to avoid surge may be two or three times the normal flow. This will result in four to nine times the pressure drop across the cooler. This additional pressure drop may increase the needed recycle valve capacity significantly.

Recycle valves need to be fast and accurately positionable. They also need to be properly sized for both the compressor and the piping system. A valve well suited for modulating recycle around the compressor may not be suitable for a shutdown. (Refer to the "REVIEW OF SYSTEM VOLUMES" section below.)

For some two-valve applications, single purpose valves may be suitable, one for controlled recycling and one for shutdown. A linear characteristic valve is appropriate for the controlled recycling and a quick opening characteristic globe or ball valve for shutdown.

For the applications where the compressor speed lines are fairly flat (little increase in head for a decrease in flow) from the design conditions to surge, extra fast depressurization may be required. To achieve this, two quick opening valves may be employed. In this case a single 6 inch linear characteristic valve is replaced by two 4 inch quick opening valves. The two 4 inch valves should have slightly less flow capacity (C_v) but they will open nearly 45 milliseconds faster. For linear valves 50 percent travel equals 50 percent capacity. For quick opening valves, capacity approximately equals the square root of travel. As such the two 4 inch valves will have 70.7 percent of their fully open capacity at 50 percent open. Comparing the two arrangements 250 ms after the shutdown is initiated, the two 4 inch quick opening valves will have 56 percent more flow capacity than the single 6 inch linear valve.

For throttling, the valves are operated in cascade or split range. For most controlled recycling only one valve is opened. Although the valves have a quick opening characteristic the valves are smaller thus the capacity per percent travel is less. The two quick opening valves operated in cascade or split range will have the same C_v as the 6 inch linear at 25 percent travel.

Valve Actuation

As previously discussed, there are two operational scenarios for the surge avoidance system: modulating (minimum flow control) and rapid depressurization for shutdown. By inserting a three-way solenoid valve into the positioner's output, the valve can be made to open with either a proportional (4-20 mA) signal for modulating control, or a discrete (24 VDC) signal for total fast opening.

The primary difference between a surge control valve and a standard control valve is in its actuation system. The preferred actuator for surge avoidance is spring return, fail open. This design is simple, reliable, and ensures the compressor is protected in the event of a power failure. Both spring and

diaphragm and spring and piston actuators are used. The spring and diaphragm actuator is most commonly used on globe valves. The spring and piston actuator is more often used on ball valves. The more powerful spring and piston actuators are required on rotary valves due to the greater forces required to accelerate the mass of the ball. Some ball valves are not suitable for surge control applications because their shafts and attachments to the ball are not strong enough to transmit the torque required to open these valves at the required speeds.

Surge control valves need to be able to open very quickly. As such their actuators will have strong springs, very large air passages, and shock absorbers at their end of travel. This must be considered when sourcing recycle valves for surge avoidance.

The accessory unique to a sound surge control valve assembly is the single sided booster or exhaust booster. This is essentially a differential pressure relief device. Opening the booster vents the actuator pressure to atmosphere. The threshold for opening is about 0.5 psid. There is a small restriction (needle valve) between the control pressure from the positioner via the three-way solenoid valve and the top of the booster. Small slow reductions in pressure (opening the valve) do not cause the booster to open. Large fast reductions in pressure developing more than 0.5 psid across the restriction, cause the booster to open. If the solenoid valve is de-energized, the top of the booster is vented to atmosphere and the booster fully opens.

Standard industry quick-exhausts are not recommended for this application. They have a high threshold for opening (typically 2 to 4 psid) and an equally high threshold for reclosing. Although they may work well for fully opening the valve, they will not work well with the positioner.

Positioners should be selected for high capacity and quick response to changes in their control signals. Most of the major valve manufacturers have released second and third generation smart positioners that are suitably fast for this application.

Figures 9 and 10 show globe and ball valves with their preferred instrumentation configurations.

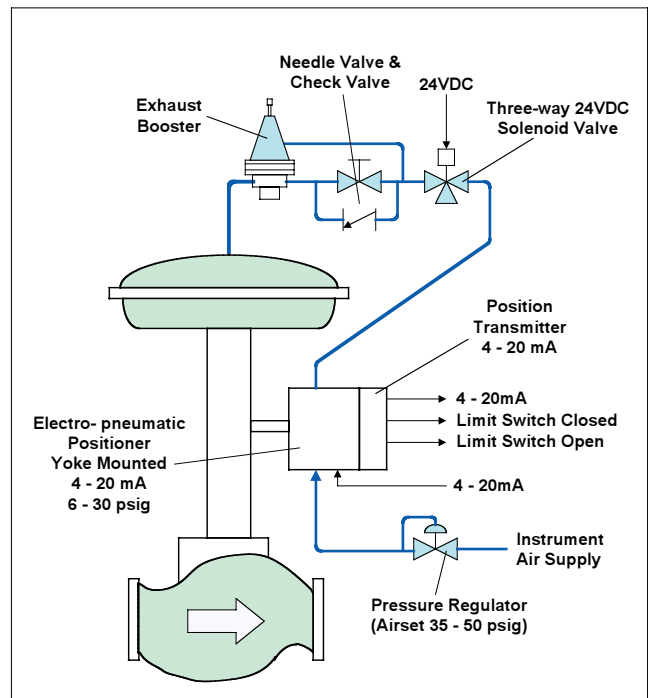


Figure 9. Globe Valve Assembly.

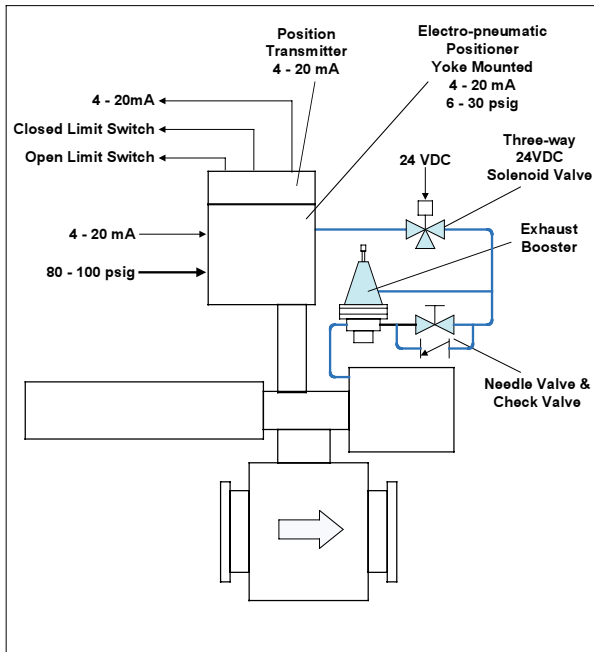


Figure 10. Ball Valve Assembly.

Recycle Valve Sizing Tool

The recycle valve needs to be sized based on the expected operating conditions of the compressor. A valve-sizing program can facilitate matching a recycle valve to a compressor. The compressor data are entered into the tool in its normal form (pressures, temperatures, heads, speeds, and flows). Various operating conditions for a specific application are then entered, such as the minimum and maximum operating speeds, pipe operating pressures, temperatures, relief valve settings, and cooler data, if applicable. The tool calculates the equivalent valve capacities or Cvs from that data.

Typically the surge limit of a compressor equates to a single valve capacity or Cv (Figure 11). The valve can be selected based on valve Cg, Cv, and Xt tables from surge control valve suppliers. As previously described, a single surge control valve application will have an equal percentage characteristic. Once a valve is selected several performance lines of a specific opening can be developed and overlaid on the compressor map. The equal percentage characteristic valve should be at about two-thirds travel at the surge conditions. The valve evaluation in Figure 12 shows such a valve with its flow characteristic when 60 percent, 70 percent, and 100 percent open, superimposed on the compressor map.

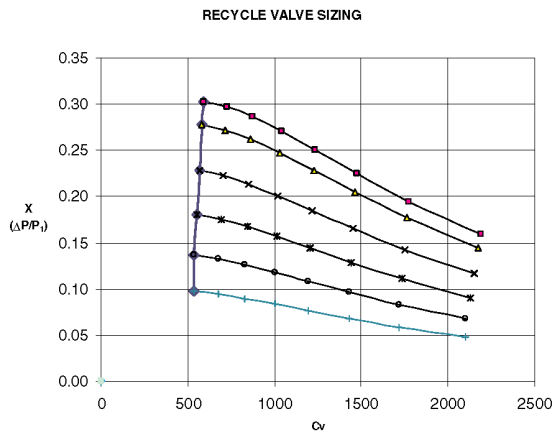


Figure 11. Almost Constant Cv at the Surge Limit.

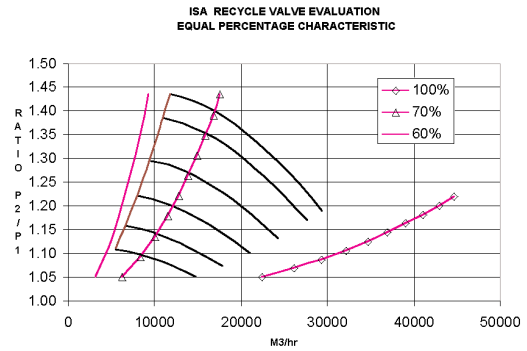


Figure 12. Valve Matched to Compressor.

REVIEW OF SYSTEM VOLUMES

Design of the piping and valves, together with the selection and the placement of instruments, will significantly affect the performance of an antisurge control system. This should be addressed during the planning stage of a project because the correction of design flaws can be very costly once the equipment is installed and in operation.

As described above, the control system monitors the compressor operating parameters, compares them to the surge limit, and opens the recycle valve as necessary to maintain the flow through the compressor at a desired margin from surge. In the event of an emergency shutdown or ESD, where the fuel to the gas turbine is shut off instantly, the surge valve opens immediately, essentially at the same time the fuel valve is closing.

The worst case scenario for a surge control system is an emergency shutdown (ESD), particularly if the compressor is already operating close to surge when the engine shutdown occurs. If an ESD is initiated, the fuel supply is shut off immediately and the compressor will decelerate rapidly under the influence of the fluid forces counteracted by the inertia of the rotor system. Figure 13, which displays data based on test data and theoretical considerations, indicates a 30 percent drop on compressor speed within the first second after shutdown. A 30 percent loss in speed equates to approximately a loss in head of about 50 percent. The valve must, therefore, reduce the head across the compressor by about half in the same time as the compressor loses 30 percent of its speed.

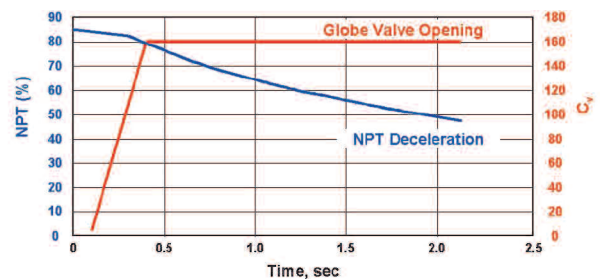


Figure 13. Simplified System and Transient Characteristic.

The larger the volumes are in the system, the longer it will take to equalize the pressures. Obviously, the larger the valve, the better its potential to avoid surge. However, the larger the valve, the poorer its controllability at partial recycle. The faster the valve can be opened, the more flow can pass through it. There are, however, limits to the valve opening speed, dictated by the need to control intermediate positions of the valve, as well as by practical limits to the power of the actuator. The situation may be improved by using a valve that is only boosted to open, thus combining high opening speed for surge avoidance with the capability to avoid oscillations by slow closing.

If the discharge volume is too large and the recycle valve cannot be designed to avoid surge, a short recycle loop (hot recycle valve) may be considered (Figure 7), where the recycle loop does not include the aftercooler.

While the behavior of the piping system can be predicted quite accurately, the question about the rate of deceleration for the compressor remains. It is possible to calculate the power consumption for a number of potential steady-state operating points. The operating points are imposed by the pressure in the discharge volume, which dictates the head of the compressor. For a given speed, this determines the flow that the compressor feeds into the discharge.

In a simple system, the boundaries for the gas volume on the discharge side are established by the discharge check valve, compressor, and recycle valve (Figure 2). The volume on the suction side is usually orders of magnitude larger than the discharge volume and, therefore, can be considered infinite. Thus, to simplify the analysis of a system, the suction pressure can be considered constant. This is not a general rule, but is used to simplify the following considerations. This yields the simplified system, consisting of a volume filled by a compressor and emptied through a valve (Figure 14).

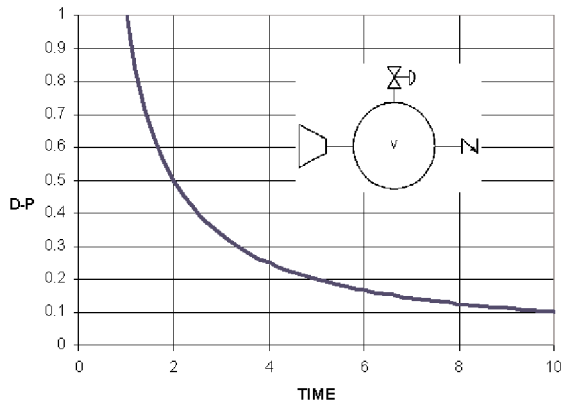


Figure 14. Train Deceleration and Valve Opening.

The basic dynamic behavior of the system is that of a fixed volume where the flow through the valve is a function of the pressure differential over the valve. In a surge avoidance system, a certain amount of the valve's flow capacity will be consumed to recycle the flow through the compressor. Only the remaining capacity is available for depressurizing the discharge volume. In such a system, mass and momentum balance have to be maintained (Sentz, 1980; Kurz and White, 2004). From this complete model, some simplifications can be derived, based on the type of questions that need to be answered.

For relatively short pipes, with limited volume (such as the systems desired for recycle lines), the pressure at the valve and the pressure at compressor discharge will not be considerably different. Also, due to the short duration of any event, the heat transfer can also be neglected. Therefore, mass and momentum conservation are reduced to:

$$\frac{dp_2}{dt} = \frac{k \cdot p_2}{V} [Q - Q_v] \quad (3)$$

The rate of flow through the valve is calculated with the standard ISA method (ANSI/ISA, 1995) (Q_{std} is the standard flow. F_p is the piping geometry factor. It is usually not known and can be assumed to be 1. The pressure is assumed to be constant in the entire pipe volume. It is thus the same just upstream of the valve and at the discharge pressure of the compressor):

$$Q_{std} = 1360 \cdot F_p \cdot c_v \cdot Y \cdot \left[\frac{p_2 - p_1}{p_2} \cdot \frac{1}{SG \cdot T_2 \cdot Z_2} \right]^{0.5} \quad (4)$$

and

$$Q_v = Q_{std} \frac{\rho_{std}}{\rho(p_2, T_2, Z_2)} \quad (5)$$

The compressibility Z_2 can be calculated with the Redlich-Kwong equation of state. Equations (3), (4), and (5) mean that the discharge pressure change depends on the capability of the valve to release flow at a higher rate than the flow coming from the compressor. It also shows that the pressure reduction for a given valve will be slower for larger pipe volumes (V). Kurz and White (2004) have shown the validity of the simplified model.

The discharge pressure p_2 in Equation (3) is a function of the compressor operating point, expressed by:

$$\frac{p_2}{p_1} = \left[1 + \frac{k-1}{k} \cdot \frac{h(Q, N) \cdot SG}{287 \cdot Z T_1} \right]^{\frac{k}{k-1}} \quad \frac{h}{N^2} = \alpha \left(\frac{Q}{N} \right)^2 + \beta \frac{Q}{N} + \gamma \quad (6)$$

Alternatively, a lookup table, showing the head-flow relationship for the compressor, can be used.

The behavior of the compressor during ESD is governed by two effects. The inertia of the system consisting of the compressor, coupling, and power turbine (and gearbox where applicable) is counteracted by the torque (T) transferred into the fluid by the compressor (mechanical losses are neglected). The balance of forces thus yields:

$$T = -2\pi \cdot J \cdot \frac{dN}{dt} \quad (7)$$

Knowing the inertia (J) of the system and measuring the speed variation with time during rundown yields the torque and, thus, the power transferred to the gas:

$$P = T \cdot N \cdot 2\pi = -(2\pi)^2 \cdot J \cdot N \cdot \frac{dN}{dt} \quad (8)$$

If the rundown would follow through similar operating points, then P~N³, which would lead to a rundown behavior of:

$$\frac{dN}{dt} = \frac{k \cdot N^2}{J (2\pi)^2} \rightarrow \int N^{-2} dN = \frac{k}{J (2\pi)^2} \int dt + c \rightarrow N(t) = \frac{1}{-\frac{k}{J (2\pi)^2} t - \frac{1}{N_{t=0}}} \quad (9)$$

Regarding the proportionality factor (k) for power and speed, this factor is fairly constant, no matter where on the operating map the rundown event starts. Thus, the rate of deceleration, which is approximately determined by the inertia and the proportionality factor, is fairly independent of the operating point of the compressor when the shutdown occurred, i.e., the time constant (dN/dt [t = 0]) for the rundown event is proportional to k/J. However, the higher the surge margin is at the moment of the trip, the more head increase can be achieved by the compressor at constant speed.

While the behavior of the piping system can thus be predicted quite accurately, the question about the rate of deceleration for the compressor remains. It is possible to calculate the power consumption for a number of potential steady-state operating points. The operating points are imposed by the pressure in the discharge volume, which dictates the head of the compressor. For a given speed, this determines the flow that the compressor feeds into the discharge.

The model described above, which accounts for the primary physical features of the discharge system, can be used to determine whether the combination of discharge volume and valve size can prevent the compressor from surge during an ESD. It allows the two important design parameters to be easily varied to avoid surge during ESD. The surge valve size and opening speed can be increased for a given discharge volume or the maximum allowable discharge volume for a given configuration of valves and compressor characteristic can be limited. The second method, which has the advantage of being more transparent for the station design, is used here.

The simplified model calculates the maximum discharge volume where the head across the compressor can be reduced by half in one second, based on the assumption that this reflects the speed decay during an ESD as outlined above. Therefore, the calculation of the instant compressor speed is replaced by a fixed, presumed to be known, deceleration rate. The assumption is made that the power turbine and compressor will lose about 20 to 30 percent speed in the first second of deceleration. This is, for example, confirmed by data from Kurz and White (2004) showing a 30 percent speed reduction of a gas turbine driven compressor set, and, where the gas turbine driven configurations lost about 20 to 25 percent speed in the first second, while the electric motor driven configuration lost 30 percent speed in the first second. As a result of the loss of 25 percent speed, the head the compressor can produce at the surge line is about 56 percent lower than at the initial speed, if the fan law is applied. A further assumption is made about the operating point to be the design point at the instant of the ESD.

Any ESD is initiated by the control system. Various delays in the system are caused by the time for the fuel valve to shut completely, the time until the hot pressurized gas supply to the power turbine seizes, and the opening time of the recycle valve. ESD data show it is a valid assumption that the surge control valve reaches full open simultaneously with the beginning of deceleration of the power turbine/compressor. This is the starting time (T_0) for the model.

Usually, the suction volume (no check valve) is more than three orders of magnitude greater than the discharge volume and is therefore considered at a constant pressure. The general idea is now to consider only the mass flow into the piping volume (from the compressor) and the mass flow leaving this volume through the recycle valve. Since the gas mass in the piping volume determines the density and, thus, the pressure in the gas, we can for any instant see whether the head required to deliver gas at the pressure in the pipe volume exceeds the maximum head that the compressor can produce at this instant. Only if the compressor is always capable of making more head than required can surge be avoided.

A further conceptual simplification can be made by splitting the flow coefficient of the recycle valve (c_v) into a part that is necessary to release the flow at the steady-state operating point of the compressor ($c_{v,ss}$) and the part that is actually available to reduce the pressure in the piping volume ($c_{v,avail}$).

The first stream and, thus, $c_{v,ss}$ of the valve necessary to cover it are known. Also known is the c_v rating of the valve. Thus, the flow portion that can effectively reduce the backpressure is determined by the difference:

$$c_{v,avail} = c_v - c_{v,ss} \quad (10)$$

The model is run at constant temperature. Most of the compressor systems modeled contain aftercoolers. The thermal capacity of the cooler and the piping are much larger than the thermal capacity of the gas; thus, the gas temperature changes are negligible within the first second. The flow calculated above in each step of the iteration is then subtracted from the gas contained in the discharge and a new pressure in the pipe volume is calculated. The calculation yields the maximum allowable piping volume for the set parameters that will not cause surge at ESD. Kurz and White (2004) validated this approach, using actual

test data. The method described can easily be expanded to situations where a relatively small suction volume leads to a fast increase in suction pressure.

Application

The model described above can be used in a software simulation program to rapidly evaluate whether a selected valve is sized correctly for the piping volume.

The model iteratively determines the maximum allowable discharge volume for a given valve configuration. This is important, because the valve size can be determined early in the project phase. With a known valve configuration, the station designer can be provided with the maximum volume of piping and coolers between the compressor and the check valve, that allows the system to avoid surge during an ESD.

The calculation requires specification of the head-flow-speed relationship of the compressor, and the definition of the surge line as a function of either compressor speed, compressor head, or compressor flow. Further, the valve needs to be described by its maximum capacity (C_v), as well as by its capacity as a function of valve travel and the opening behavior, including the delay. The discharge check valve is assumed to be closed as soon as the recycle flow exceeds the compressor flow, i.e., once the depressurization begins. The calculation procedure is started by initiating the deceleration of the train and the valve opening. For each time step, the compressor head and flow, based on speed and system pressures, and the flow through the valve, based on system pressures and valve opening, are calculated. The mass of the gas trapped between the recycle valve and compressor discharge is subsequently determined, yielding a new discharge pressure. If surge occurs, i.e., if the flow drops below the flow at the surge line, the backwards flow through the compressor is assumed to increase with time in surge, with a recovery once the required head drops 1 percent below the head at surge. The modeling of the backwards flow is not critical, and is only made to avoid numerical instabilities, because the only information that is expected from the model is whether or not the compressor will surge for the given configuration.

Figure 13 shows the deceleration of the engine's power turbine, and thus the driven compressor, following an ESD. Also shown is the response of the recycle valve.

Figures 15, 16, and 17 show typical results of these simulations. In Figure 15, the discharge volume is small enough, and while the actual flow of the compressor approaches the minimum allowable flow (surge flow) at about 500 ms after the initiation of the ESD, so surge can be avoided. In Figure 16 the compressor surges about 700 ms after the initiation of the ESD. For this configuration, either the valve size has to be increased, or the discharge volume has to be reduced, to avoid compressor surge during an ESD. In Figure 17, the system is severely underdesigned and will require significant changes including the possible addition of another valve in a hot bypass mode.

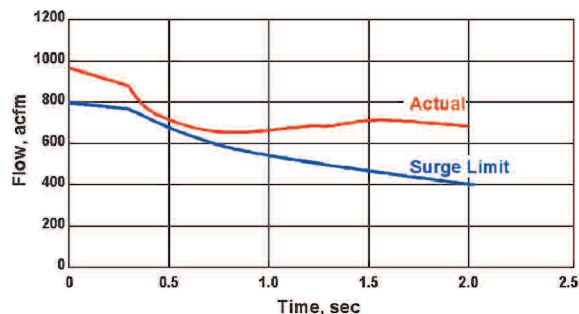


Figure 15. Actual Flow and Flow at the Surge Line During ESD—Surge Avoided.

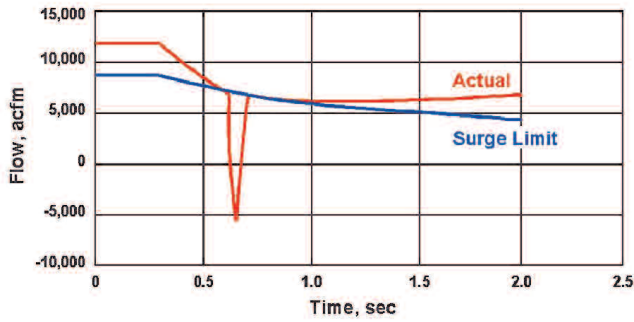


Figure 16. Actual Flow and Flow at the Surge Line During ESD—Compressor - Surge at 0.7 Sec.

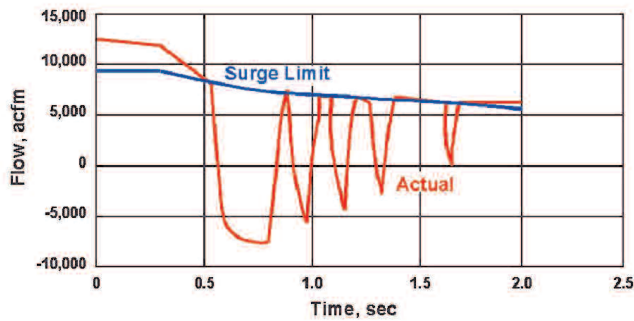


Figure 17. Actual Flow and Flow at the Surge Line During ESD—Multiple Surges.

RECYCLE ARRANGEMENTS FOR SPECIFIC APPLICATIONS

The arrangement of recycle loops impacts the operational flexibility, as well as the startup and transient behavior of a compressor station. In the following section, various arrangement concepts are described, together with their basic advantages and disadvantages.

Recycle Configurations for Single Compressors

- *Basic recycle system (Figure 18)*
 - + Small discharge volume, fast recycle response
 - Although some partial recycle can be maintained, 100 percent recycle cannot.

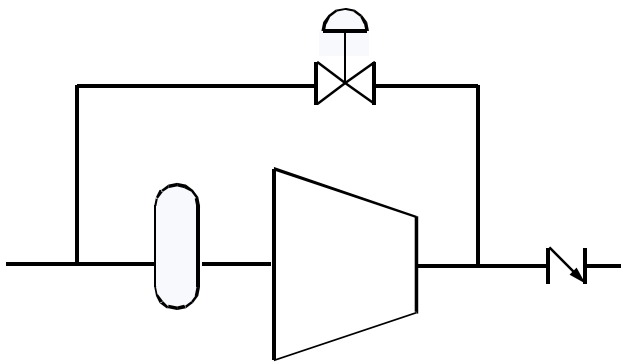


Figure 18. The Basic Recycle System (Hot Recycle).

- *Aftercooler inside recycle loop (Figure 19)*
 - + 100 percent recycle possible
 - Additional discharge volume impacts recycle response.

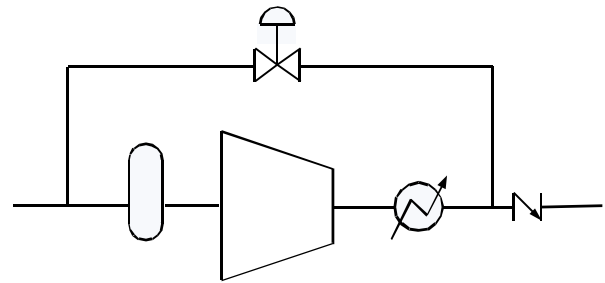


Figure 19. Aftercooler Inside Recycle Loop.

- *Precooler and aftercooling (Figure 20)*
 - + Small discharge volume, fast recycle response
 - + If there is significant heat in the suction header may improve compressor performance
 - Requires an additional cooler

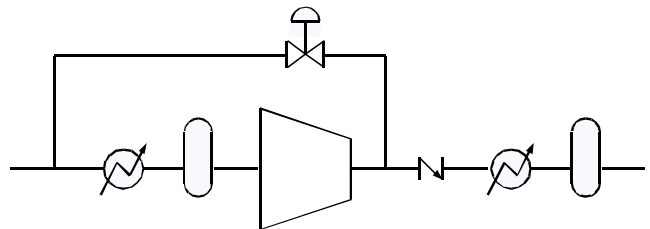


Figure 20. Precooler and Post Cooling.

- *Cooled recycle loop (Figure 21)*
 - + Small discharge volume, fast recycle response
 - + No inline pressure loss
 - Requires an additional cooler, although smaller than a pre-cooler

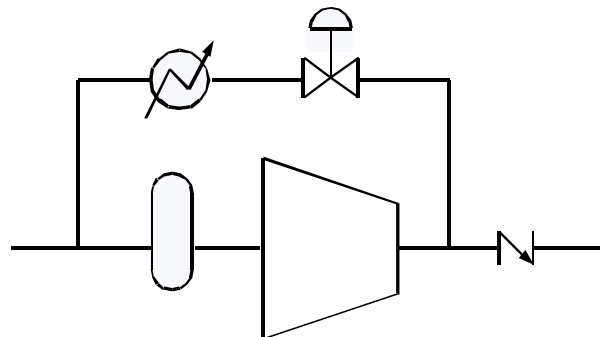


Figure 21. Recycle Cooling.

- *Hot recycle valve and cooled recycle valve (Figure 22)*
 - + Provides modulating surge control valve and shutdown valve ideally suited for their purpose
 - More components, more cost

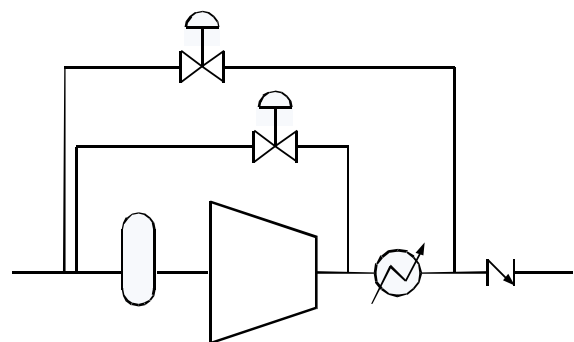


Figure 22. Hot and Cooled Recycle Valves.

- *Parallel recycle valves (Figure 23)*
- + Provide good modulating surge control and fast shutdown valves
- + Provide some level of redundancy
- More components, more cost

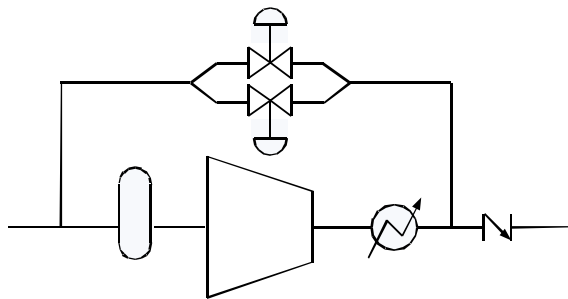


Figure 23. Parallel Recycle Valves.

Recycle Configurations for Multiple Compressors

- *Hot unit valves and cooled overall recycle valve (Figure 24)*
- + Provides good modulating surge control and fast shutdown valves
- + Provides some level of redundancy
- More components, more cost

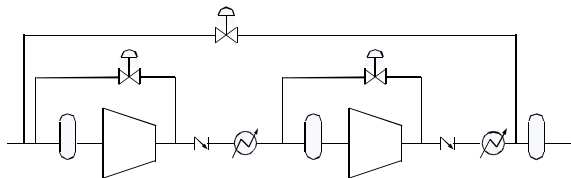


Figure 24. Hot Unit Valves Cooled Overall Recycle Valve.

- *Compressor 2 significantly larger than compressor 1: recycle valve for second compressor and overall recycle valve (Figure 25)*
- + Provides good modulating surge control and fast shutdown valves
- + Provided some level of redundancy
- More components, more cost

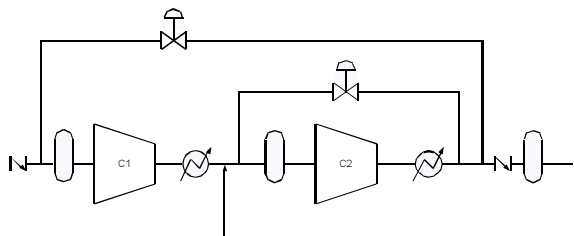


Figure 25. Recycle Valve for Second Larger Compressor and Overall Recycle Valve.

- *Low ratio (low discharge temperature) individual compressors connected in series (Figures 26 and 27)*

In either configuration any compressor can be started and put online with the recycle valve closed. As the compressor begins to make head its discharge check valve will open and its bypass check valve will close. In the configuration with a common recycle header (Figure 27), multiple parallel recycle valves are necessary. This is due to the fact that the pressure ratio over the recycle valve is reduced if one or more of the compressors upstream are shut down. This in turn requires added flow capacity in the recycle valve. This configuration also requires a number of additional block valves.

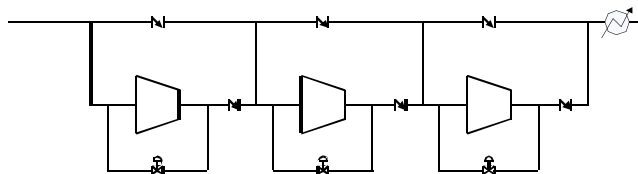


Figure 26. Low Ratio Compressors in Series: Recycle for Individual Compressor.

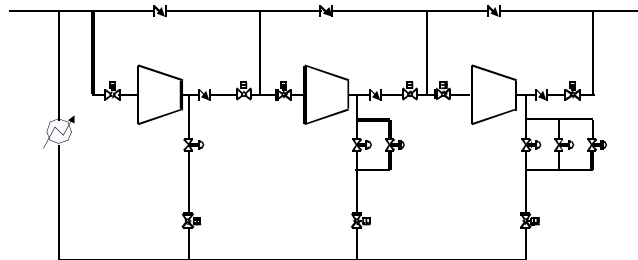


Figure 27. Low Ratio Compressors in Series: Common Recycle Header.

RECYCLE CONNECTION CONSIDERATIONS

Connecting Upstream or Downstream of Scrubbers

Connecting upstream of the suction scrubber will prevent debris left in piping from entering the compressor and will add volume in the recycle path to extend time before overheat. Connecting downstream of the suction scrubber will avoid the pressure drop across the scrubber thus pressure rise at the compressor suction will be faster and higher.

Startup Considerations

The design of the antisurge and recycle system also impacts the startup of the station. Particular attention has to be given to the capability to start up the station without having to abort the start due to conditions where allowable operating conditions are exceeded. Problems may arise from the fact that the compressor may spend a certain amount of time recycling gas, until sufficient discharge pressure is produced to open the discharge check valve (Figure 3) and gas is flowing into the pipeline.

Virtually all of the mechanical energy absorbed by the compressor is converted into heat in the discharged gas. In an uncooled recycle system, this heat is recycled into the compressor suction and then more energy added to it. A cubic foot of natural gas at 600 psi weighs about 2 lb (depending on composition). The specific heat of natural gas is about 0.5 Btu/lb (again depending on composition). One Btu/sec equals 1.416 hp. If the recycle system contains 1000 cubic feet, there is a ton of gas in it. One thousand four hundred sixteen hp will raise the temperature of the gas about 1 degree per second. This approximates what happens with 100 percent recycle. At 100 percent recycle, eventually this will lead to overheating at the compressor discharge. The problem usually occurs when there is a long period between the initial rotation of the compressor and overcoming the pressure downstream of the check valve.

Low pressure-ratio compressors often do not require aftercoolers. There are several strategies that can be employed to avoid overheating the uncooled compressor during startup:

- Accelerate quickly
- Delay hot gas re-entering the compressor
- Dilute hot gas re-entering the compressor
- Throttled recycle

Compressors without cooling must be accelerated and placed online quickly to avoid overheating. Uncooled compressor sets

cannot be started and accelerated to idle. They must be accelerated quickly through the point where the discharge check valve opens and the recycle valve closes. If acceleration slows when the discharge pressure is met, and recycle valve closes slowly, a shutdown may still occur. Often standard start sequences are very conservative and can be shortened to reduce the time it takes to get a compressor online.

Extending the length of the recycle line downstream of the recycle valve increases the total volume of gas in the recycle system. This reduces the heat buildup rate by delaying when the hot gas from the compressor discharge reaches the suction. Some heat will be radiated through the pipe walls. If the outlet is far upstream into a flowing suction header, dilution will occur.

Figures 28 and 29 outline a solution to a rather difficult starting problem for a compressor station without aftercooling capacity. In Figure 28, to start the first unit is relatively easy, because there is virtually no pressure differential across the main line check valve, and therefore the unit check valve will open almost immediately, allowing the flow of compressed gas into the pipeline. However, if one additional unit is to be started, the station already operates at a considerable pressure ratio, and therefore the unit check-valve will not open until the pressure ratio of the starting unit exceeds the station pressure ratio. Ordinarily the unit would invariably shut down on high temperature before this can be achieved. By routing the recycle line into the common station header (Figure 29), the heat from the unit coming online is mixed with the station suction flow. This equalizes the inlet temperature of all compressors; higher for the compressors already online, lower for the compressor coming online. With this arrangement overheating of a compressor coming online is nearly always avoided.

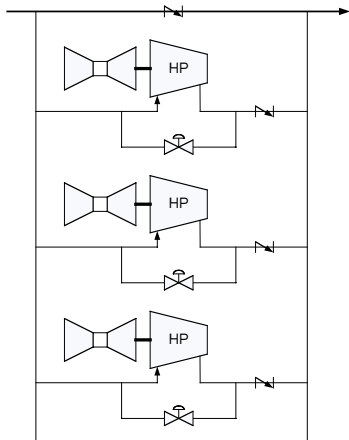


Figure 28. Original Station Layout.

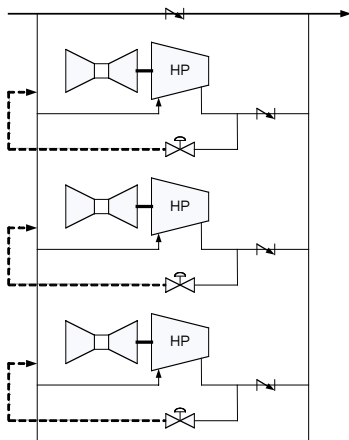


Figure 29. Improved Station Layout.

Figure 30 shows the problem of a conventional system that includes 3000 ft of 24 inch pipe without aftercooling. The temperature in the recycle line starts to rise and, assuming a shutdown set point of 350°F, the compressor would shut down after about 20 minutes.

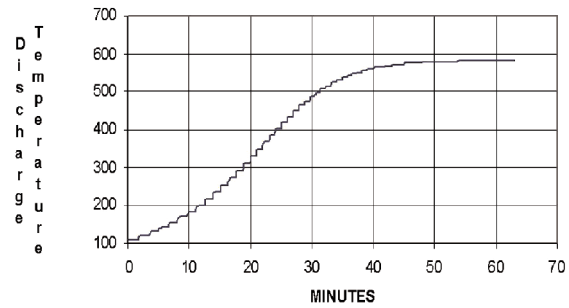


Figure 30. Temperature Build Up.

Figures 31 and 32 outline the startup event with the revised system. The power turbine and the compressor start to rotate once the gas producer provides sufficient power. Subsequently, the gas temperature rises, but, because the discharge pressure required to open the check valve is reached fast enough, overheating can be avoided. The temperature rise in the recycle loop during startup is shown as a function of power turbine and compressor speed (Figure 31), gas producer speed (Figure 32), and time (in minutes) (Figure 33). The power turbine starts to turn at about 75 percent gas producer speed, at which point the temperature starts to rise. After the discharge check valve opens (at 0.2 minutes after the compressor starts to rotate, 95 percent gas producer speed and 70 percent power turbine speed), the temperature drops rapidly.

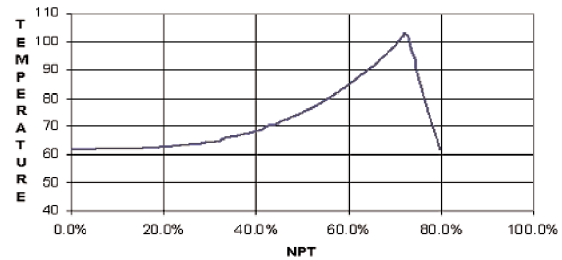


Figure 31. Temperature (°F) Versus NPT (Percent).

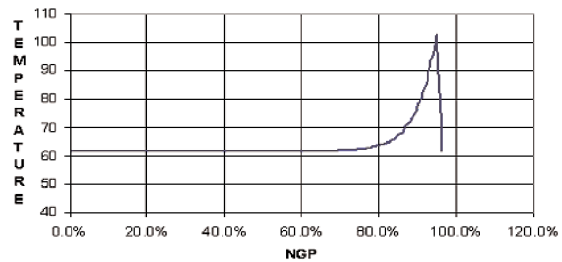


Figure 32. Temperature (°F) Versus NGP (Percent).

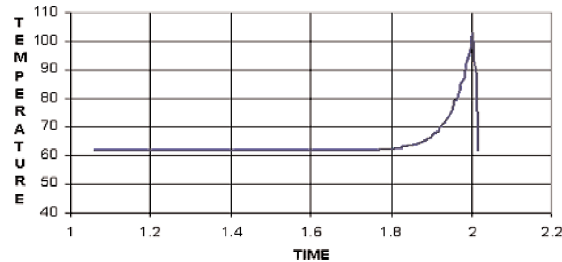


Figure 33. Temperature (°F) Versus Time (Min.).

Further analysis of the startup problem indicates the advantage of throttling the recycle valve, rather than starting the unit with the recycle valve fully open.

With the valve sizing tool described previously it can be determined exactly what valve opening will be required to maintain a specific surge margin at steady-state operation. As the compressor is accelerating, flow is increasing. The pressure in the discharge is lower and the pressure in the suction is higher than they would be if the compressor at this speed were steady-state. This is due to the effect of the suction and discharge volumes. This also causes the flow to be higher and subsequently the surge margin will be higher. As such, if the valve is set at a fixed position to obtain a fixed small surge margin, the actual surge margin will be higher during acceleration.

To use this strategy safely the control must be able to sense a loss of acceleration (flame out) and if detected open all recycle valves immediately. As the volumes up and downstream of the compressor cause the surge margin to be higher during acceleration they make surge avoidance more challenging with loss of speed.

Figure 12 illustrates this: at 70 percent open setting, the startup of the compressor is significantly closer to the surge line than at 100 percent open setting. For any given speed, the power requirement of the compressor is lower when it is closer to surge than when it is farther in choke. Therefore, for a given amount of available power, the start is quicker if the compressor operates closer to surge. If the rate of acceleration is quicker, the heat input into the system is lower. Actively modulating the surge during startup is virtually impossible as the parameters defining the surge limit of the compressor are too low to be practically measured. Returning to Figure 5 the surge limit of a compressor matches well with a fixed travel (constant C_v) line for a recycle valve. As such, a compressor can be started with a fixed recycle valve position.

OUTLOOK

There are ongoing efforts to improve surge avoidance systems. One line of efforts attempts to increase the stability margin of a compressor by active (Epstein, et al., 1994; Blanchini, et al., 2001) or passive means (Arnulfi, et al., 2000). Other efforts try to increase the accuracy of determining the surge margin (McKee and Deffenbaugh, 2003) by detecting the certain precursors of surge. However, most of the ideas will remain valid even if some of the new methods, currently in an experimental stage, are introduced. This is due to the fact that surge avoidance is a systems issue and meaningful gains can be made by better understanding the interaction between the compressor, the antisurge devices (control system, valves), and the station piping layout (coolers, scrubbers, check valves).

CONCLUSION

This paper has addressed the key factors that must be considered in the design of surge avoidance systems. The most important point is the realization that surge avoidance must be viewed in terms of the total system and not as an isolated item looking only at the compressor itself.

NOMENCLATURE

A	= Flow area
c_v	= Flow coefficient ($c_v = Q/\sqrt{SG/\Delta p}$)
C	= Compressible valve coefficient
F_p	= Piping geometry factor
h	= Head
H_{cooler}	= Gas cooler heat transfer
J	= Inertia
k	= Isentropic exponent
K	= Constant
K_v	= Valve coefficient
L	= Pipe length
N	= Speed (1/s)
p	= Pressure

Q	= Volumetric flow
SG	= Specific gravity
SM	= Surge margin (percent)
T	= Temperature
t	= Time
V	= Volume
Y	= Expansion factor
Z	= Compressibility factor
α, β, γ	= Constants
ρ	= Density

Subscripts

avail	= Available
compr	= Compressor
op	= Operating point
p	= Polytropic
surge	= At surge
std	= At standard conditions
ss	= Steady-state
v	= Valve
1	= Compressor inlet
2	= Compressor discharge

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