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TUTORIAL: SURGE CONTROL AND DYNAMIC BEHAVIOR FOR CENTRIFUGAL GAS COMPRESSORS

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ABSTRACT

This tutorial discusses the design and function of surge control systems, or more precisely, surge avoidance systems

on the basis of the behavior of centrifugal compressors when they interact with compression systems.

Therefore, the control system, as well as the phenomenon of surge and stall is discussed. The paper emphasizes the interrelation between the process system and the compressor. Regarding the compressor, different methods of controlling the compressor are discussed, together with different operating regions like stall and choke. The differences between stall and surge are explained. Additionally, the impact of the process and how the process dynamics interact with the compressor is analyzed, categorized, and explained.

Based on the above, process control and in particular, surge control systems are discussed. Critical features of surge control systems are explained, and options for different arrangements are given.

INTRODUCTION

The control of centrifugal compressors has to be considered both from the perspective of the compressor and the perspective of the process.

Regarding the compressor, it is necessary to discuss the different control devices, such as variable speed, guide vanes, throttles or recycle valves. It is also important whether a steady state compressor map is still valid in the case of fast transients. Further, the different operating conditions of the compressor, such as surge, stall, and choke have to be explained.

For the process, one must understand the relationship between the flow through the system, and the pressures imposed on the compressor. These relationships are different depending on their rate of change, in other words, one must expect different system responses for fast and slow changes as well as steady state conditions.

Lastly, in a discussion of control, one has to define the goal of a control system. The requirements to protect the process as well as the equipment have, of course, priority. But other goals need to be defined, too, in particular if the station involves multiple compression units, either in series or in parallel. Possible goals can be to minimize fuel consumption, to minimize maintenance cost or to maximize throughput.

The framework described above also defines the structure of this tutorial. The centrifugal compressor and its control, will be addressed first, and is followed by a description of the process behavior under various scenarios. A prominent role in these descriptions will be taken by surge avoidance considerations. The tutorial will close with considerations on how to control multiple units.

THE CENTRIFUGAL COMPRESSOR

Components

A centrifugal compressor is a machine that reacts to an externally imposed suction and discharge pressure with a certain flow, assuming the driver provides enough power.

First the essential components of a centrifugal compressor that accomplish the tasks specified above (Figure 1) are introduced. The gas entering the inlet nozzle of the compressor is guided (often with the help of guide vanes) to the inlet of the impeller. An impeller consists of a number of rotating vanes that impart mechanical energy to the gas. As we will see later, the gas will leave the impeller with an increased velocity and increased static pressure. In the diffuser, part of the velocity is converted into static pressure. Diffusers can be vaneless or contain a number of vanes. If the compressor has more than one impeller, the gas will be again brought in front of the next impeller through the return channel and the return vanes. If the compressor has only one impeller, or after the diffuser of the last impeller in a multi stage compressor, the gas enters the discharge system. The discharge system can either make use of a volute, which can further convert velocity into static pressure, or a simple cavity that collects the gas before it exits the compressor through the discharge nozzle.

Aerodynamics

The rotating part of the compressor consists of all the impellers. This rotor runs on two radial bearings (on all modern compressors, these are hydrodynamic tilt pad bearings), while the axial thrust generated by the impellers is balanced by a balance piston, and the residual axial force is balanced by a hydrodynamic tilt pad thrust bearing.

To keep the gas from escaping at the shaft ends, dry gas seals are used on both shaft ends. Other seal types have been used in the past, but virtually all modern centrifugal compressors in the pipeline service use dry gas seals. The sealing is accomplished by a stationary and a rotating disk, with a very small gap (about 5µm) between them. At standstill, springs press the movable seal disc onto the stationary disk. Once the compressor shaft starts to rotate, the groove pattern on one of the discs causes a separating force, making the seals run without mechanical contact of sealing surfaces. The entire assembly is contained in a casing, which for typical oil and gas applications at pressures above 30 or 40 bar, is usually of the barrel type.
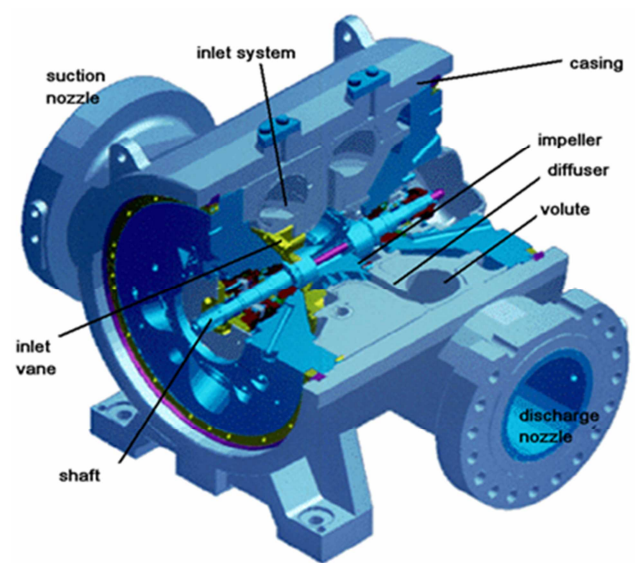

Figure 1. Components of a centrifugal compressor

The impellers impart mechanical energy to the gas and in the diffuser part of the velocity is converted into static

pressure. Bernoulli's law (which is strictly true only for incompressible flows, but which can be modified for the subsonic compressible flows found in gas compressors) describes the interchangeability of two forms of energy: pressure energy (static pressure) and kinetic energy (velocity). Another requirement is, that mass cannot appear or disappear. This requirement is valid for compressible and incompressible flows, with the caveat that for compressible flows the density is a function of pressure and temperatures, and thus ultimately a function of the velocity.

These two concepts explain the working principles of the vanes and diffusers used. Due to requirement for mass conservation, any flow channel that has a wider flow area at its inlet and a smaller flow area at its exit will require a velocity increase from inlet to exit. If no energy is introduced to the system, Bernoulli's law requires a drop in static pressure. Examples for flow channels like this are turbine blades and nozzles, inlet vanes in compressors and others. Conversely, any flow channel that has a smaller flow area A at its inlet and a larger flow area at its exit will require a velocity decrease from inlet to exit. If no energy is introduced to the system, Bernoulli's law requires an increase in static pressure. Examples for flow channels like this are vaned or vaneless diffusers, flow channels in impellers, rotor and stator blades of axial compressors volutes and other.

If these flow channels are in a rotating system (for example in an impeller), mechanical energy is added to or removed from the system. Nevertheless, if the velocities are considered in a rotating system of coordinates, above principles are applicable as well.

Another important concept is the conservation of momentum. The change in momentum M of gas flowing from a point 1 to a point 2 is the change its mass times velocity (m c), and is equal to the net force F acting on it. The change in momentum is

$$\frac{d\vec{M}}{dt} = \dot{m}(\vec{c}_2 - \vec{c}_1) = \vec{F} \qquad (1)$$

To change the momentum of this gas, either by changing the velocity or the direction of the gas (or both) from w_1 to w_2, a force is necessary. Figure 2 outlines this concept for the case of a bent, conical pipe. The gas flows in through the area A_1 with w_1, p_1, and out through the flow area A_2 with w_2, p_2. The differences in the force due the pressure (p_1A_1 and p_2A_2, respectively), and the fact that a certain mass flow of gas is forced to change its direction generates a reaction force F_R. Split into x and y coordinates, and considering that

$$\dot{m} = \rho_1 A_1 w_1 = \rho_2 A_2 w_2 \qquad (2)$$

one gets (due to the choice of coordinates, $w_{1y}=0$)

$$x: \quad \rho A_1 w_1 (w_{2x} - w_1) = p_1 A_1 - (p_2 A_2)_x + F_{Rx}$$
$$y: \quad \rho A_1 w_1 (w_{2y}) = -(p_2 A_2)_y + F_{Ry}$$
$$(3)$$

It should also be noted that this formulation is also valid for viscous flows, because the friction forces become internal forces. All these concepts are applied in a very similar way in pipeline flows.
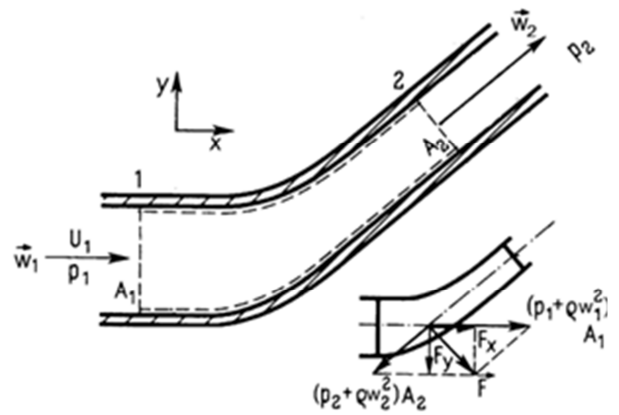

Figure 2. Conservation of momentum

For a rotating row of vanes in order to change the velocity of the gas, the vanes have to exert a force upon the gas. This is fundamentally the same force F_{Ry} that acts in the previous example for the pipe. This force has to act in direction of the circumferential rotation of the vanes in order to do work on the gas. According to the conservation of momentum, the force that the blades exert is balanced by the change in circumferential velocity times the associated mass of the gas. This relationship is often referred to as Euler's Law:

$$P = \dot{m} \cdot \Delta h = \dot{m} \cdot (u_2 c_{u2} - u_1 c_{u1}) \qquad (4)$$

where u is the circumferential blade velocity at the inlet (1) and exit (2) of the impeller, and c_u is the circumferential component of the gas velocity, taken in an absolute reference frame at the inlet (1) and exit (2).

The general behavior of any gas compressor can be gauged by some additional, fundamental relationships: The vanes of the rotating impeller 'see' the gas in a coordinate system that rotates with the impeller. The transformation of velocity coordinates from an absolute frame of reference (c) to a frame of reference rotating with a velocity u is by:

$$\vec{w} = \vec{c} - \vec{u} \qquad (5)$$

where, for any diameter D and speed N of the impeller $u=\pi DN$.

At this point, one of the advantages of centrifugal compressors over axial compressors becomes apparent: In the axial compressor, the entire energy transfer has to come from the turning of the flow imposed by the blade ($c_{u2}-c_{u1}$), while the centrifugal compressor at a speed N has added support from the centrifugal forces on the gas while flowing from the diameter D_i at the impeller inlet, rotating at with the velocity $u_1=\pi D_i N$, to the higher diameter at the impeller exit D_{tip}, rotating with the velocity $u_2=\pi D_{tip} N$ (Figure 3).
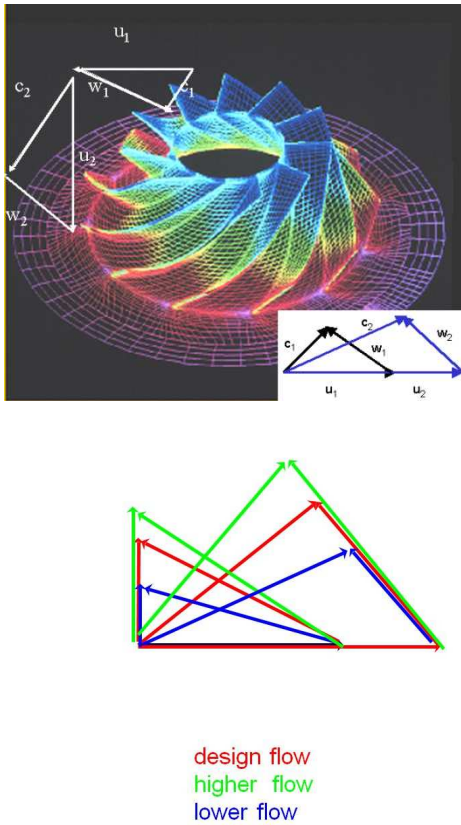
Figure 3. Velocity vectors in a centrifugal impeller

The importance of Euler's law lies in the fact that it connects aerodynamic considerations (i.e., the velocities involved) with the thermodynamics of the compression process.
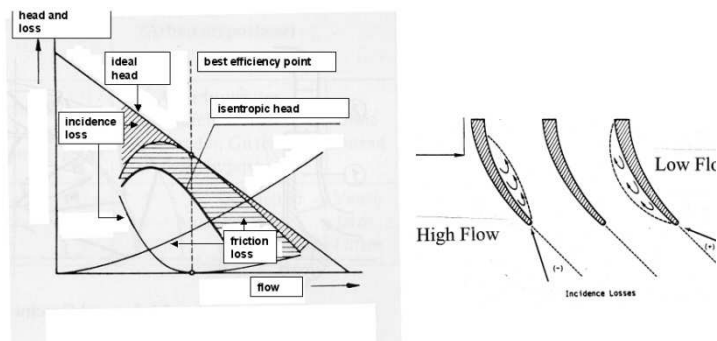

Figure 4. Head versus flow relationship at constant speed

The impeller exit geometry ('backsweep') determines the direction of the relative velocity w_2 at the impeller exit. The basic 'ideal' slope of head vs. flow is dictated by the kinematic flow relationship of the compressor, in particular the amount of backsweep of the impeller. Any increase in flow at constant speed (Figure 3) causes a reduction of the circumferential component of the absolute exit velocity (c_{u2}). It follows from Euler's equation above, that this causes a reduction in head. Adding the influence of various losses to this basic relationship shape the head-flow-efficiency characteristic of a compressor (*Figure 4*).

Whenever the flow deviates from the flow the stage was designed for, the components of the stage operate at lower efficiency. This is due to incidence losses. *Figure 5* illustrates this, using an airfoil as an example: At the 'design flow' the air follows the contours of the airfoil. If the direction of the incoming air is changed, increasing zones occur where the airflow ceases to follow the contours of the airfoil, and create increasing losses. Furthermore, the higher the flow, the higher the velocities and, thus, the friction losses.
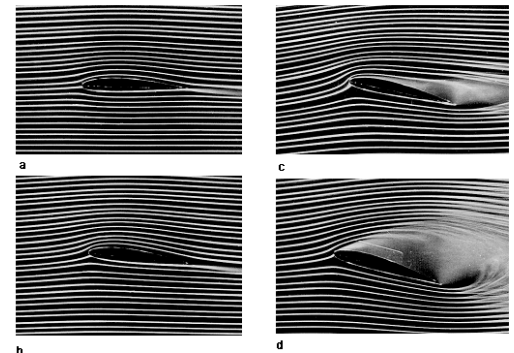

Figure 5. Unseparated (a, b), partially separated (c), and fully separated (d) flow over an airfoil at increasing angle of attack (Nakajama [1])

A compressor, operated at constant speed, is operated at its best efficiency point ('Best Efficiency Point' in Figure 4). If, without changing the speed, the flow through the compressor is reduced (for example, because the discharge pressure that the compressor has to overcome is increased), then the compressor efficiency decreases as well. At a certain flow, stall, probably in the form of rotating stall, in one or more of the compressor components will occur. At further flow reduction, the compressor will eventually reach its stability limit, and go into surge.

If, again starting from the best efficiency point, the flow is increased, then the efficiency is also reduced, accompanied by a reduction in head. Eventually the head and efficiency will drop steeply, until the compressor will not produce any head at all. This operating scenario is called choke.

Stall

If the flow through a compressor at constant speed is reduced, the losses in all aerodynamic components will increase. Eventually the flow in one of the aerodynamic components, usually in the diffuser, but sometimes in the impeller inlet, will separate (*Figure 5d* shows such a flow separation for an airfoil).

Flow separation in a vaneless diffuser means, that all or parts of the flow will not exit the diffuser on its discharge end, but will form areas where the flow stagnates or reverses its direction back to the inlet of the diffuser (i.e., the impeller exit).

Stall in the impeller inlet or a vaned diffuser is due to the fact, that the direction of the incoming flow (relative to the rotating impeller) changes with the flow rate through the compressor. Therefore, a reduction in flow will lead to an increased mismatch between the direction of the incoming flow the impeller or the diffuser vanes were designed for and the actual direction of the incoming flow. At one point this mismatch becomes so significant that the flow through the impeller or the vanes breaks down. Usually, vanes in the diffuser reduce the operating range of a stage compared to a vaneless diffuser

Flow separation can take on the characteristics of a rotating stall. When the flow through the compressor stage is reduced, parts of the diffuser experience flow separations. Rotating stall occurs if the regions of flow separation are not stationary, but move in the direction of the rotating impeller (typically at 15-30% of the impeller speed). Rotating stall can often be detected from increasing vibration signatures in the sub-synchronous region. Onset of stall does not necessarily constitute an operating limit of the compressor. In fact, in many cases the flow can be reduced further before the actual stability limit is reached.

Choke

At high flow the head and efficiency will drop steeply, until the compressor will not produce any head at all. This operating scenario is called choke. However, for practical applications, the compressor is usually considered to be in choke when the head falls below a certain percentage of the head at the best efficiency point. Some compressor manufacturers do not allow operation of their machines in deep choke. In these cases, the compressor map has a distinct high flow limit for each speed line.

The efficiency starts to drop off at higher flows, because a higher flow causes higher internal velocities, and thus higher friction losses. The increased mismatch between design and actual incidence further increases the losses. The head reduction is a result of both the increased losses and the basic kinematic relationships in a centrifugal compressor: Even without any losses, a compressor with backwards bent blades (as they are used in virtually every industrial centrifugal compressor) will experience a reduction in head with increased flow (*Figure 4*). 'Choke' and 'Stonewall' are different terms for the same phenomenon.

Surge

At flows lower than the stability limit, practical operation of the compressor is not possible. At flows on the left of the stability limit, the compressor cannot produce the same head as at the stability limit. It is therefore no longer able to overcome the pressure differential between suction and discharge side. Because the gas volumes upstream (at discharge pressure) is now at a higher pressure than the compressor can achieve, the gas will follow its natural tendency to flow from the higher to the lower pressure: The flow through the compressor is reversed. Due to the flow reversal, the system pressure at the discharge side will be reduced over time, and eventually the compressor will be able to overcome the pressure on the discharge side again. If no

corrective action is taken, the compressor will again operate to the left of the stability limit and the above described cycle is repeated: The compressor is in surge. The observer will detect strong oscillations of pressure and flow in the compression system.

It must be noted that the violence, frequency and the onset of surge are a function of the interaction between the compressor and the piping system.

Surge Margin and Turndown

Any operating point O can be characterized by its distance from the onset of surge at the flow Q_s. Two definitions are widely used: The surge margin

$$SM(\%) = \frac{Q_O - Q_S}{Q_O} \cdot 100 \quad (6)$$

which is based on the flow margin between the operating point O and the surge point at constant speed B, and the turndown

$$Turndown(\%) = \frac{Q_O - Q_C}{Q_O} \cdot 100 \quad (7)$$

which is based on on the flow margin between the operating point O and the surge point at constant head C.

Compressor Control

There are two objectives for compressor control: meeting the external process requirements and keeping the compressor within its operational boundaries. Typical control scenarios that have to be considered are process control, starting and stopping of units, and fast or emergency shutdowns.

Compressor configurations within a station can include: single compressors, single compressors supplied from or delivering into multiple headers, multiple compressors operated in parallel, multiple compressors operated in series, multiple compressors, or compressors with multiple sections operated in a train. Variations may include multiple compressors in a train, with control of intermediate pressures, or multiple compressor trains in parallel.

The interaction between a compressor and a compression system, in conjunction with control mechanisms and the compressor characteristic determine the operating point of the compressor in a given situation.

Control mechanisms can be:

- Power input from driver.
- Compressor speed
- Guide vane settings
- Compressor suction or discharge pressure set by a throttle
- Compressor throughput modified by recycling gas
- Multiple Units

For the single compressor the application of these control functions is fairly simple. For more complex compressor

applications as described above, these control strategies can become very complex.

External process objectives can be minimum suction pressure, maximum discharge pressure, or delivered flow.

The operating envelope of a centrifugal compressor is limited by the maximum allowable speed (or, for other control means, the maximum guide vane angle), the minimum flow (surge flow), and the maximum flow (choke or stonewall), and the minimum speed (
Figure 6). Another limiting factor may be the available driver power. Since each operating condition of a centrifugal compressor requires a certain amount of power, power limitations can significantly reduce the possible range of compressors. For variable speed compressors, setting the amount of driver power can be used to set the operating point and operating speed of the compressor.

Only the minimum flow requires special attention, because it is defined by an aerodynamic stability limit of the compressor. Crossing this limit to lower flows will cause a flow reversal in the compressor, which can damage the compressor. Modern control systems prevent this situation by automatically opening a recycle valve. For this reason, virtually all modern compressor installations use a recycle line with a control valve that allows the flow through the compressor to increase if it comes near the stability limit. The control system constantly monitors the operating point of the compressor in relation to its surge line, and automatically opens or closes the recycle valve if necessary. For most applications, the operating mode with open, or partially open recycle valve is only used for start-up and shut-down, or for brief periods during upset operating conditions.

Variable Speed

Compressor drivers that can operate at variable speed (two shaft gas turbines, steam turbines, turbo expanders, electric motors with variable frequency drives or variable speed gearboxes) allow the compressor to operate over a range of different speeds. The faster the compressor runs, the more head and flow it generates, and the more power it consumes. The efficiency characteristics of the compressor are retained for different speeds, so this is a very efficient way of adjusting the compressor to a wide range of different operating conditions.
Figure 6 shows the resulting map.
It should be noted that in many cases, variable speed control is really used as variable power control. If the compressor is driven by a two shaft gas turbine, then the parameter that is controlled is the output power of the gas turbine. After the power from the driver is decreased or increased, the compressor settles at a speed that is determined by the balance between driver provided power and compressor absorbed power. The operating points displayed in
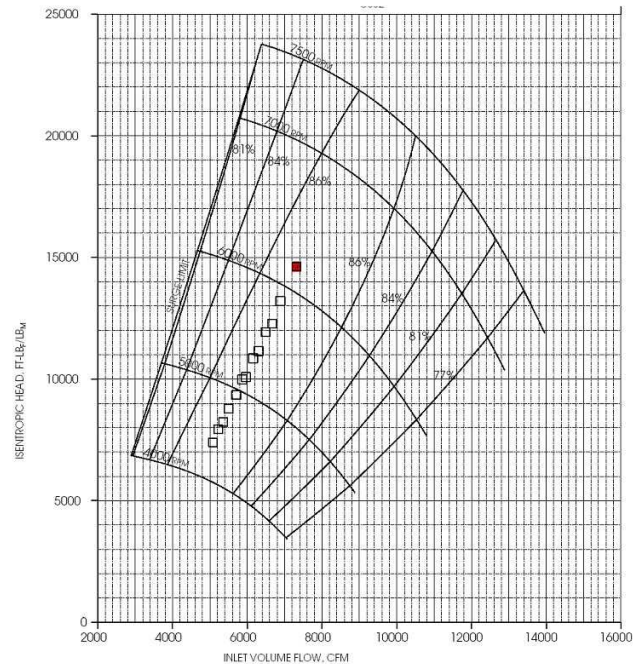*Figure 6* are reached by changing the driver power setting.

Figure 6. Typical pipeline operating points plotted on a typical centrifugal compressor performance map

Adjustable Inlet Vanes

Modifying the swirl of the flow into the impeller allows the operating characteristics of the stage to be adjusted (***Figure 7***). This can be accomplished by adjustable vanes upstream of the impeller. Increasing the swirl against the rotation of the impeller increases the head and flow through the stage. Increasing the swirl with the rotation of the impeller reduces the head and flow through the stage. This is a very effective way to increase the range for a single stage (***Figure 7***). In multistage compressors, the range increase is limited if only the first stage has adjustable vanes. The technical difficulty for high pressure compressors lies in the fact that complicated mechanical linkage has to be actuated from outside the pressure containing body.

Adjustable Diffusor Vanes

Vaned diffusers tend to limit the operating range of the compressor because the vanes are subject to increased incidence at off design conditions, thus eventual causing stall. Adjustable diffuser vanes allow the alignment of the vanes with the changing flow conditions, thus effectively allowing for operation at much lower flows by delaying the onset of diffuser stall (Figure 7). They will not increase the head or flow capability of the stage. In multistage compressors, the range increase is limited if only one stage has adjustable vanes. The technical difficulty for high pressure compressors lies in the fact that complicated mechanical linkage has to be actuated from outside the pressure containing body. Another issue is that for the vanes to operate, small gaps between the vanes and the diffuser walls have to exist. Unavoidable leakage through these gaps causes efficiency and range penalties, particularly in machines with narrow diffusers.

Figure 7. Control methods for centrifugal compressors: Throttling, variable speed, and adjustable guide vanes (Rasmussen et al. [2])

Figure 8. Power consumption for different control methods

Throttling (Suction, Discharge)

A throttle valve on the suction or discharge side of the compressor increases the pressure ratio the compressor sees, and therefore moves the operating point to lower flows on the constant speed map. It is a very effective, but inefficient way of controlling compressors (***Figure 7*** and ***Figure 8***).

Recycling

A controlled recycle loop allows a certain amount of the process flow to go from the compressor discharge back to compressor suction. The compressor therefore sees a flow that is higher than the process flow. This is a very effective, but inefficient way to allow the compressor system to operate at a low flow (***Figure 8***).

In general, multiple variables interact with other. For example, in a variable speed compressor, changing the amount of input power will vary the speed of the machine as well. In this case, the control variable is fuel flow or current (which determines the power input into the compressor), and speed becomes a dependent variable.

Multiple Units

If a compressor station operates multiple compressors, either identical units or dissimilar units, turning these units on or off provides a powerful means of control. For particular applications that see large swings in load, this control method allows to operate the drivers near their optimum efficiency conditions for most of time. It also has a positive impact on maintenance costs, since the drivers are only operated when needed. The method works well if the selected equipment has a high starting reliability. It is also advantageous if the compressors can stay pressurized and can be restarted without having to vent gas into the atmosphere. Modern centrifugal compressors with dry gas seals are capable of this 'pressurized hold'.

Having multiple units on a station, the question becomes how to control them to achieve certain objectives. These objectives may be minimizing the running hours of units, optimizing the capability to absorb load swings, minimizing fuel consumption or emissions. The first objective requires running as few units as possible, while the second may require running all or most units at part load for most of the time. Minimizing fuel consumption (which also equates to minimizing CO_2 production) will usually involve strategies to cover the load with as few units as possible running (i.e., running units as close to full load as possible or not running them at all). It then becomes a question whether it is better to run the operating units but one at full load, with the remaining unit at part load; or, to run all units at part load.

For two identical units running, the answer is virtually always to run both of them at equal part load. This is accommodated by operating the compressors at equal surge margin or at equal gas turbine load (i.e., equal gas producer speed).

For more than two identical units running, the difference in fuel usage between n units running at the same load, and some running units at full load, and the remaining at part load, is usually very small. The optimum there is more often determined by the resulting operating points of the compressors.

For units that are not identical, it is usually better to load the more efficient unit and capture the load swings with the less efficient unit. In some instances, these schemes are also dictated by the starting reliability of individual units, i.e., a
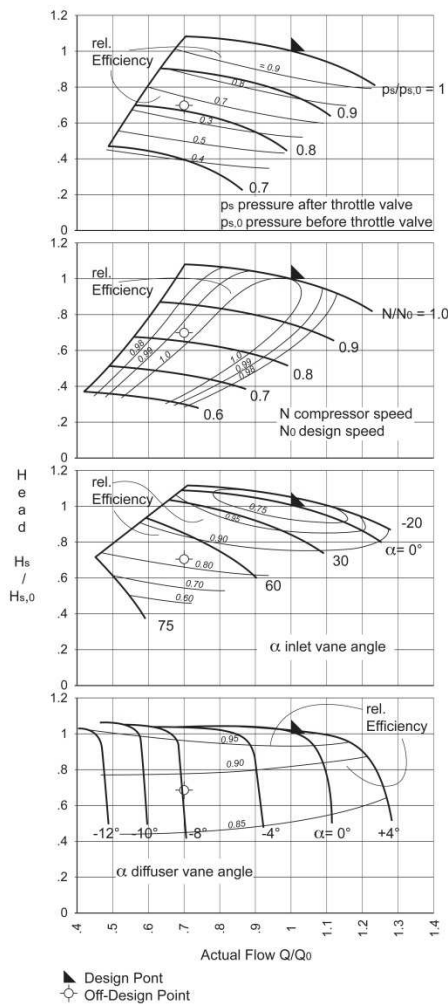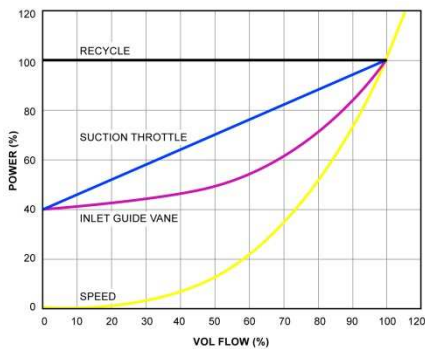
low starting reliability may dictate operational schemes that are otherwise less fuel efficient.

Two methods are frequently used for load equalization:

- Turndown equalization (surge margin equalization), i.e., operating all compressors at the same turndown.
- Power equalization, i.e., operating all drivers at the same load. For gas turbines, this would be accomplished by operating all gas turbines at the same gas producer speed.

Turndown equalization works equally well for compressor sets in series or parallel. The challenge comes when the compressors reach their respective surge protection margins and begin recycling. At this point the turndown of the compressors is fixed by the surge avoidance system. Continuing to turn down the compressor sets after recycle has begun becomes more challenging.

If the compressor sets are identical and the recycle valves are identical the task is simplified. Load sharing can be accomplished by either equalizing the turndown, equalizing compressor speed or equalizing the driver power.

Equalizing the driver power will keep the compressors operating the same to the degree that the compressor sets (engine and compressor) are matched. Equalizing their compressor speeds will keep the compressors operating same as long as the compressors are matched.

At the point recycle begins or ceases it is necessary that all compressor sets switch to the new control mode simultaneously. In recycle, the recycle valve commands are equalized. Then further turndown can be accomplished by reducing compressor speeds equally. If demand increases and recycle ceases to be necessary they switch back to surge margin equalization.

Load sharing can also be applied to dissimilar compressors. There can be one or more large compressor sets in parallel with one or more, smaller compressor sets. The larger units would form the baseline compression, as they typically are more efficient. The smaller less efficient units operate as peaking units. Typically the smaller units would be started and shutdown as necessary and recycling would be avoided.

Multiple compressor sets in series may have the same drivers but may not have the same compressors nor operate at the same power. Again, multiple compressor sets in series can be operated at equal turndown. If part of the operational envelope includes recycling again an alternate control strategy must be adopted.

Another method of control is to maintain the compressors at their design percentage of the total pressure rise. With this control strategy the surge control systems can be allowed to operate independently.

If multiple compressors are driven by the same engine and each has set points, a new control strategy is required. Engine power (and with it the compressor speed) must be allowed to increase until the last compressor achieves its control objective. The other compressors, having passed their set points, are controlled by recycling.

Applications with Multiple Services

Many compressor applications require the simultaneous control of several gas streams. This is the case where multiple gas streams at different pressure levels have to be compressed, for example in Flash gas compressors, or in gas gathering applications where the wells produce at different pressure levels. As a general rule, each independent gas stream needs a separate control mechanism. In these cases, combinations of speed control, suction throttling, or recycling can be used. It is usually advantageous to control the larger stream via speed control and the smaller stream via recycle or throttle control, because the speed control is more energy efficient. Circumstances may require that both streams have to have a throttle valve. In this case, the control system needs to assure that always at least one of the throttle valves is fully open.
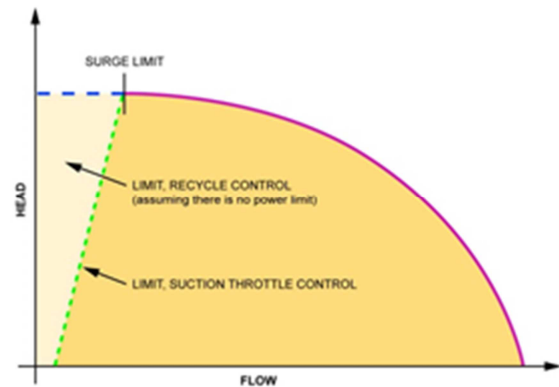

Figure 9. Operating range for constant speed compressor with suction throttling or recycle (assuming the driver imposes no additional power limitations)

Constant Speed Applications

For compressors where the driver does not allow for speed variation, and where variable vanes are not practical, all possible operating points of the compressor lie on a single head-flow curve (***Figure 9***). In many oil and gas applications, the control of these compressors is accomplished by either suction throttling or recycling. This is an effective, but inefficient way of controlling the compressor.

Compressor Performance under Transient Conditions

Data presented by Blieske et al. [3] support the general practice to use steady state speed, head, and flow data for the transient behavior of centrifugal compressors during emergency shutdowns. The data identify heat transfer from the compressor body (which has a significant heat storage capacity due to its large amount of material) into the gas during transient events as the major source of deviations. This has a measurable impact on the apparent compressor efficiency and the absorbed power. Increased accuracy for transient simulation can thus be gained by taking this heat transfer into account.

Shutdown simulations will use the compressor absorbed power calculated from a steady state performance map. Analyzing the test data and comparing the power calculated from the steady state map with the actually absorbed power the authors identified a significant difference. Therefore, using

the power calculated from the map in a shutdown simulation will introduce errors in the assumed rate of speed change.

THE COMPRESSION PROCESS

System Behavior

Compression systems enforce certain relationships between pressure ratio (or isentropic head) of the compressor and the flow through the compressor. In the context of compressor applications, it is important to understand this relationship, since it has a profound impact on the selection of the correct compressor. Further, these relationships tend to be different in steady state operation versus transient operation.

The pipe system within which the compressor operates will impose its characteristic on the compressor. There are three fundamental steady state system characteristics that need to be considered:

- Strong head-flow relationship
- weak head-flow relationship
- integrative relationship
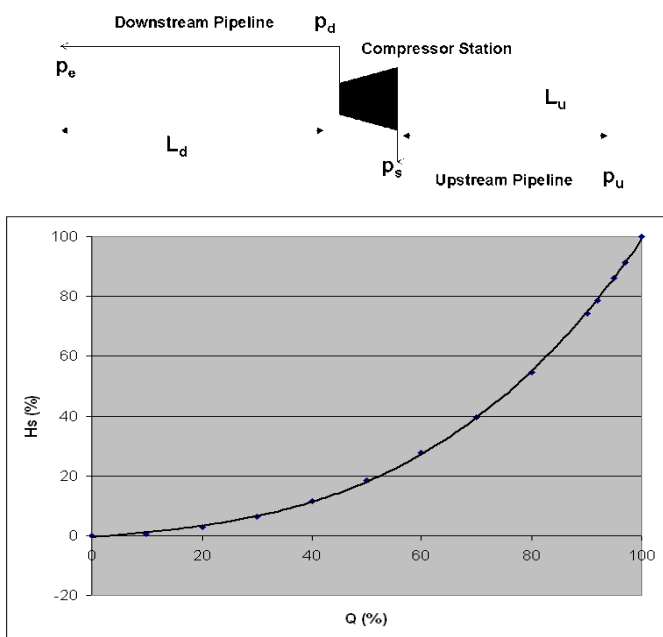
Figure 10. Steady state pipeline characteristic

The case of strong head-flow relationship is, for example, seen in pipelines (

Figure 10). Under steady state conditions, the pressure loss in the pipeline which imposes the suction and discharge pressure on the compressor station increases significantly when the flow through the pipeline has to be increased. The pressure levels are thus dictated by friction losses, which depend on the gas velocity in the pipe.

In a weak head flow relationship, the head requirement for the compressor head stays more or less constant with changes in flow. This behavior is found in refrigeration compressors, but also for situations where the process dictates

a constant suction pressure (e.g., separator pressure), while the discharge gas is fed via a short pipe into a larger flowing pipeline, so the compressor discharge pressure is more or less dictated by the pressure in the large pipeline. Friction losses have therefore a very small effect.

In an integrative relationship, which exists for example in storage applications (Kurz and Brun [4]), the compressor fills a large cavity. That means, the compressor discharge pressure is increased as a function of the cumulative flow into the cavity. Similar conditions can be found in gas gathering applications where (on a much slower scale) the field pressure declines as a function of the cumulative flow out of the gas field. These fields additionally also have a strong head-flow relationship, i.e., increasing the flow at any given time would lower the compressor suction pressure.

Further, the transient system behavior must be considered. For example, a pipeline can be operated in a transient condition by feeding more gas into the pipeline than what is taken off on the other end. This is usually referred to as line packing. In general, pipelines are operated under slowly changing operating conditions (*Figure 11*). While a pipeline under steady state conditions requires a unique station pressure ratio for a given flow, this is no longer true under transient conditions: If the pipeline operates under transient conditions, for example during line pack after a fast increase in driver power, or, if one of the compressors has to be shut down, the steady state relationships are no longer valid (*Figure 12*). Dynamic studies of pipeline behavior reveal a distinctly different reaction of a pipeline to changes in station operating conditions than a steady state calculation. In steady state (or, for slow changes), pipeline hydraulics dictate an increase in station pressure ratio with increased flow, due to the fact that the pipeline pressure losses increase with increased flow through the pipeline. However, if a centrifugal compressor receives more driver power, and increases its speed and throughput rapidly, the station pressure ratio will react very slowly to this change. This is due to fact that initially the additional flow has to pack the pipeline (with its considerable volume) until changes in pressure become apparent. Thus, the dynamic change in operating conditions would lead (in the limit case of a very fast change in compressor power) to a change in flow without a change in head. If the power setting is maintained, the compressor operating point would then start to approach the steady state line again, albeit at a higher speed, pressure ratio, flow, and power.
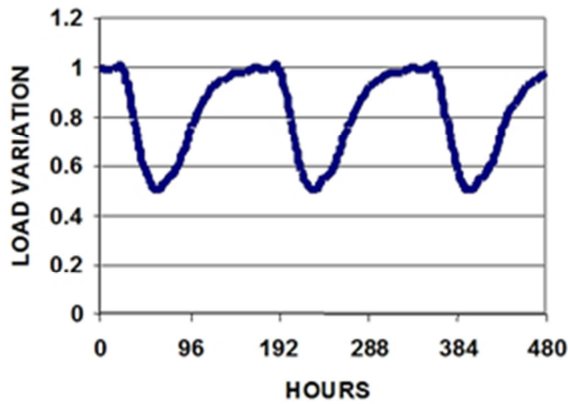
Figure 11. Averaged load variation for four stations of an interstate pipeline in South America during a 20-day period
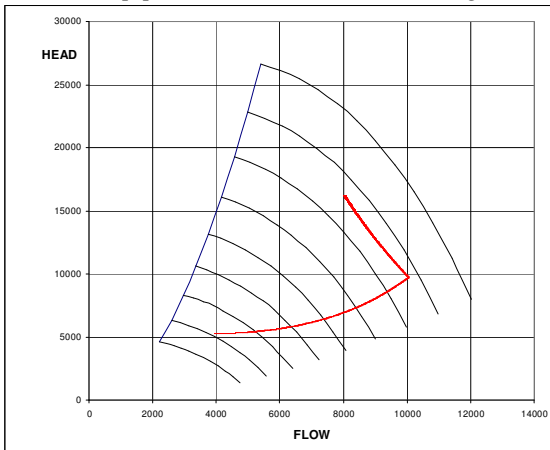

Figure 12. Typical Operating points if transient conditions are considered, in this case due to a fast engine acceleration from 50% to 100% load [5]

In general, changes are slow enough that the steady state compressor maps remain valid. However, for very fast changes, such as they occur during station emergency shutdown conditions, this may only be approximately true (Blieske et al [3], Morini et al [6]).

Installations capable of operating at variable compressor speed are often not controlled by prescribing the operating speed. When driven by two shaft gas turbines, for example, the power setting of the gas turbine and the system characteristic will determine the speed and operating point of the driven compressor. *Figure 13* illustrates this for a pipeline system. This figure also shows the effect of changing the pipeline characteristic, for example by looping the pipeline. Alternatively, compressors can be controlled by pre-scribing discharge pressure, suction pressure, or flow. In each case, the controller will adjust the fuel flow (and thus the power output) to automatically keep the unit at the prescribed operating condition.

Centrifugal Compressors, when driven by two shaft gas turbines, are usually adapted to varying process conditions by means of speed control. This is the most natural way of controlling a system, because both the centrifugal compressor and the power turbine of a two shaft gas turbine can operate over a wide range of speeds without any adverse effects. A typical configuration can operate down to 50% of its maximum continuous speed, and in many cases even lower.

Reaction times are very fast, thus allowing a continuous load following using modern, PLC based controllers.

As an example, discharge pressure control is described: The discharge pressure is sensed by a transmitter. A pressure set point is selected by the operator. If the discharge pressure decreases due to process changes, the controller will increase the fuel flow into the gas turbine. As a result the power turbine will produce more power and cause the power turbine, together with the driven compressor, to accelerate. Thus, the compressor discharge pressure is maintained. From *Figure 13*, it can be seen that both the power turbine speed (and thus the compressor speed) and the power increase in that situation.

Similar control mechanisms are available to keep the flow constant, or to keep the suction pressure constant. Another possible control mode is to run the unit at maximum available driver power (or any other, constant driver output). In this case, the operating points are on a line of constant power in *Figure 13*. The control scheme works for one or more compressors, and can be set up for machines operating in series as well as in parallel.
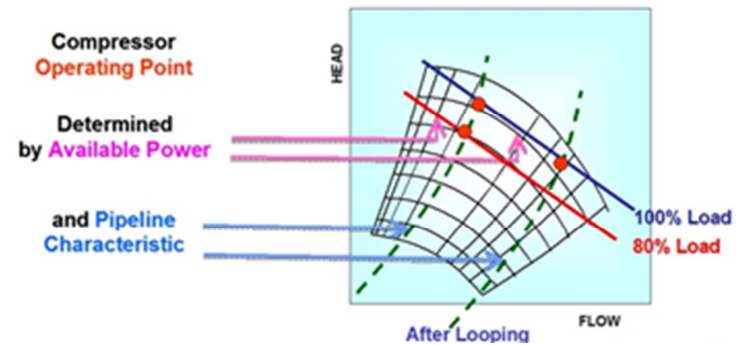

Figure 13. Available power, compressor characteristic and pipeline characteristic

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Dynamic Behavior

We have so far discussed the behavior of compressors to achieve process changes. In general, these lead to relatively slow changes in process conditions, so inertia effects of the turbomachinery can be neglected, that is, process changes can be treated as a series of steady state operating points.

This is not the case for certain effects related to emergency shutdowns and other, fast process upsets. A surge control system is set up to prevent the compressor from entering surge (*Figure 14*).

A very critical situation arises upon emergency shutdown (ESD). Here, the fuel supply to the gas turbine driver, or the electricity supply to the electric motor is cut off instantly, thus letting the power turbine and the driven compressor coast down on their own inertia[1] (*Figure 15*). Because the head-

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[1] Some installations maintain fuel flow to the turbine for 1 to 2 seconds while the recycle valve opens. However, this can generate a safety hazard.

making capability of the compressor is reduced by the square of its running speed, while the pressure ratio across the machine is imposed by the upstream and downstream piping system, the compressor would surge if the surge valve cannot provide fast relief of the pressure. The deceleration rate of the compressor as a result of inertia and energy dissipation is a decisive factor. The speed at which the discharge pressure can be relieved depends on the reaction time of the valve, and also on the time constants imposed by the piping system. The transient behavior of the piping system depends largely on the volumes of gas enclosed by the various components of the piping system, which may include, besides the piping itself, various scrubbers, knockout drums, and coolers (Kurz and White [7]). Another factor is the closing time of the downstream check valve (Botros [8])
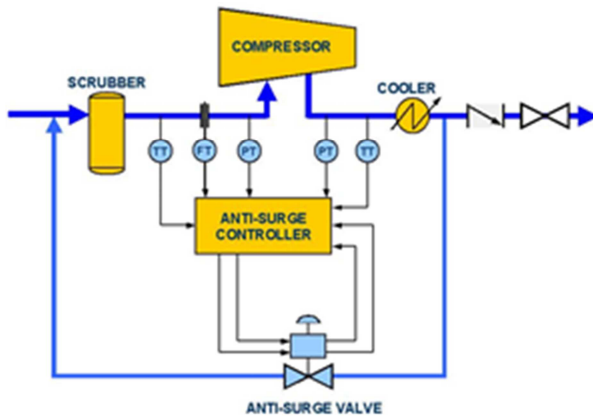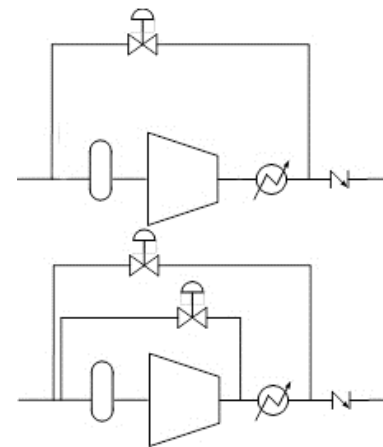

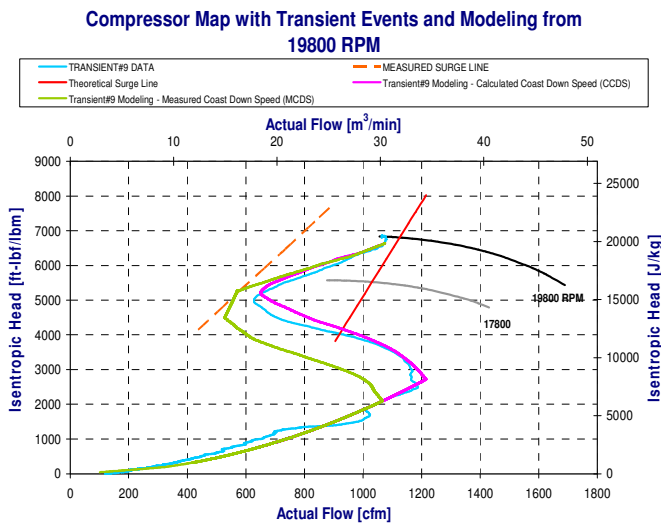*Figure 14. Anti-surge control system*

Figure 15. Compressor behavior during an emergency shutdown [9]

Part of the challenge for the surge avoidance/ recycle system is, that it really has three separate functions:

- Process control (see above)
- Surge prevention during rapid process changes, such as emergency shutdown.
- Allowing the startup of the compressor without surge and overheating.

The first task essentially requires a certain, preset position of the recycle valve. The second task requires a valve that can

be precisely controlled to allow exactly the right amount of gas to be recycled when the process flow becomes low, and which also allows for a bumpless transition when the recycle valve opens.

The third task requires a large fast acting valve that can open very quickly, to relief the discharge side pressure. The second and the third tasks therefore have contradicting requirements for the valve. In many cases, a compromise can be found, allowing a single valve installation. In other cases, especially for high pressure applications with compressor having a low inertia, or systems where the pipe volume downstream of the compressor, that is between compressor discharge, recycle valve and check valve, is large, two loops have to be installed *Figure 16*). Botros [10] has discussed considerations that allow the estimation whether a single or dual loop system may be required.

Figure 16. Single loop (top) and dual loop (bottom) control system (Botros, [10])

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 that the peak head capability. This is a particular issue in systems with low frequency pulsations (Kurz et al [11]). 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 5*). Changing the operating point of a compressor always involves a change in incidence angles for the aerodynamic components. Just as with an Airfoil (*Figure 5*), increasing the incidence angle will eventually lead to a 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 [12]).

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 (*Figure 14*). A typical system consists of pressure and temperature transmitters on the compressor suction and discharge lines, a flow differential pressure transmitter across the compressor flow meter, 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 a 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.

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. 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 12*).

The first two models on the left of *Figure 17* involve speed as a parameter. 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, using the pressure at the compressor or flow meter inlet (S) and discharge (D).
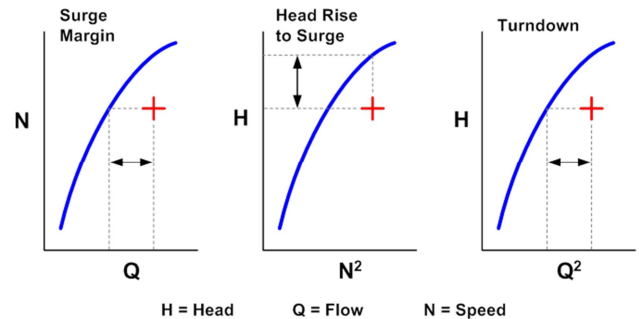

Figure 17. Surge limit in different systems of reference

$$H_p = K \cdot \frac{\left(\frac{P_D}{P_S}\right)^\sigma - 1}{\sigma} \cdot T \cdot SG \cdot Z \tag{8}$$

This is the basic equation for head. In the head across the compressor and the head across the flow meter 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.

$$H_{REDUCED} = \frac{\left(\frac{P_D}{P_S}\right)^\sigma - 1}{\sigma} \tag{9}$$

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 there is less concern 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. *Figure 18* 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.
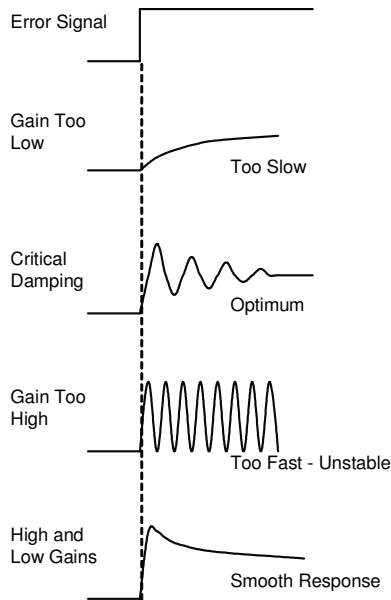

Figure 18. Reaction of a control system to an error signal

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 on line 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 made recently. However, the response of the gas volume in the pipe 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 differential pressure (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 (see *Figure 14*).

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 flow meter developing a 100 inch H_2O signal and a 50% PPL flowing 100 MMSCFD (50 lb/sec) is equivalent to about 20 hp.

As noted, strong signal devices are highly preferred. Pitot type flow meters have a relatively low signal level, around 25 inches H_2O. In the middle are orifices and venture type flow meters with a moderate signal of around 100 inches H_2O. Compressor suction-to-eye provides a strong signal, around 700 inches H_2O with the added benefit of not causing any additional pressure loss.

Suction-to-eye uses the inlet shroud or inlet volute of the compressor as a flow-measuring device (*Figure 19*). 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 H_2O). 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.
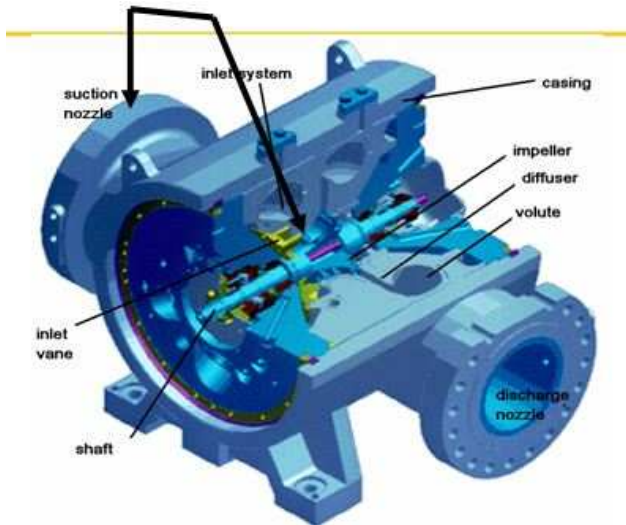

Figure 19. Suction to-eye flow metering

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 1 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%. The final control elements (recycle valves) probably can resolve 1% changes in their command signals; hence, the process variables should be resolved to at least 0.1% (10 bits) of their normal operating range. Over-ranging transmitters degrades resolution.

The Surge Control Valve

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 20**): 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 towards the end of the valve's travel. All three types

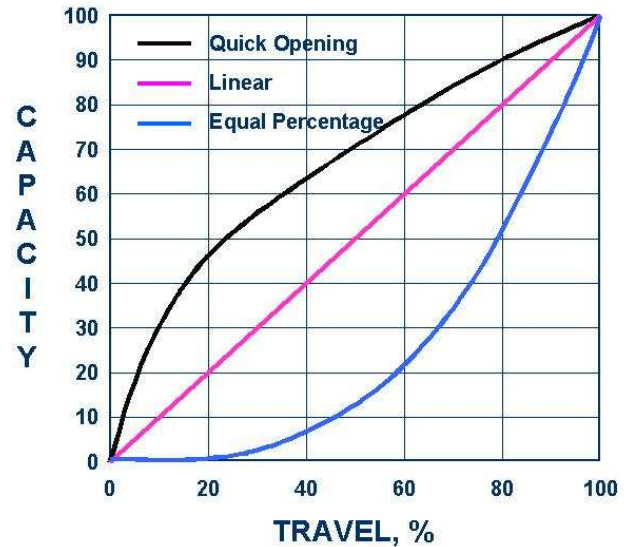of valve have been used in various configurations as recycle valves.

Figure 20. Valve characteristics

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 21** 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 (C_v) or capacity. Also, it is more reliable as it is less susceptible to fouling and improper maintenance.
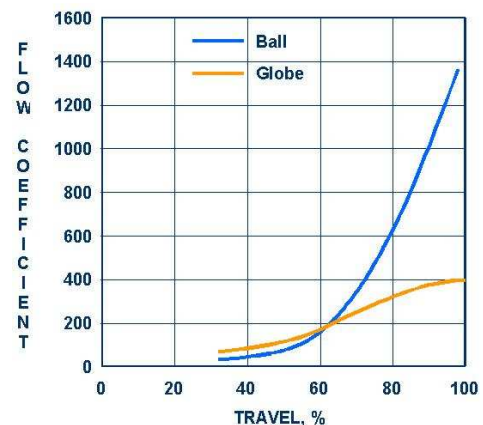

Figure 21. Ball and globe valves compared

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% 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. The impact of valve stick and slip is addressed by using a deadband at the control line.

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 effective is the hot and cold recycle configuration (*Figure 22*). 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.
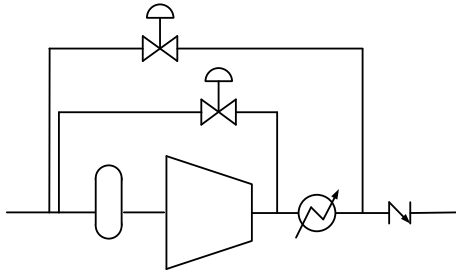

Figure 22. Hot and cold recycle valve arrangement

An alternate to this configuration is having a second cooled valve in parallel with the first (*Figure 23*). 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% and 10% surge margin. Under normal movements of the operating conditions only the 10% surge margin valve (primary valve) will open. If movement is fast enough to push the operating point down to 9% 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 become the primary valve. This change can be made without taking the compressor off line.
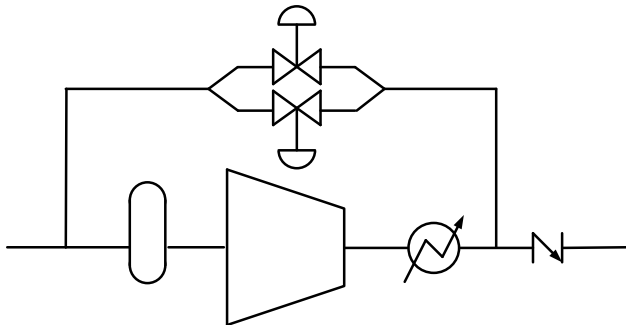

Figure 23. Parallel recycle valves

The advantages of the two parallel valves do not come without a price. In normal operation 2% to 5% 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 2 or 3 times the normal flow. This will result in 4 to 9 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. (See the Review of Piping 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% travel equals 50% capacity. For quick opening valves, capacity approximately equals the square root of travel. As such the two 4 inch valves will have 70.7% of their fully open capacity at 50% open. Comparing the two arrangements 250 ms after the shutdown is initiated, the two 4 inch quick opening valves will have 56% 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 ranged will have the same Cv as the 6 inch linear at 25% travel.

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.
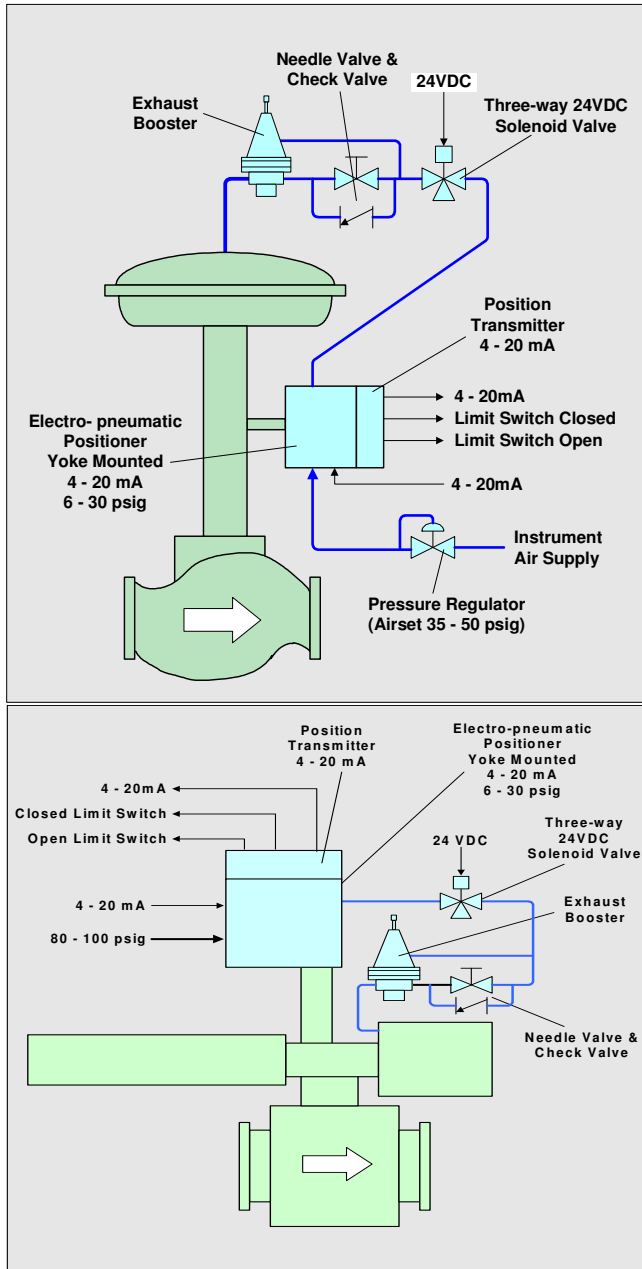

Figure 24. Globe (top) and ball (bottom) valve configurations

Standard industry quick-exhausts are not recommended for this application. They have a high threshold for opening (typically 2 – 4 psid) and an equally high threshold for re-closing. 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.

Figure 24 shows globe and ball valves with their preferred instrumentation configurations.

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 is 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 capacity or C_v from that data.
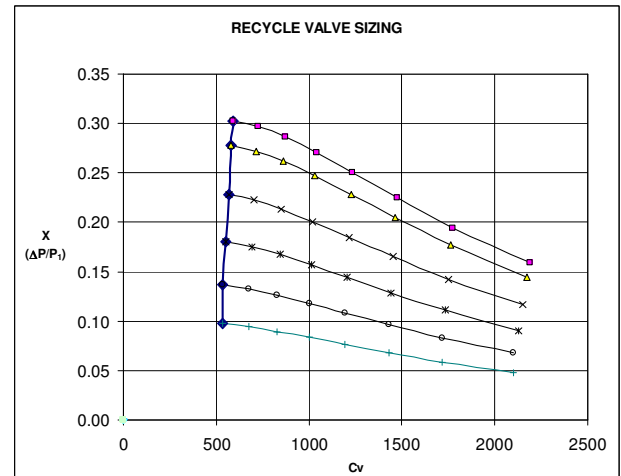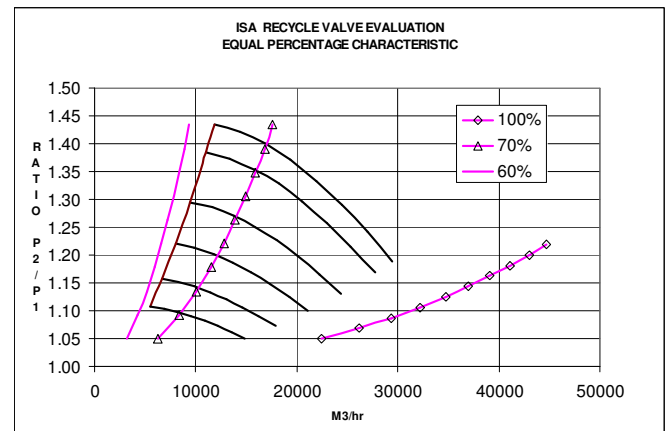

Figure 25. Almost constant C_v at the surge limit

Figure 26. Valve matched to compressor map

Typically the surge limit of a compressor equates to a single valve capacity or C_v (Figure 25). The valve can be selected based on valve 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 26 shows such a valve with its flow characteristic when 60%, 70%, and 100% open, superimposed on the compressor map.

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 anti-surge 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 avoidance system is an Emergency Shutdown (ESD), particularly if the compressor is already operating close to surge when the driver 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 counter-acted by the inertia of the rotor system. *Figure 27*, which displays data based on test data and theoretical considerations, indicates a 30% drop on compressor speed within the first second after shutdown. A 30% loss in speed equates to approximately a loss in head of about 50%. The valve must, therefore, reduce the head across the compressor by about half in the same time as the compressor loses 30% of its speed.
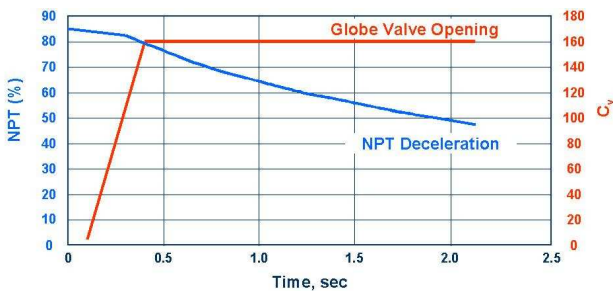

Figure 27. Compressor deceleration (NPT) and globe valve opening

The larger the volumes in the system, the longer it will take to equalize the pressures. The larger and faster the valve, the better is its potential to avoid surge. In general, large valves react slower than small valves, so a detailed study is required to find the optimum size. 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 22*), where the recycle loop does not include the after cooler.

While the behavior of the piping system can be predicted quite accurately (assuming all valve reaction times, including the check valve, the cooler loss in transient situations, etc are properly modelled), 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 procedure is outlined further below.

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 28). The volume on the suction side is usually orders of magnitude larger than the discharge volume and, therefore, can be considered infinite, unless a check valve is employed. 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 28*).
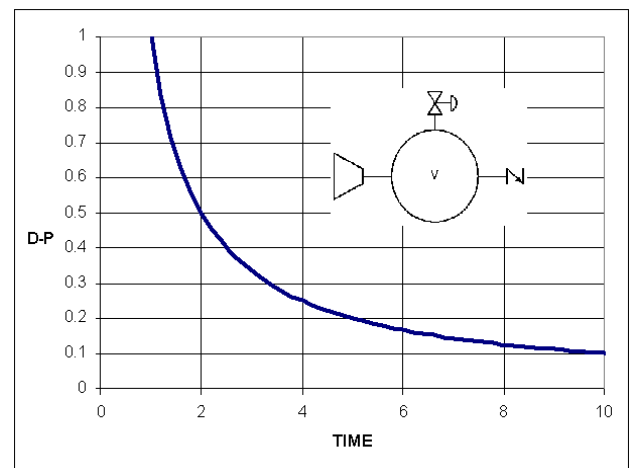

Figure 28. Simplified system and transient characteristic

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 d-p across 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 (Kurz and White [7]). 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 is reduced to:

$$\frac{dp_2}{dt} = \frac{k \cdot p_2}{V}\left[Q - Q_v\right]$$

(10)

The rate of flow through the valve is calculated with the standard ISA method [13][2]:

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

and

$$Q_v = Q_{std} \frac{\rho_{std}}{\rho(p_2, T_2, Z_2)} \qquad (11)$$

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 [7] have shown the validity of the simplified model. However, details of the system are omitted and this will impact the accuracy of results. For every time step, the compressor discharge and suction pressure are calculated based on the system response. The compressor will react by providing a certain amount of flow compatible with its current speed and the pressures in the system. For this state, the compressor will absorb a certain amount of power, defined by head, efficiency, and flow.

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:

$$T = -2\pi \cdot J \cdot \frac{dN}{dt} \qquad (12)$$

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:

$$P = T \cdot N \cdot 2\pi = -(2\pi)^2 \cdot J \cdot N \cdot \frac{dN}{dt} \qquad (13)$$

If the rundown would follow through similar operating points, then $P \sim N^3$, which would lead to a rundown behavior of:

$$\frac{dN}{dt} = \frac{k}{J} \frac{N^2}{(2\pi)^2} \rightarrow \int N^{-2} dN = \frac{k}{J(2\pi)^2} \int dt + c \rightarrow N(t) = -\frac{1}{-\frac{k}{J(2\pi)^2} t - \frac{1}{N_{t=0}}} \qquad (14)$$

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.

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 % speed in the first second of deceleration. This is, for example, confirmed by data from Kurz and White [7] showing a 30% speed reduction of a gas turbine driven compressor set, and Bakken et. al. [14], where the gas turbine driven configurations lost about 20-to-25% speed in the first second, while the electric motor driven configuration lost 30% speed in the first second. As a result of the loss of 25% speed, the head the compressor can produce at the surge line is about 56% 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 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, one 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.

With simplified calculations the maximum allowable piping volume for the set parameters that will not cause surge at ESD can quickly be evaluated.

A more comprehensive approach which allows the modelling of the exact piping geometry as well as detailed features of the compressor and the valves, including the check valves, should be used in particular in situations where the results from simpler studies indicate potential problems.

Recycle Arrangements for Specific Applications

The arrangement of recycle loops impacts the operational flexibility, as well as the start-up and transient behavior of a compressor station. In the following section, various arrangement concepts (*Figure 29* to *Figure 37*) are described, together with their basic advantages and disadvantages. *Figure 29* to *Figure 34* describe single compressor systems, while *Figure 35* to *Figure 37* show some configurations for systems with multiple compressors.
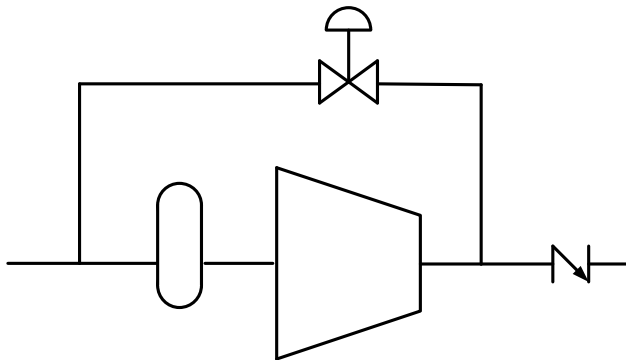

Figure 29. The basic recycle system (hot recycle)

Basic Recycle System (*Figure 29*)

+ Small discharge volume, fast recycle response

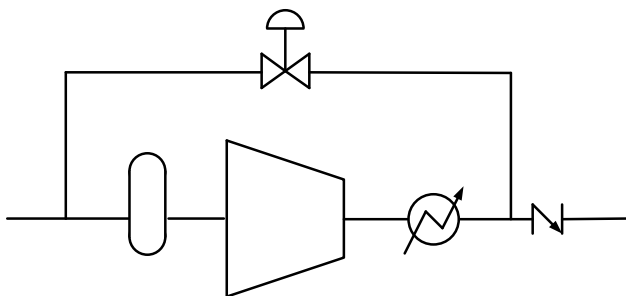- Although some partial recycle can be maintained, 100% recycle cannot.

Figure 30. After-cooler inside recycle loop

Aftercooler inside Recycle Loop (*Figure 30*)

+ 100% recycle possible

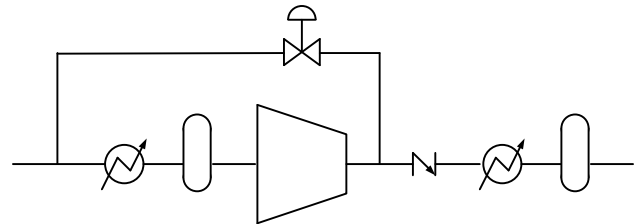- additional discharge volume impacts recycle response

Figure 31. Pre-Cooler & Post Cooling

Pre-Cooler and Aftercooling (*Figure 31*)

+ Small discharge volume, fast recycle response

+ if there is significant heat in the suction header may improve compressor performance
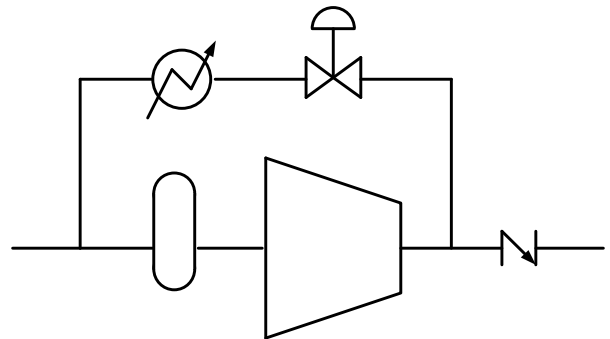
- requires an additional cooler

Figure 32. Recycle cooling

Cooled Recycle Loop (*Figure 32*)

+ Small discharge volume, fast recycle response

+ No in-line pressure loss

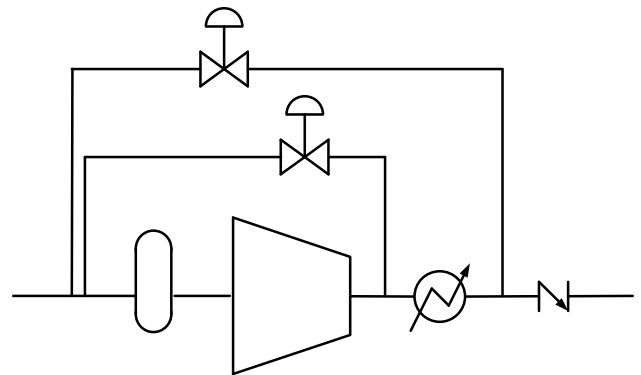- Requires an additional cooler, although smaller than a pre-cooler

Figure 33. Hot & cooled recycle valves

Hot Recycle Valve and Cooled Recycle Valve (*Figure 33*)

+ Provides modulating surge control valve and shutdown valve ideally suited for their purpose.
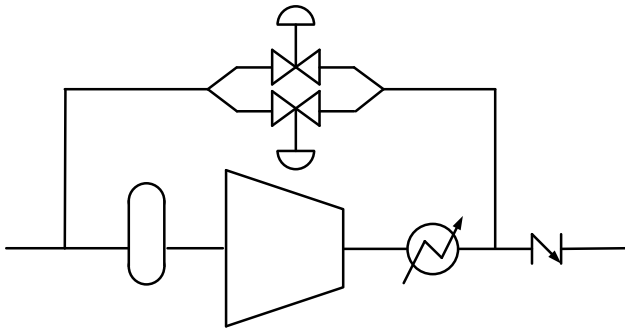
- More components, more cost

Figure 34. Parallel recycle valves

Parallel Recycle Valves (*Figure 34*)

+ Provide good modulating surge control and fast shutdown valves.

+ Provide some level of redundancy
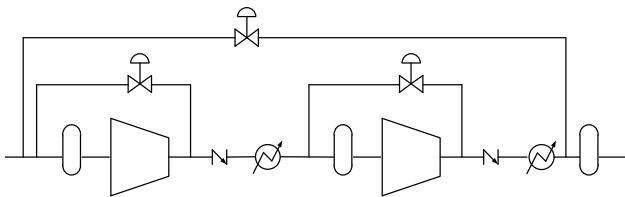
- More components, more cost

Figure 35. Hot unit valves cooled overall recycle valve

Multiple compressors with Hot Unit Valves and Cooled Overall Recycle Valve (*Figure 35*)

+ Provides good modulating surge control and fast

shutdown valves.

+ Provides some level of redundancy

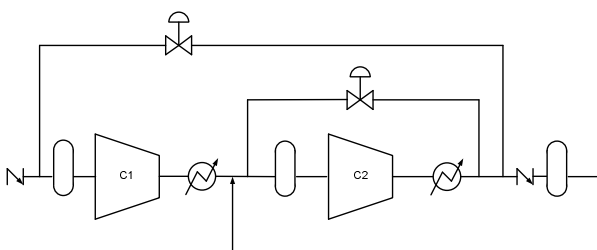- More components, more cost

Figure 36. Recycle valve for second larger compressor and overall recycle valve

Multiple Compressors, Compressor 2 Significantly larger than Compressor 1: Recycle Valve for Second Compressor and Overall Recycle Valve (*Figure 36*)

+ Provides good modulating surge control and fast shutdown valves.

+ Provided some level of redundancy
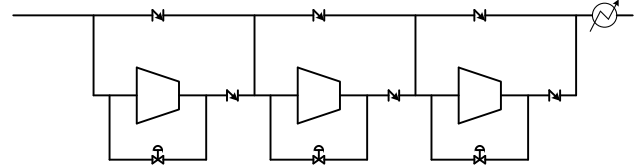
- More components, more cost

Figure 37. Low ratio compressors in series: Recycle for individual compressor

Multiple, low ratio (low discharge temperature) individual compressors connected in Series (*Figure 37* and *Figure 38*)
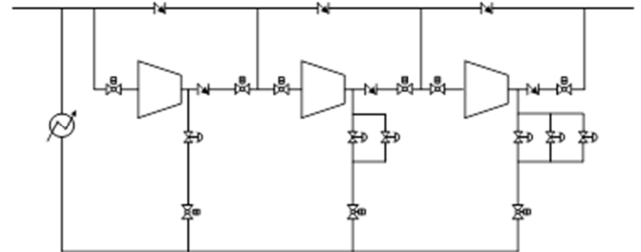

Figure 38. Low ratio compressors in series: Common recycle header

In either configuration any compressor can be started and brought on line 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 38*), 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 is shut down. This in turn requires added flow capacity in the recycle valve. This configuration also requires a number of additional block valves.

START-UP CONSIDERATIONS

The design of the anti-surge and recycle system also impacts the start-up 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 14*), and gas is flowing into the pipeline.

Virtually all of the mechanical energy absorbed by the compressor is converted into energy in the discharged gas. In an un-cooled recycle system, this energy, and the resulting heat is recycled into the compressor suction and then more energy added to it. At significant 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 after-coolers. There are several strategies that can be employed to avoid overheating the un-cooled compressor during start-up:

- 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 on line quickly to avoid overheating. Un-cooled 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 on line.

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.

Figure 39 and *Figure 40* outline a solution to a rather difficult starting problem for a compressor station without aftercooling capacity: For the original configuration (*Figure 39*), starting 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, for any additional unit 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 40*), the heat from the unit coming on line is mixed with the station suction flow. This equalizes the inlet temperature of all compressors; higher for the compressors already on line, lower for the compressor coming on line. With this arrangement overheating of any additional compressor coming on line is nearly always avoided.

Analysis of the start-up 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 one can determine 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 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.
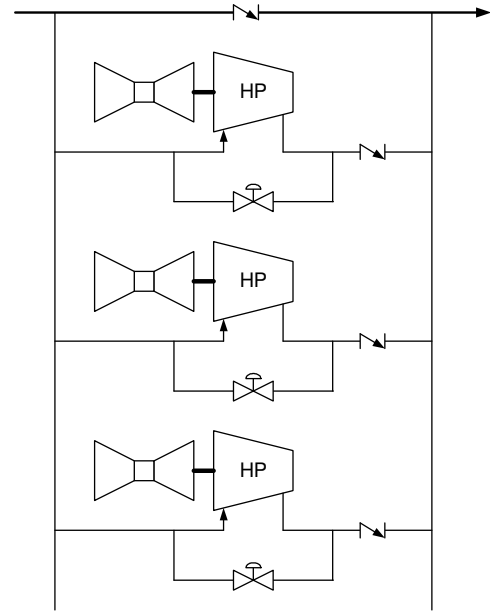

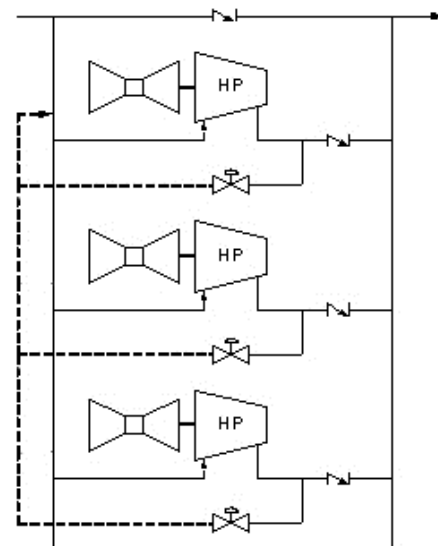Figure 39. Original station layout with 3 parallel units

Figure 40. Improved station layout

Figure 26 illustrates this: At 70% open setting, the startup of the compressor is significantly closer to the surge line than at 100% 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 start-up is virtually impossible, as the parameters defining the surge limit of the compressor are too low to be practically measured. Returning to *Figure 25* the surge limit of a compressor matches well with a fixed travel (constant Cv) line for a recycle valve. As such, a compressor can be started with a fixed recycle valve position.
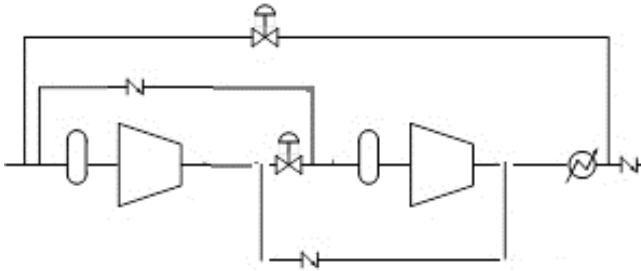
Figure 41: Compressors for series and parallel operation.

SERIES-PARALLEL CONSIDERATIONS

For some applications, for example for gas storage, it is advantageous to be able to switch two compressors from series to parallel operation and vice versa. In doing so, the operating range can be significantly increased. In series operation, the units can provide high head, while in parallel, the flow range is increased.

It is desirable to be able to switch from series to parallel operation, and vice versa, while the compressors are running at or near full load. This is possible with an appropriate arrangement of valves. The most elegant solution involves a control valve and two check valves (Figure 41). With the control valve open, the machines operate in series, while with a closed valve they operate in parallel. The check valve will automatically open and close based on the pressure differential over these valves, and therefore don't have to be controlled. In this arrangement, the opening and closing speed of the control valve is not particularly important.

The same can also be applied for casings with two compressor sections. In particular for back-to-back machines, the requirement to balance the thrust load can impose additional limitations.

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. [15], Blanchini et. al. [16]) or passive means (Arnulfi et. al. [17]). Other efforts try to increase the accuracy of determining the surge margin (McKee and Deffenbaugh [18]) by detecting the certain precursors of surge. However, most of the ideas will remain valid even if some of the new methods, still 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 anti-surge 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.

Because of the relations between compressor characteristic, control system, and system characteristic, successful application of the equipment requires the understanding of the behavior of all components. This allows the evaluation of the overall behavior of the compressor and its surrounding systems. Only with this knowledge, compressors can be adapted for optimum system performance.

NOMENCLATURE

A	Flow area
c,u,w	Velocity
C_v	Flow coefficient ($c_v = Q/\sqrt{SG/\Delta p}$)
C	Compressible valve coefficient
D	Diameter
F_p	Piping geometry factor
F	Force
h	Head
J	Inertia
k	Isentropic exponent
k	Constant
K_v	Valve coefficient
L	Pipe length
$\dot{m}$	mass flow
M	Momentum
N	Speed (1/s)
p	Pressure
Q	Volumetric flow
SG	Specific gravity
SM	Surge margin (%)
T	Temperature
T	Torque
t	Time
V	Volume
x,y	coordinates
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
u	circumferrential
v	Valve
1	inlet
2	outlet

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