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TURBOMACHINERY CONTROL VALVES SIZING AND SELECTION

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ABSTRACT

Turbomachinery Controls dedicated to centrifugal and axial compressors use several types of control valves, such as:

- Antisurge (recycle) valve
- Suction throttle valve
- Hot-gas bypass valve
- Quench control valve

As the final control element in its control loop, these control valves are vital to implementing good turbomachinery controls. This tutorial will examine the control objective of each type of valve, its ideal location relative to the turbocompressor and the optimum performance characteristics for the valve. Valve selection criteria and sizing methodologies with examples will be addressed. Recommendations for valve noise abatement will be provided, as well as valve noise-abatement pitfalls that should be avoided will be identified.

INTRODUCTION

All turbomachinery control systems use control valves as final control elements. The control valve manipulates a flowing fluid, such as steam, gas or vapor, or a liquid, to compensate for the load disturbance and keep the desired control variable as close as possible to the desired set point. When we refer to a "control valve" we are actually referring to a "control valve assembly", that includes:

- the valve body and its trim,
- a suitable actuation system to provide the required motive power,
- the other necessary components such as the positioner and/or the electro-pneumatic transducer (converter),
- and some useful accessories such as position transmitters, limit switches, …. etc.

Since this is not a tutorial about control valves in general, and limited in scope to the control valves commonly used in turbomachinery controls, we shall only address each turbomachinery controls requirement for a control valve and expand on that.

PART 1: ANTISURGE VALVES

General

This is the final control element for the antisurge control loop. When process conditions force the compressor (stage) to operate with low flowrates, and to ensure that the compressor always handles more flow than the surge value, the antisurge control valve is opened when necessary to allow the gas delivered by the compressor to either be recycled, or blown-off to the atmosphere. When the gas being compressed is recycled via a control valve, it may be called a spillback, kickback or recycle valve. When the gas being compressed is air or nitrogen, antisurge control is not usually done via recycling discharge gas back to the suction – as this would require a cooling system to remove the heat of compression – but rather by blowing off into the atmosphere. In these cases, the antisurge valve is commonly called a blow-off valve. See Figure 1.

Centrifugal or Axial Compressor Surge

Basically, for a given speed of rotation, if the process resistance that is perceived at the compressor discharge flange rises to a value that exceeds the compressor's capacity to generate head – the motive force to push the gas forwards – then the compressor will surge. To prevent this, the antisurge valve is opened, so as to reduce the resistance felt at the discharge flange of the compressor, and ensure that the gas continues to move forward even if it has to be recycled or blown off to the atmosphere.

Basis for Sizing Antisurge Control Valves

Heuristically, it is logical to base the antisurge valve required capacity on the surge flow characteristics of the compressor in question. This would establish a clear and logical connection between the minimum forward flow that needs to be ensured through the compressor and the capacity (C_V) of the antisurge valve to deliver that required flow. Basing the sizing of the antisurge valve on any other characteristic (such as the design point of the compressor, or a process flow requirement, …. etc.) would clearly break the connection to the surge flow value of the compressor, and while this might produce a workable outcome in certain conditions, this approach will fail to produce satisfactory outcomes in other operating conditions.

If it accepted that sizing the antisurge valve needs to be based on compressor surge characteristics, then it follows that deriving the antisurge valve sizing parameters may be based on the supplied compressor data sheets and performance curves. However, the compressor configuration will dictate the parameters used for sizing the control valve. We shall examine some common compressor configurations.

Single Stage Compressors or Compressor Sections

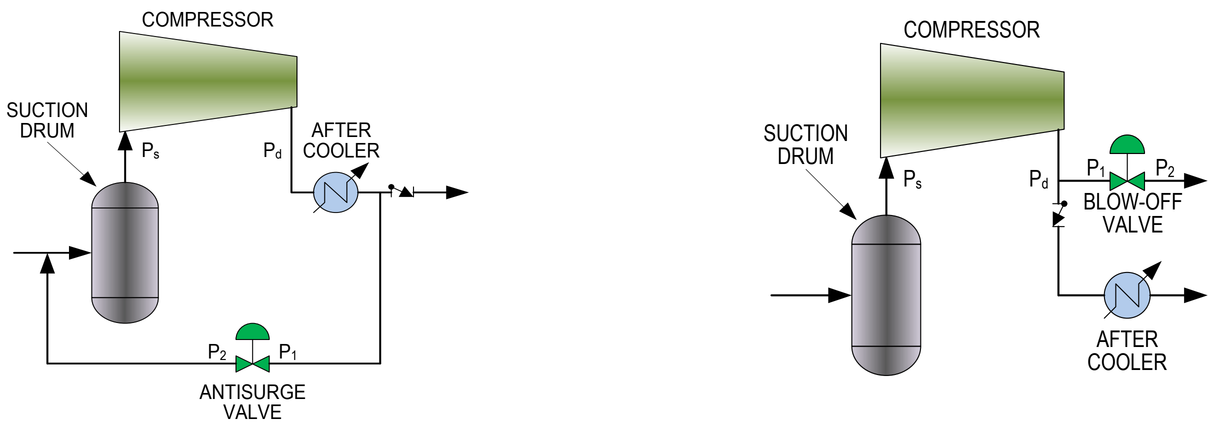

Figure 1 – Example of a Single Compressor or Compressor Section

The performance curves associated with these types of compressors can be:

- Single fixed-speed performance curve, and,
- Multiple performance curves (variable speed, variable inlet guide vanes, or IGVs, or variable suction throttle valve opening).

Single Fixed-speed Performance Curve

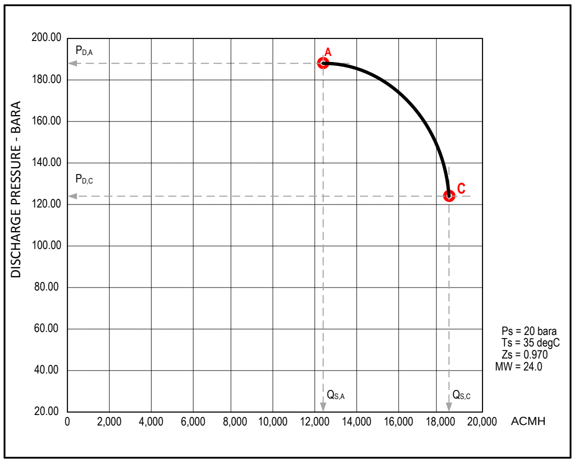

Figure 2 – Example of a Fixed-speed Compressor Performance Curve

For the compressor whose performance is characterized by a single fixed-speed performance curve as shown Figure 2, the single surge point (A) is considered; with its associated surge point suction flow, $Q_{s,A}$ and surge point discharge pressure $P_{d,A}$. Once a suitable antisurge control valve capacity (C_v) is determined, it should be compared to the C_v of the choke point (C); with its associated choke point suction flow, $Q_{s,C}$ and choke point discharge pressure $P_{d,C}$. The antisurge valve sizing parameters would then be:

P_1 = Valve inlet pressure =	Compressor discharge pressure (P_d) minus appropriate piping losses between compressor discharge and valve inlet
P_2 = Valve outlet pressure =	For a recycle layout: compressor suction pressure (P_s) plus appropriate piping losses between compressor suction and valve inlet
	For a blow-off valve: atmospheric pressure plus appropriate pressure drop for a stack-mounted silencer
T_1 = Valve inlet temperature =	Compressor discharge temperature (T_d) minus appropriate temperature drops between compressor discharge and valve inlet
Z_1 = Valve inlet compressibility =	Compressor discharge compressibility (Z_d) at discharge pressure and temperature
k_1 = Valve inlet specific heat ratio =	Compressor discharge specific heat ratio (k_d) at discharge pressure and temperature
MW = Valve inlet molecular weight =	Compressor molecular weight

The required antisurge or blow-off valve C_V is between 1.8 and 2.2 times the surge point C_V, with the further requirement that this should not exceed the choke point C_V.

In the many years of experience of the author's company, the antisurge valve sizing that provides the most suitable dynamic response to surge-inducing upsets to the compressor would be approximately twice the capacity required to operate the compressor at the surge point, with a practical tolerance of about 10% in either direction, hence between 1.8 and 2.2 times the surge point C_V.

Example

For the compressor performance curve depicted in above Figure 2, and assuming there are no significant pressure losses between the antisurge valve and the compressor suction, the following antisurge valve parameters may be derived:

$Q_{s,A}$	= Surge point suction volumetric flow rate	= 12,200 ACMH
$P_{d,A}$	= Surge point discharge pressure	= 190.0 bara
$Q_{s,C}$	= Choke point suction volumetric flow rate	= 18,500 ACMH
$P_{d,C}$	= Choke point discharge pressure	= 123.0 bara
T_{AC}	= Aftercooler outlet temperature	= 35.0 degC
ΔP_{AC}	= Aftercooler pressure drop	= 2.0 bar

Parameter			Surge Point	Choke Point
P_1	Valve inlet pressure	bara	188.0	121.0
P_2	Valve outlet pressure	bara	20.0	20.0
T_1	Valve inlet temperature	degC	35.0	35.0
Z_1	Valve inlet gas compressibility	---	0.900	0.930
k_1	Valve inlet specific heat ratio	---	1.25	1.25
MW	Valve inlet molecular weight	---	24.0	24.0

Based on the above parameters, and assuming a globe valve with a pressure drop ratio factor (x_T) of 0.75, the calculated valve capacity at the surge point is 82.5, and at the choke point is 197.6. Therefore, an antisurge valve with a full-open capacity of between 148.5 and 181.5 is required for adequate surge control. Note that this range of valve C_V values is less than the choke point C_V.

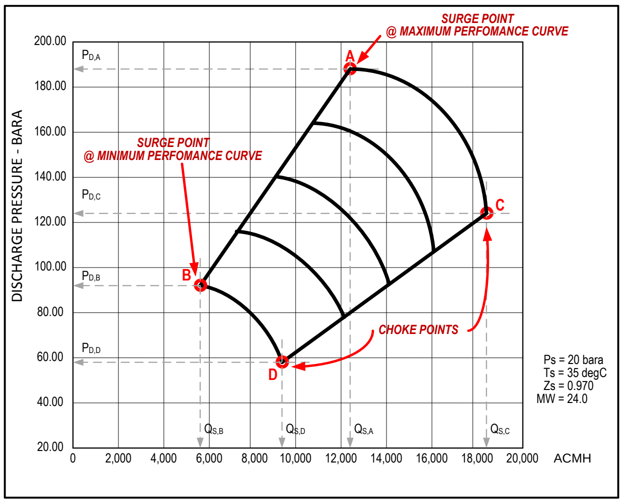

Figure 3 – Example of Variable Compressor Performance Curves

When the compressor performance is characterized by a family of variable performance curves as shown in Figure 3, two surge points are considered:

- The maximum curve's surge point suction flow, $Q_{s,A}$ and associated maximum surge point discharge pressure $P_{d,A}$; and,
- The minimum curve's surge point suction flow, $Q_{s,B}$ and associated minimum surge point discharge pressure $P_{d,B}$

Once a suitable antisurge control valve capacity (C_v) is determined, it should be compared to the C_v of two choke points:

- The maximum curve's choke point suction flow, $Q_{s,C}$ and associated maximum surge point discharge pressure $P_{d,C}$; and,
- The minimum curve's choke point suction flow, $Q_{s,D}$ and its associated minimum surge point discharge pressure $P_{d,D}$

Example

For the compressor performance curve depicted in above Figure 3, and assuming there are no significant pressure losses between the antisurge valve and the compressor suction, the following antisurge valve parameters may be derived:

$Q_{s,A}$	= Maximum surge point suction volumetric flow rate	= 12,200 ACMH
$P_{d,A}$	= Maximum surge point discharge pressure	= 190.0 bara
$Q_{s,B}$	= Minimum surge point suction volumetric flow rate	= 5,500 ACMH
$P_{d,B}$	= Minimum surge point discharge pressure	= 93.0 bara
$Q_{s,C}$	= Maximum choke point suction volumetric flow rate	= 18,500 ACMH
$P_{d,C}$	= Maximum choke point discharge pressure	= 123.0 bara
$Q_{S,B}$	= Minimum choke point suction volumetric flow rate	= 9,500 ACMH
$P_{D,D}$	= Minimum choke point discharge pressure	= 58.0 bara
T_{AC}	= Aftercooler outlet temperature	= 35.0 degC
ΔP_{AC}	= Aftercooler pressure drop	= 2.0 bar

Parameter			Maximum Surge Point	Maximum Choke Point	Minimum Surge Point	Minimum Choke Point
P_1	Valve inlet pressure	bara	188.0	121.0	91.0	56.0
P_2	Valve outlet pressure	bara	20.0	20.0	20.0	20.0
T_1	Valve inlet temperature	degC	35.0	35.0	35.0	35.0
Z_1	Valve inlet gas compressibility	---	0.900	0.930	0.950	0.960
k_1	Valve inlet specific heat ratio	---	1.25	1.25	1.25	1.25
MW	Valve inlet molecular weight	---	24.0	24.0	24.0	24.0

Based on the above parameters, and assuming a globe valve with a pressure drop ratio factor (x_T) of 0.75, the calculated valve capacity at the maximum surge point is 82.5, and at the minimum surge point is choke point is 78.9. The higher of these two values is then selected to represent the surge point C_V. Therefore, an antisurge valve with a full-open capacity of between 148.5 and 181.5 is required for adequate surge control. Also, the calculated valve capacity at the maximum choke point is 197.6 and the calculated valve capacity at the minimum choke point is 222.8. The lower of these two values is selected to represent the choke point C_V. Note that range of selected valve C_V values is less than the choke point C_V.

Multi-stage Compressors or Compressor Sections

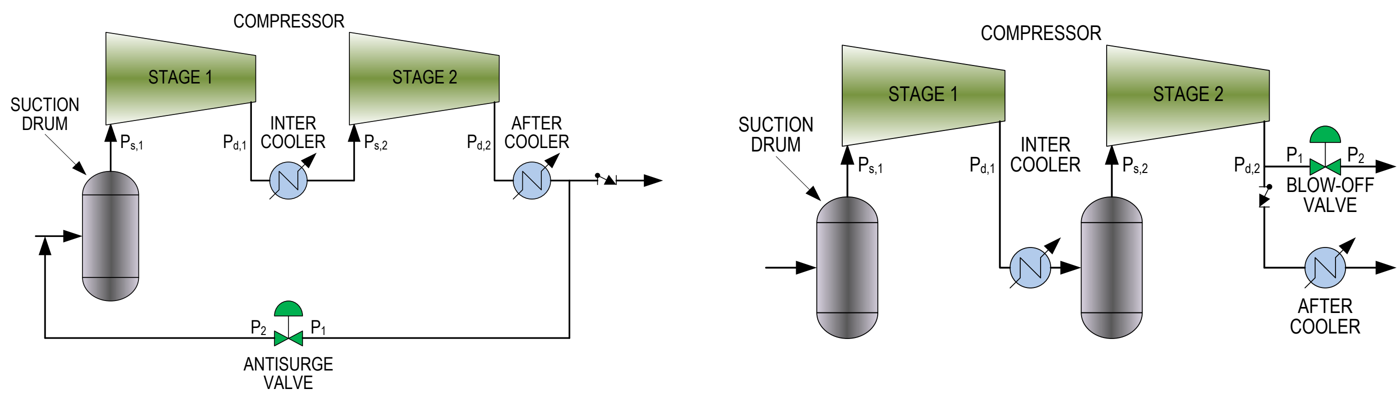

Figure 4 – Example of a Multi-Stage Compressor

When a single antisurge valve is required to provide recycle or blow-off, as in the above Figure 4, then the size of the common valve must cater to the C_V requirements of each of the compressor stages. It is thus more convenient to use the composite or "overall" performance curves for the multi-stage compressor, and apply a similar methodology as described previously for single and multiple curves. In this case, the antisurge valve sizing parameters would then be:

P_1	= Valve inlet pressure =	Compressor discharge pressure ($P_{d,2}$) minus appropriate piping losses between compressor discharge and valve inlet
P_2	= Valve outlet pressure =	For a recycle layout: compressor suction pressure ($P_{s,1}$) plus appropriate piping losses between compressor suction and valve inlet
		For a blow-off valve: atmospheric pressure plus appropriate pressure drop for a

T_1 = Valve inlet temperature = Compressor discharge temperature (T_d) minus appropriate temperature drops between compressor discharge and valve inlet

Z_1 = Valve inlet compressibility = Compressor discharge compressibility (Z_d) at discharge pressure and temperature

k_1 = Valve inlet specific heat ratio = Compressor discharge specific heat ratio (k_d) at discharge pressure and temperature

MW = Valve inlet molecular weight = Compressor molecular weight

As before, the selected antisurge or blow-off valve C_V should be between 1.8 and 2.2 times the surge point C_V, with the further requirement that this should not exceed the choke point C_V.

In some cases, the compressor manufacturer will supply multiple curves (for variable speed machines) for each individual stage. In this case, it is required to calculate the C_V requirement for each stage, and select an antisurge valve that meets the largest stage requirement.

It should be remembered that the individual stages are mounted on the same drive shaft, and hence they will rotate at the same speed, or relative speed if interstage gearboxes are used. In this case the compressor curves for each of the second and subsequent stages are valid only at inlet conditions that match the design point of the first stage. Hence, for each stage, a horizontal line is drawn through the design point and its intersection with the surge limit line and the choke line produce that stage's surge point and choke point for the antisurge valve requirements as shown in Figure 5 below.

Again, the selected antisurge or blow-off valve C_V should be between 1.8 and 2.2 times the surge point C_V, with the further requirement that this should not exceed the choke point C_V.

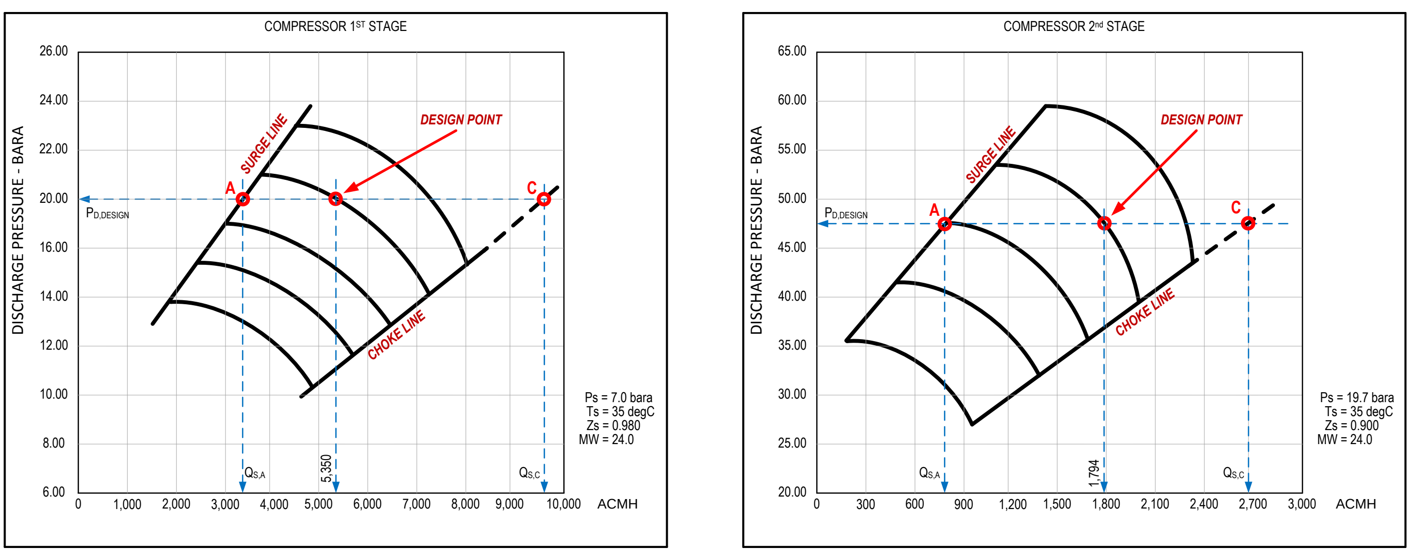

Figure 5 – Determining the Surge and Choke Points for Variable Speed Multi-Stage Compressors

For the compressor performance curve depicted in above Figure 5, and assuming there are no significant pressure losses between the antisurge valve and the compressor suction, the following antisurge valve parameters may be derived:

$Q_{s,A,1^{st}}$	= 1st stage surge point suction volumetric flow rate	= 3,300 ACMH
$Q_{s,A,2^{nd}}$	= 2nd stage surge point suction volumetric flow rate	= 800 ACMH
$Q_{s,C,1^{st}}$	= 1st stage choke point suction volumetric flow rate	= 9,800 ACMH
$Q_{s,C,2^{nd}}$	= 2nd stage choke point suction volumetric flow rate	= 2,700 ACMH
$P_{d,2}$	= 2nd stage design discharge pressure	= 47.5 bara
T_{AC}	= Aftercooler outlet temperature	= 35.0 degC
ΔP_{AC}	= Aftercooler pressure drop	= 1.0 bar

Parameter			1st Stage Surge Point	2nd Stage Surge Point	1st Stage Choke Point	2nd Stage Choke Point
P_1	Valve inlet pressure	bara	46.5	46.5	46.5	46.5
P_2	Valve outlet pressure	bara	7.0	7.0	7.0	7.0
T_1	Valve inlet temperature	degC	35.0	35.0	35.0	35.0
Z_1	Valve inlet gas compressibility	---	0.88	0.88	0.88	0.88
k_1	Valve inlet specific heat ratio	---	1.25	1.25	1.25	1.25
MW	Valve inlet molecular weight	---	24.0	24.0	24.0	24.0

Based on the above parameters, and assuming a globe valve with a pressure drop ratio factor (x_T) of 0.75, the calculated valve capacity at the 1st stage surge point is 30.9 and at the 2nd stage surge point is 23.0. The higher value is selected, and the required antisurge valve capacity (C_V) range is 55.6 ~ 68.0. Also, the 1st stage choke point C_V is 91.8, and the 2nd stage choke point C_V is 77.5. The lower value is selected to represent the compressor's choke point C_V. Therefore, the selected antisurge valve capacity will not exceed the choke point C_V.

Multi-stage Compressors with Induction Sidestream

Multi-stage compressors with side-streams are often used in refrigeration applications. In this type of compressor, the previous stage discharge flow is mixed with the admission sidestream flow, and the combined flow becomes the inlet flow to the next stage compressor stage. See Figure 6.

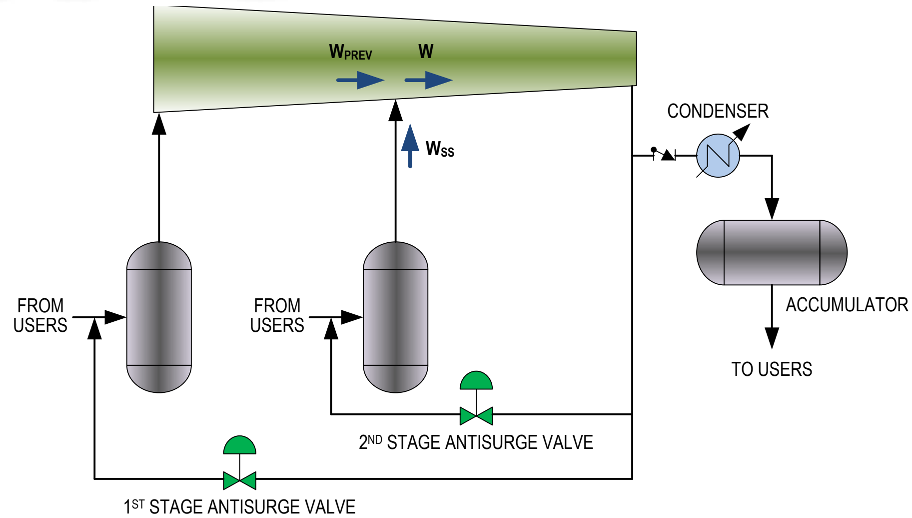

Figure 6 – Example of a 2-Stage Admission Sidestream Compressor

For the 1st stage, sizing its antisurge valve would proceed as per the methodology of a single stage compressor section, whether single speed or with multiple performance curves, as appropriate. However, the antisurge valve parameters would be:

P_1	= Valve inlet pressure =	Compressor final stage discharge pressure (P_d)
P_2	= Valve outlet pressure =	1st stage suction pressure (P_s)
T_1	= Valve inlet temperature =	Compressor final stage discharge temperature (T_d)
Z_1	= Valve inlet compressibility =	Compressor final stage discharge compressibility (Z_d)
k_1	= Valve inlet specific heat ratio =	Compressor final stage discharge specific heat ratio (k_d)
MW	= Valve inlet molecular weight =	Compressor molecular weight

For the 2nd stage with the sidestream flow, it is necessary to consider that its surge flow needs to be similar to the examples illustrated in above Figure 5, i.e. at the intersection of a horizontal line drawn through the design point and the surge limit line. It is also necessary to consider that internal flow from the previous stage is present, and credit must be taken for it. The minimum flow that the 2nd stage antisurge valve needs to provide may then be considered as:

$$W_{min} = 1.8 \cdot W - \left\{ \left[\left(0.8 \cdot \frac{W_{prev}}{W} \right) + 1 \right] \cdot W_{prev} \right\}$$

This minimum flowrate can then be used to calculate the minimum C_v required from the 2nd stage antisurge valve. In a similar manner, the maximum flow through the 2nd stage antisurge valve, used to calculate its maximum C_v value, may be considered as:

$$W_{max} = 2.2 \cdot W - \left\{ \left[\left(1.2 \cdot \frac{W_{prev}}{W} \right) + 1 \right] \cdot W_{prev} \right\}$$

The 2nd stage antisurge valve parameters would be:

P_2 = Valve outlet pressure = 2^{nd} stage suction pressure (P_s)
T_1 = Valve inlet temperature = Compressor final stage discharge temperature (T_d)
Z_1 = Valve inlet compressibility = Compressor final stage discharge compressibility (Z_d)
k_1 = Valve inlet specific heat ratio = Compressor final stage discharge specific heat ratio (k_d)
MW = Valve inlet molecular weight = Compressor molecular weight

Superposing Antisurge Valve Capacity onto Compressor Performance Curves

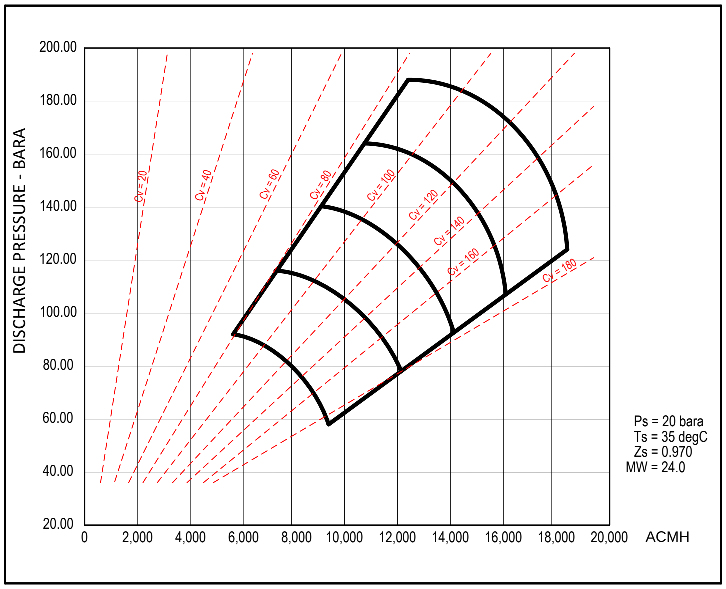

Figure 7 – Compressor Performance Curves (rotor bundle wheels properly matched) With Antisurge Valve Capacities Superposed

Once the antisurge valve type is selected, and hence its pressure drop ratio factor (x_T) at full opening is determined, it is possible to superpose different full opening valve C_v values onto the supplied compressor performance curves. This is a useful validation tool for antisurge valve sizing.

In the example given in Figure 7, above, it is readily seen that an antisurge valve C_v value of approx. 80 would be derived for all the surge points at all the indicated operating speeds of the compressor. Note that this is in line with the example given previously and illustrated in Figure 3. This indicated that the various wheels (impellers) that make up the compressor rotor bundle are closely matched insofar as their surge points are. In the author's experience, this proper matching of the wheels of the compressor bundle is exhibited in the majority of multiple-wheel compressors.

As may be deduced from the above Figure 7, a single antisurge valve capacity is adequate to protect the compressor during operations over the entirety of the "operating envelope, including the minimum speed that will be used during compressor idling.

In rare cases, however, wheel miss-match in the compressor rotor bundle may produce an operating envelope such as illustrated in Figure 8.

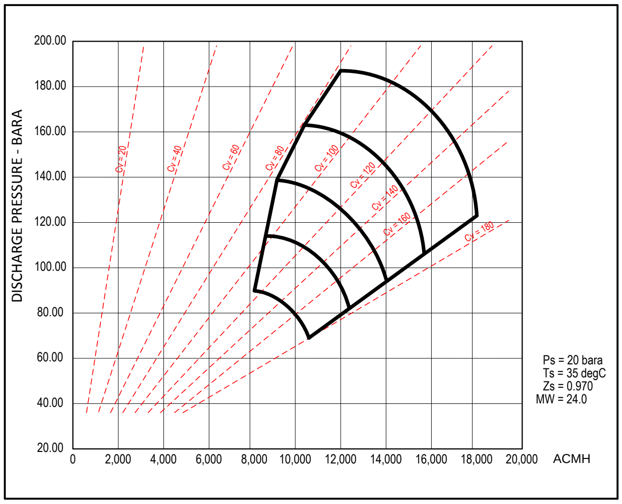

Figure 8 – Compressor Performance Curves (rotor bundle wheels mismatched) With Antisurge Valve Capacities Superposed

As may be seen from the above Figure 8, the antisurge valve capacity (C_V value) needed to protect the compressor from surging at the minimum operating speed is about 50% more than needed for higher operating speeds. This may be problematic, as choosing an antisurge valve capacity that corresponds to the minimum speed conditions could easily choke the compressor at higher speeds if allowed to open fully. For example, if the antisurge valve capacity selection was done at the minimum speed condition (C_V = 115), then the required valve capacity would be in the range of approx. 207 ~ 253. An antisurge valve with that capacity, if allowed to open fully, would drive the compressor into choke at any operating speed.

It is possible to develop a complicated solution involving more than one antisurge valve piped in parallel and arranged so that they open in a "staggered" manner, providing a higher total full opening C_V value as compressor speed diminishes, but this could increase the risk of surging or operating the compressor in the choke region, and so lower the reliability of the antisurge loop.

A better option, in the author's opinion, would be to restrict the compressor operating envelope so that the one single antisurge valve, with an appropriate full open capacity, is used to provide adequate surge control.

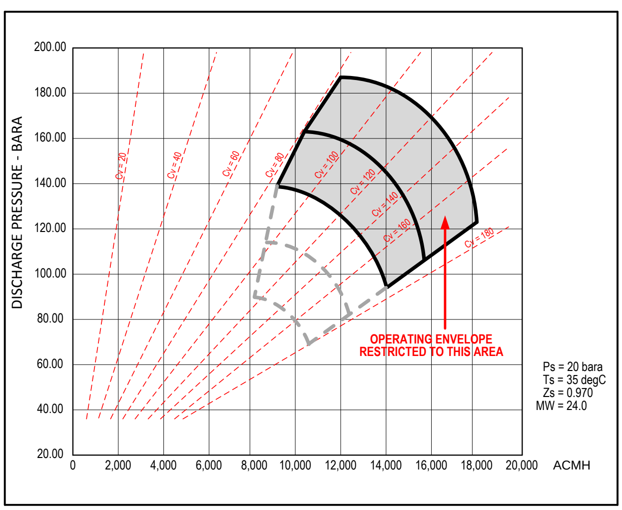

Figure 9 – Restricted Compressor Performance Curves to Suit Single Antisurge Valve Capacity

An actual example of a Hydrogen Recycle Compressor in a refinery will illustrate the proper sizing of the antisurge valve to suit all operating conditions.

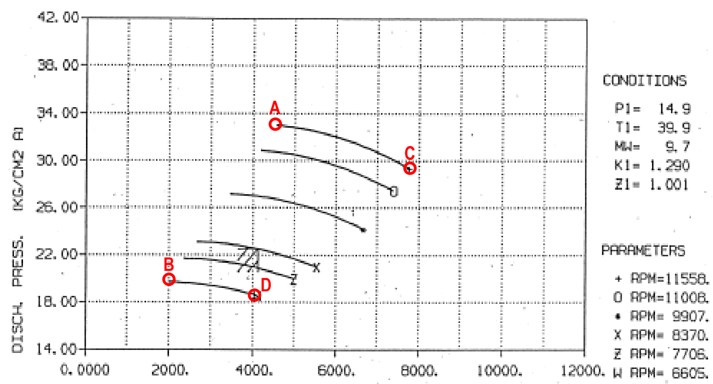

Figure 10 – Example of Hydrogen Recycle Compressor Performance Curves for Start of Run (SOR) Conditions

In the Start of Run (SOR), the molecular weight of the hydrogen-rich recycle gas is 9.7. Using the methodologies presented here, the antisurge valve capacity (C_V) for the points illustrated in Figure 10, above are:

- Required valve capacity C_V at the surge point A @ max. performance curve = 91
- Required valve capacity C_V at the surge point B @ min. performance curve = 87
- Required valve capacity C_V at the choke point C @ max. performance curve = 205
- Required valve capacity C_V at the choke point C @ min. performance curve = 200

At the End of Run, the molecular weight of the hydrogen-rich gas drops to 7.9. The performance curves therefore shift, as per Figure 11.

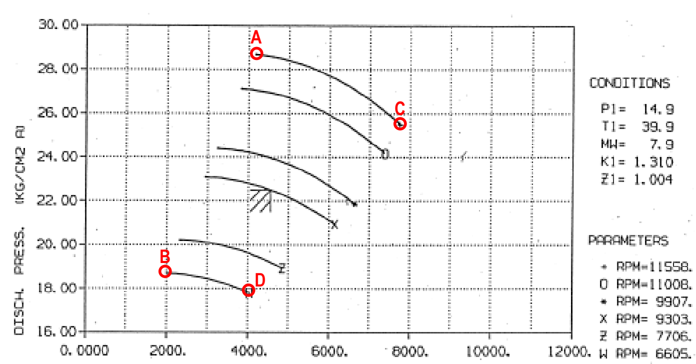

Figure 11 – Example of Hydrogen Recycle Compressor Performance Curves for End of Run (EOR) Conditions

The antisurge valve capacity for the same points become:

- Required valve capacity C_V at the surge point A @ max. performance curve = 93
- Required valve capacity C_V at the surge point B @ min. performance curve = 88
- Required valve capacity C_V at the choke point C @ max. performance curve = 200
- Required valve capacity C_V at the choke point C @ min. performance curve = 204

utilize the compressor to provide pressurized Nitrogen to dry out the process, and for that operating condition, the provided performance curve is as per the following Figure 12:

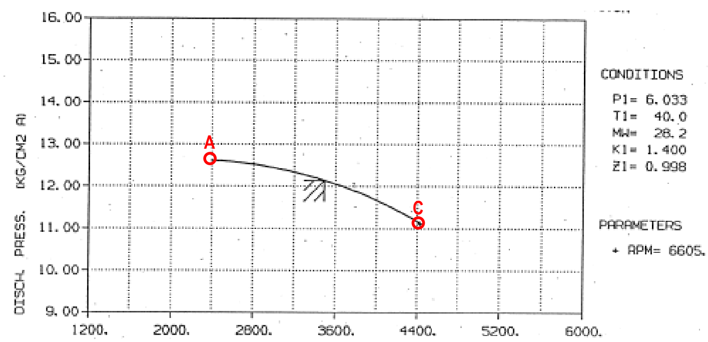

Figure 12 – Example of Hydrogen Recycle Compressor Performance Curves for Drying Conditions

The antisurge valve capacity for the surge and choke points become:

- Required valve capacity C_V at the surge point A = 94
- Required valve capacity C_V at the choke point C = 197

Thus in order to size the antisurge valve to suit all the provided operating conditions, the highest surge point flow is considered, which is $C_{V,surge}$ = 94. The oversizing factor is then applied (1.8 ~ 2.2) to obtain the initial recommended antisurge valve full opening capacity range of 169 ~ 207.

It is further noted that the Drying operating condition performance curve indicates that the compressor choke point is reached at an antisurge valve capacity of 197, hence the final recommended antisurge valve full opening capacity range of 169 ~ 190 is selected.

Dynamic Characteristics of the Antisurge Valve

The antisurge valve must stroke quickly and precisely in response to complex command signal profiles generated by an antisurge controller. Often the antisurge controller output, which represents the position command signal to the antisurge valve, is made up of a combination of closed-loop P+I responses, as well as open-loop step changes, followed by a decaying profile that is configured by the antisurge controller.

The actuation system of the antisurge control valve must therefore be engineered to produce the required smooth and precise stroking of the valve that matches the position command signal of the antisurge controller.

The antisurge control valve actuation system must include such components as:

- A digital positioner that provides for both the open-loop step changes and closed-loop P+I changes (position command signal) of the antisurge controller.
- Devices that amplify the motive fluid of the actuator in both the opening and closing directions (e.g. volume boosters for pneumatic actuators), and,
- A quick-dump device (e.g. solenoid valve) that permits the quick opening of the antisurge valve in response to an ESD (emergency shutdown) signal that may be generated by a Safety Instrumented System (SIS) independently of the antisurge controller.

Examples of such complex command signals from the antisurge controller are shown in figure 13.

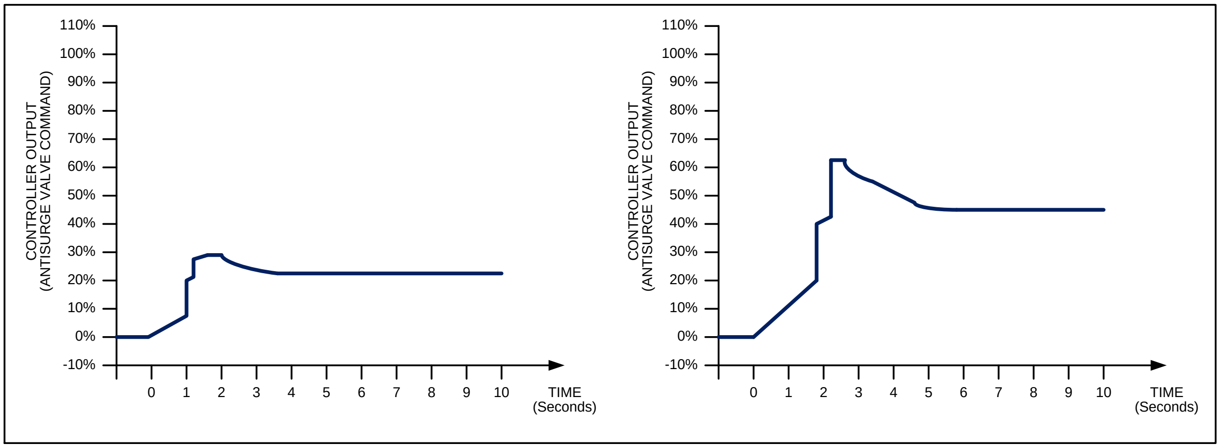

Figure 13 – Examples of Complex Antisurge Controller Output (Valve Position Command) Signals

In order to assist the antisurge valve manufacturer to meet the performance goals for the antisurge valves, the following dynamic characteristics for the valve actuation should be achieved:

Fast and precise full-stroking of the valve under positioner control:

As a minimum, under positioner control, the valve must stroke from fully closed to at least 95% open in 2 seconds or less. Normally, it is desirable to have the antisurge valve stroke from fully open to at least 95% closed in the same time (2 seconds or less), but it is acceptable that the valve strokes from fully open to at least 95% closed in no more than 8-10 seconds. See Figure 14 below.

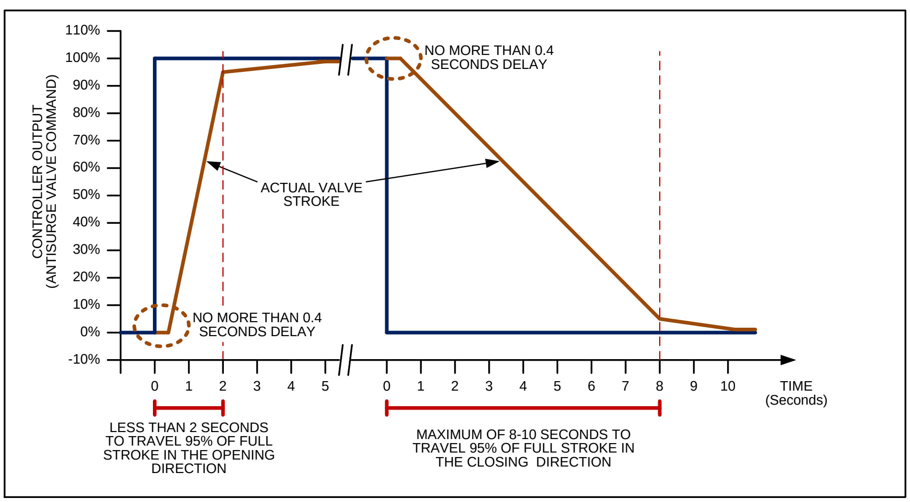

Figure 14 – Full-Stroke Speed of the Antisurge Valve Under Positioner Control

To be noted, the above difference in opening and closing times is not required for the purposes of the antisurge control, but rather to provide valve manufacturers practical guidelines to deliver the needed valve stroking quality. The antisurge control strategy will normally select slow closing of the antisurge valve following an open-loop step opening of the valve, but it is recommended that this be achieved electronically within the controller by means of dedicated algorithms that set the controller output signal value (controlled decay), rather than engineering the antisurge valve actuation system to have different stroking speeds depending on the direction of travel. When the antisurge controller commands the valve to fully open or close, the valve actuation system must exhibit no more than a 0.4 second delay, or dead time.

Finally, it might be useful to consider that the valve actuation system includes a mechanism to avoid end-of-stroke slamming, which could potentially damage the actuator, when the valve is commanded to open or close fully.

No significant hysteresis or overshoot of the valve for partial stroking under positioner control:

The antisurge valve must partially stroke for closed-loop P+I, or open-loop step change command signals from the antisurge controller without significant hysteresis or dead-band. Hysteresis or dead-band is a range or band of controller output values that do not produce a change in the proximity-to-surge variable when the controller output changes direction. It is desirable to have the antisurge control valve exhibit 1% or less (of full-span travel) of hysteresis or dead-band.

Since the antisurge controller may send the antisurge valve a command to step open and then resume modulating control action, it is desirable to have the valve actuation system achieve the step change (in the opening direction) with as little instability as possible. While “overshoot” (antisurge valve actuation system initially opens the valve more than the target position then settles to the target position) may be somewhat acceptable, instability in valve actuation that may cause an overshoot in close direction is not acceptable. See Figure 15.

In general, it is recommended that one-sided “overshoot” (i.e. in the opening direction only) should not exceed 20% of the step change in the controller output.

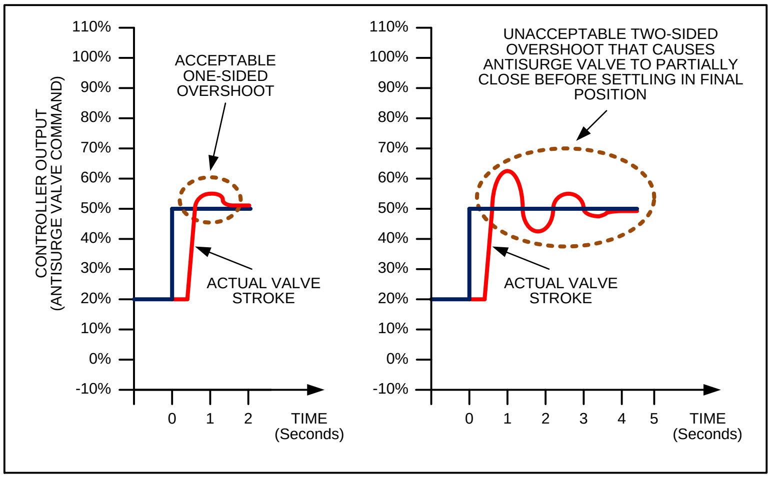

Figure 15 – Overshoot in Antisurge Valve Actuation System

Smooth continuous stroking under positioner control:

The antisurge valve must stroke smoothly, without observable jumps or jerkiness, when a continuously variable command signal from the antisurge controller is applied as shown in Figure 16.

In order to validate the proper dynamic characteristics of the selected antisurge control valve, it is recommended to subject the antisurge valve with its actuation system to a series of controlled performance tests ***<u>at an internal valve static pressure and flowing conditions approximating actual process conditions</u>*** in order to:

- Confirm the smooth stroking for a continuously variable command signal (i.e. 5% per second, in both the opening and closing direction), and
- Record the overshoot and hysteresis for step changes of 10%, 20% and 50%, as shown in Figure 17.

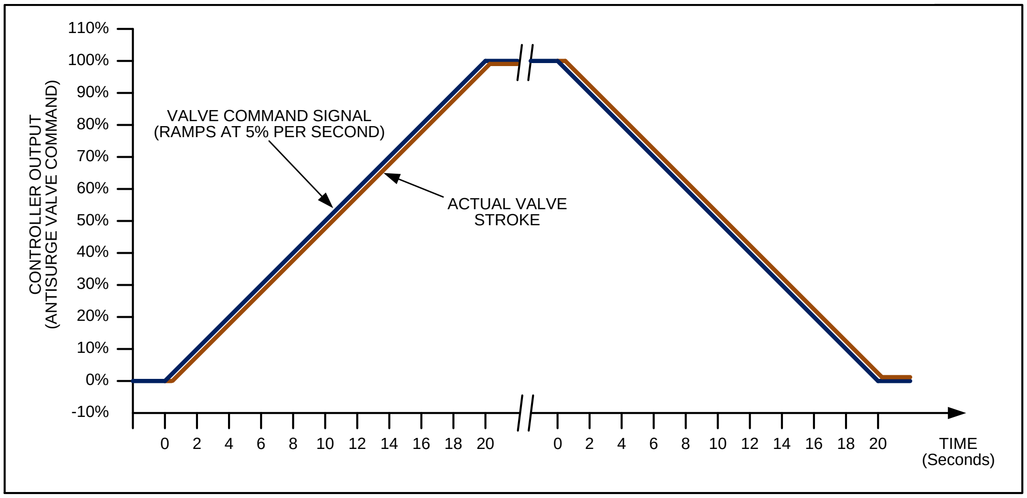

Figure 16 – Continuously Variable Command Signal Stroke Testing of the Antisurge Valve Under Positioner Control

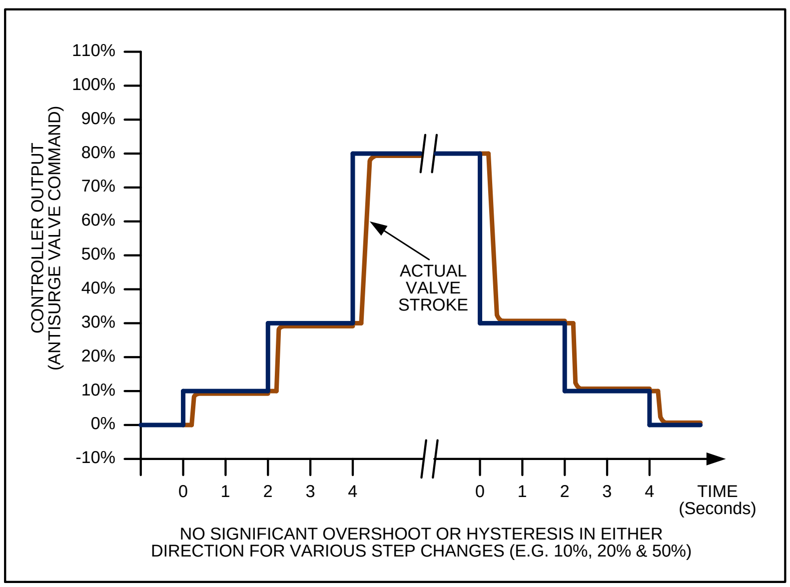

Figure 17 – Step Change Stroke Testing of the Antisurge Valve Under Positioner Control

Noise Abatement Concerns for the Antisurge Control Valve

Experience has shown that a large percentage of antisurge control valves will experience high pressure differentials and hence have a tendency to generate excessive noise levels.

There are two generally accepted methods to deal with control valve noise:

- Allow the noise to be generated inside the valve and install an acoustic enclosure. Alternatively, use an external restriction, such as a silencer or diffuser, in-line with the antisurge control valve (Figure 18), to reduce the pressure differential the antisurge valve is handling. These methods are commonly referred to as “path treatment”.
- Use a special valve trim (internals) that provide a torturous path for the gas inside the valve, thereby reducing its velocity, and the capacity to generate noise. This is commonly referred to as “source treatment”.

In many antisurge control valve applications, the high differential pressure that the valve is operating under will result in the internal noise level being so high as to create the risk of mechanical damage to the valve (typically at noise levels above 120 dBA). This would preclude the use of an acoustic enclosure. In many antisurge control valve piping layouts, designers install an in-line silencer or diffuser, with the intent to reduce the differential pressure available to the valve, and hence expecting noise levels in the antisurge valve to be lower than the 85 ~ 90 dBA levels considered acceptable. There is a pitfall in this approach, however, that is almost never considered. Let us consider a system that uses a diffuser downstream of an antisurge control valve, in the following arrangement:

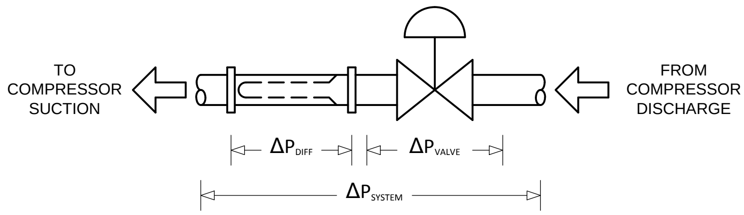

Figure 18 – Antisurge Valve in Series with an In-line Diffuser or Silencer

Let us further consider that the design intent is to have the in-line diffuser or silencer absorb half the available system pressure drop, thus reducing the pressure drop available to the antisurge valve to half its original value. The problem lies in the fact that, while the antisurge control valve has a variable capacity (C_V) as a function of its stroke, the diffuser or silencer has a fixed capacity or C_V.

Thus it becomes problematic to correctly size the in-line diffuser or silencer. Remember that the antisurge valve is sized for approximately twice the compressor's surge flow rate. So let's first assume that the in-line silencer or diffuser will be sized so that it absorbs half the available system pressure drop when the compressor operates at the surge control line. It is known that the mass flow through the silencer is proportional to the square root of the differential pressure across it, or $W_S \propto C_{V,S} \cdot \sqrt{\Delta P_S}$, where:

W_S = mass flow through the silencer.

$C_{V,S}$ = silencer flow coefficient.

ΔP_S = differential pressure across the silencer (being half the available system pressure drop).

The maximum differential pressure that the silencer can ever experience in such a system is limited to twice the original differential pressure (i.e. the silencer absorbs the full available system pressure drop, leaving no pressure drop across the valve). It then follows that the maximum flow the silencer can handle will be limited to approximately $\sqrt{2} \cdot W_S$, or 1.41 times the surge point flow before it chokes, thereby reducing the extra flow capacity of the overall antisurge control piping (valve plus in-line diffuser or silencer) to only 1.41 times the surge point flow, instead of the desired 1.8 ~ 2.2 times the surge point flow.

On the other hand, if the in-line silencer or diffuser was to be sized to produce half the system pressure drop at twice the surge point flow (similar to the antisurge valve), then at steady-state flow at the surge control line, or half its design flowrate, the silencer will absorb about a quarter of its rated pressure drop, thus leaving the antisurge valve with 75% of the overall system pressure drop, which would make the valve too noisy. It is therefore recommended that the antisurge control valve, if predicted to be too noisy, to be equipped with a suitable internal noise abatement trim internally, so that it can provide the required noise attenuation without the need for an external in-line device.

PART 2: SUCTION THROTTLE VALVES

General

It is possible to control the throughput (loading) of an electric motor-driven single speed centrifugal compressor by means of

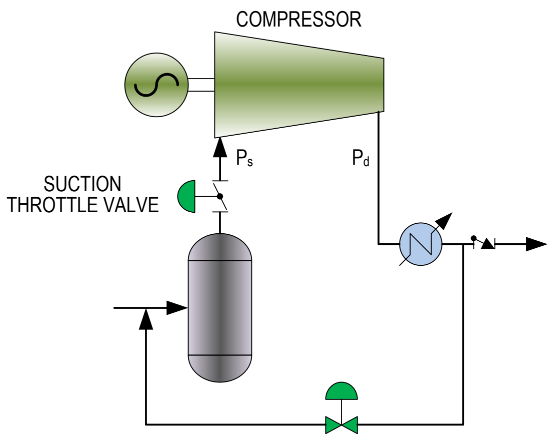

Figure 19 – Single Speed Compressor with a Suction Throttle Valve

The compressor design point is normally depicted on the performance curve with the suction throttle valve fully open, i.e. with a negligible pressure drop across it. Hence the process inlet pressure is equivalent to the compressor suction pressure P_S.

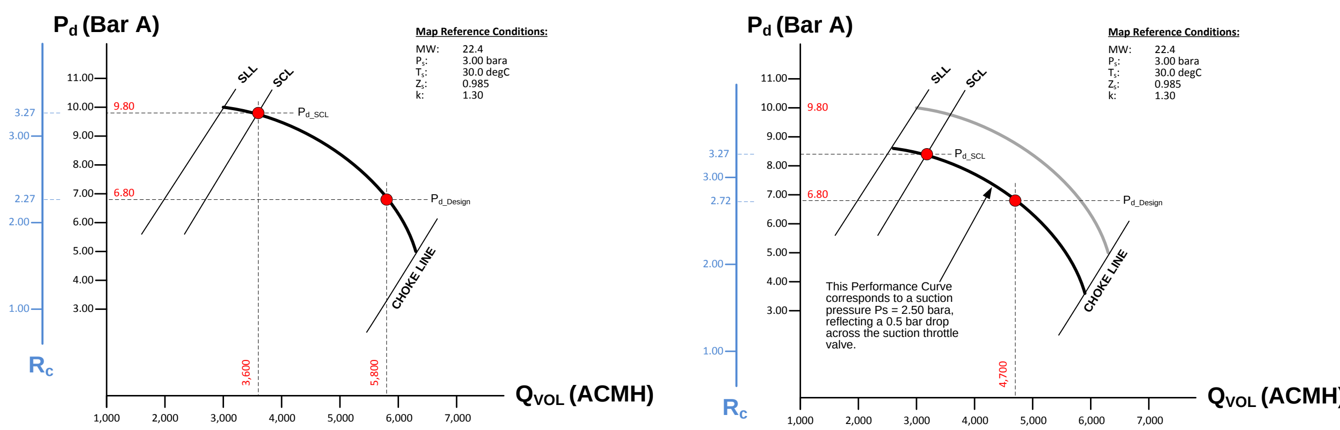

Figure 20 – Compressor Performance Curve with the Suction Throttle Valve Fully Open and Partially Closed

As may be seen in the left pane of the above Figure 20, the volumetric flow at the Design Point is 5,800 ACMH for the design discharge pressure of 6.80 bara. In order to reduce the throughput of the compressor it is necessary to force the compressor's operating point to "ride the performance curve". In order to do this while keeping the system resistance essentially the same, it is possible to "shift" the location of the performance curve with respect to the axes illustrated in Figure 20. This is achievable by closing the suction throttle valve, and so increasing the pressure drop across it. This in turn lowers the suction pressure of the compressor while keeping the pressure of the gas source (upstream of the suction throttle valve) constant. For example, if the suction throttle valve was closed enough to produce a pressure drop of 0.50 bar across it, and the discharge pressure was kept the same (at 6.8 bara), the pressure ratio of the compressor would rise from 2.27 to 2.72.

It would be possible to "shift" the performance curve and the associated R_C scale of above Figure 20 to a new location within the same suction flow and discharge pressure coordinates as per the right pane of the above Figure 20. Note that the flow through the compressor drops to approx. 4,700 ACMH.

Location of the Suction Throttle Valve

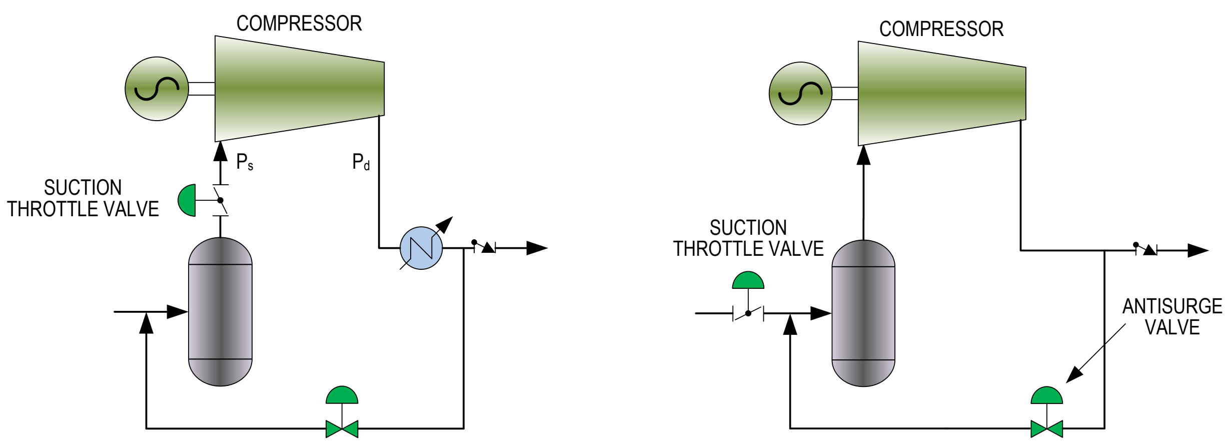

Figure 21 – Suction Throttle Valve Location

If the suction throttle valve is located outside the recycle loop, as shown in the right pane of above Figure 21, then it cannot influence motor power or current, once the antisurge valve opens.

That is why, if the suction throttle valve is to act as the control element that may be used for motor power or current limiting, then it must be located inside the recycle loop, as shown in the left pane of Figure 21.

Sizing the Suction Throttle Valve

Generally, a butterfly control valve is used in the vast majority of applications requiring a suction throttle valve. A good rule of thumb is to select a butterfly valve with the same size as the inlet piping of the compressor. Such a valve will have a high C_V, and hence produce the smallest possible pressure drop across it when fully open, hence reducing the energy penalty associated with the pressure drop across the valve. If the suction throttle valve is located outside the recycle piping circuit, i,e, upstream of the recycle line tie-in at the compressor suction; then the suction throttle valve may be allowed to close fully as this will have no impact on the recycle flow.

On the other hand, if the suction throttle valve is located inside the recycle loop, as shown in Figure 21 in the left pane, it is necessary to prevent the full closure of the valve, in order to ensure that the suction throttle valve has sufficient capacity at all times to allow the recycle flow that is delivered by the antisurge valve. It is therefore necessary to establish the suitable minimum opening of the suction throttle valve. In order for the suction throttle valve to be used to limit the electric motor driver's amperage during train start-up as well as normal operation, the location of the valve must be inside the recycle piping circuit.

Estimating the Minimum Opening of the Suction Throttle Valve

Assume that the compressor suction line is a 10 inch line and equipped with a 10 inch butterfly suction throttle valve with a fully open C_V of approx. 3,000.

Referring to the left pane of Figure 20, above, the highest pressure ratio that the compressor may tolerate at the Surge Control Line is $R_{c\_SCL} = \dfrac{P_{d\_SCL}}{P_{s\_design}} = \dfrac{9.80}{3.00} = 3.267$.

So, if the discharge pressure of the compressor was to be kept constant at the design value of 6.80 bara, this would imply that the suction pressure can be allowed to drop to no more than $\frac{P_{d\_design}}{R_{c\_SCL}} = \frac{6.80}{3.267} = 2.08$ bara.

Therefore, the pressure drop across the suction throttle valve at its minimum opening is limited to 3.00 – 2.08 = 0.92 bars.

At this minimum allowed opening, the suction throttle valve would have to have sufficient capacity (C_V) to pass the volumetric flow that the compressor needs at the Surge Control Line, in the example above in the left pane of Figure 14, i.e.3,600 ACMH. From the design inlet conditions of the compressor it is possible to calculate the inlet density (ρ_S) of the gas flowing through the compressor as:

$$\rho_s = \frac{MW}{T_s \cdot Z_s \cdot R_o} = 2.7067 \text{ kg/m}^3$$

This would establish the equivalent mass flow through the valve as 9,744.2 kg/h.

The suction throttle valve parameters would be:

P_1	= Valve inlet pressure =	Compressor design suction pressure (P_S)
T_1	= Valve inlet temperature =	Compressor rated suction temperature (T_S)
Z_1	= Valve inlet compressibility =	Compressor design suction compressibility (Z_S)
k_1	= Valve inlet specific heat ratio =	Compressor design suction specific heat ratio (k_S)
MW	= Valve inlet molecular weight =	Compressor molecular weight

Example

For the compressor performance curve depicted in the left pane of above Figure 20, the following antisurge valve parameters may be derived:

W_{STV}	= Mass flowrate through the suction throttle valve	= 9,744.2 kg/h
ΔP_{STV}	= Differential pressure across the suction throttle valve	= 0.92 bars
$P_{1,STV}$	= Suction throttle valve inlet pressure	= 3.00 bara
$T_{1,STV}$	= Suction throttle valve inlet temperature	= 30.0 degC
$Z_{1,STV}$	= Suction throttle valve inlet compressibility factor	= 0.985
$k_{1,STV}$	= Suction throttle valve inlet gas specific heat ratio	= 1.30
MW	= Valve inlet molecular weight	= 22.40

Based on the above parameters, and assuming a butterfly valve with a pressure drop ratio factor (x_T) of 0.20, the calculated valve capacity at the minimum opening is 436.0. Note that this represents about 15% of the Cv of the fully open valve. Thus, it it had a linear actuator, then the minimum clamp should prevent closure more than 15% of the valve stroke.

PART 3: HOT GAS BYPASS VALVES

General

While the compressor is running within its operating envelope, the antisurge valve (recycle or blow-off) is sized so as to provide enough to cause surge events. As seen in Part 1 of this tutorial, the sizing of the antisurge valve is based entirely on the surge limit characteristics of the compressor and does NOT take into account the discharge volume of the compressor. During an ESD scenario for the compressor, it is expected that the antisurge valve will start to open at approximately the same time as the compressor driver is stopped. As the compressor rotor decelerates towards standstill, the gas in the discharge volume is evacuated by the opening or opened antisurge valve.

According to the Fan Law, the head produced by the decelerating rotor drops at a rate equivalent to the square of drop in speed. At the other end, the opening antisurge valve will evacuate the compressor discharge volume based, and hence reduce the stored energy it contains, at a rate that depends on the size of that volume, the full open capacity (C_V) of the antisurge valve, and the time it takes to fully open the valve. In this race, if the discharge volume is too large, and so it is evacuated too slowly compared to the rate at which the head of the decelerating rotor is decreasing, it could quite easily cause a single or multiple surge events for the compressor.

Evaluating the Discharge Volume

The discharge volume of the compressor (see Figure 22) is volume of the piping and vessels between three flanges:

- the discharge flange of the compressor,
- the flange of the process check valve, and,
- the inlet flange of the antisurge valve.

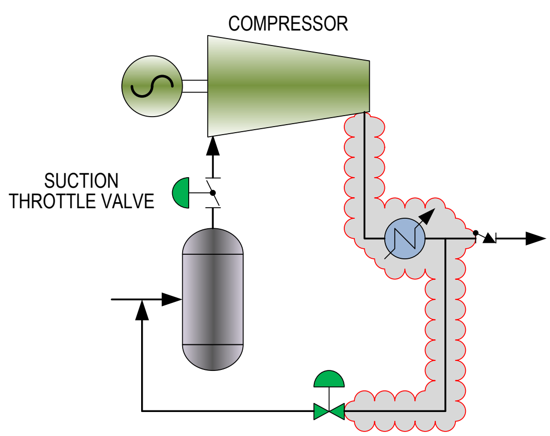

Figure 22 – The Discharge Volume of a Compressor

In order to evaluate the efficacity of the antisurge valve with respect to evacuating the discharge volume, it is possible to develop a high fidelity dynamic simulation which takes into account (amongst other factors):

- the speed at which the antisurge valve strokes to the fully open position on the receipt of an ESD command, i.e. via a solenoid on the actuator and not through the positioner.
- the upstream pressure of the antisurge valve as it decreases over time, hence the flowrate through the wide-open valve, which will also decrease over time.
- the discharge pressure of the compressor as it decreases over time, hence the surge point of the decelerating compressor will also decrease over time, in terms of volumetric flow and pressure ratio.

This can result in the need to develop a complex and therefore costly dynamic simulation model of the system. However, experience has demonstrated that most industrial compressors will decelerate by 20 – 30% in speed after one second after receiving a trip or ESD command. This means that we can consider the capacity to generate head will decrease by approximately 50% after that one second. It is thus possible to adopt a simplified approach to evaluating the efficacy of the antisurge valve in terms of reliving the y, it is recommended to obtain the approximate deceleration rate of the actual compressor train in order to determine the time it takes for the compressor to reach 50% head capacity.) First, we can simplify the discharge piping layout as per the following Figure 23.

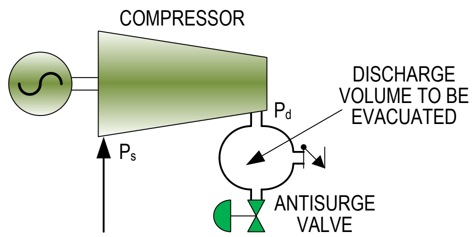

Figure 23 – Simplified Discharge Volume of a Compressor

Next, we can make the following assumptions for the one second duration that is considered:

- the antisurge valve becomes fully open and the compressor starts decelerating instantaneously upon the receipt of the ESD command
- the antisurge valve is required to drop the pressure of the discharge volume by 50%, i.e. at least at the same rate as the compressor's capacity to generate head is decreasing.
- the inlet pressure of the antisurge valve is assumed to be constant, with a value corresponding to 75% of the compressor's discharge pressure P_d.
- the outlet pressure of the antisurge valve is assumed to be constant, with a value corresponding to the compressor's suction pressure P_s.
- The expansion factor of the valve Y_1 is assumed to be at the worst-case value of 0.667.

Antisurge Valve Performance During an ESD scenario

Consider a discharge volume, V_d, in units [m^3]. It is possible to calculate the average density of the gas, in units [kg/m^3], in this discharge volume over the 1 second time frame as:

$$\rho_{V,d} \quad = \text{average gas density in the discharge volume} = \frac{P_{V,d} \cdot MW}{Z_d \cdot R_o \cdot T_d}$$

Where:

$P_{V,d}$ = average pressure, in units [kPa], in the discharge volume = $0.75 \cdot P_d$.
P_d = design discharge pressure, in units [kPa]
T_d = design discharge temperature, in units [K]
Z_d = design discharge gas compressibility factor
MW = design gas molecular weight
R_o = universal gas constant = 8.31441 kJ/kg·moleK

Thus, the mass of the gas in the discharge volume may be estimated as:

$$W_d = V_d \cdot \rho_d \text{ [kg]}$$

P_1	= Valve inlet pressure =	75% of the compressor design discharge pressure (0.75 P_d)
P_2	= Valve outlet pressure =	Compressor design discharge pressure (P_s)
T_1	= Valve inlet temperature =	Compressor design discharge temperature (T_d)
Z_1	= Valve inlet compressibility =	Compressor design discharge compressibility (Z_d)
k_1	= Valve inlet specific heat ratio =	Compressor design discharge specific heat ratio (k_d)
MW	= Valve inlet molecular weight =	Compressor molecular weight
Y_1	= Gas expansion factor =	0.67

Example

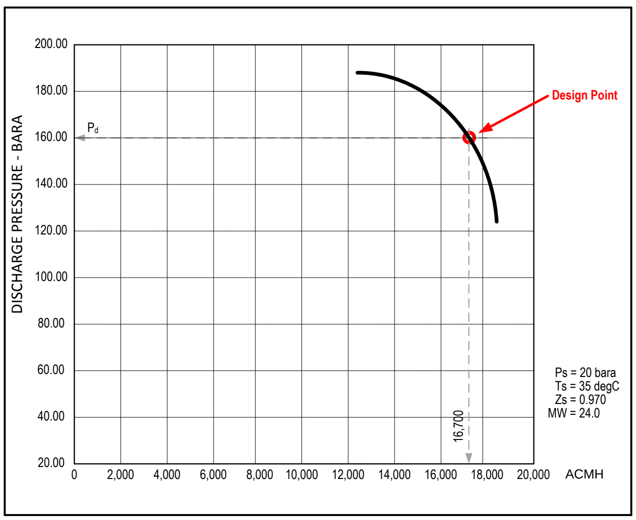

Figure 24 – Sample Compressor Performance Curve

Let us consider a sample compressor performance curve as per Figure 24, similar to that of Figure 2. From the previously worked example, it was determined that an antisurge valve with a capacity (C_v) of 148.5 ~ 181.5 is required for adequate surge control.

Case A:

If the discharge piping volume consisted of 20 meters of 6” piping and a water-chilled aftercooler with an internal volume of approximately 0.500 m^3, then the discharge volume may be estimated as approx. 0.853 m^3. In this case an antisurge valve with a capacity (C_v) of 160 would be capable of evacuating slightly over 98% of the discharge volume inventory in one second, indicating that it will prevent surging of the compressor during an ESD scenario.

Case B:

However, if the discharge piping consisted of 50 meters of 6” piping, and an air-cooled aftercooler with an internal volume of 1.5 m^3, then the discharge volume may be estimated as approx. 2.384 m^3. In this case, the same antisurge valve with a C_v of 160 would be capable of evacuating only approx. 35% of the discharge volume inventory in one second, hence it much more likely that the compressor would surge at least once during an ESD scenario.

In some cases, as in Case B above, a second valve is installed in parallel with the antisurge valve. This is called a "Cold Bypass Valve" and is designed to fully open rapidly upon the receipt of the ESD signal, simultaneously with the antisurge valve opening via its actuator's ESD solenoid. This is installed as per the following Figure 25:

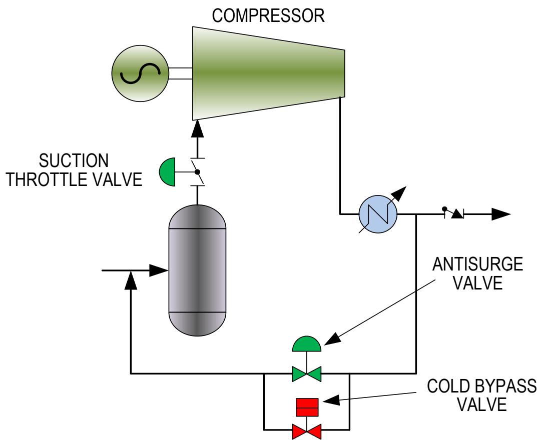

Figure 25 – Adding a Cold Bypass Valve

For the example cited in Case B, above, the Cold Bypass Valve needs to have a capacity (C_V) of approx. 68, raising the combined capacities of both the antisurge and cold bypass valves to 228, in order to evacuate the discharge volume fast enough during an ESD scenario.

Installing a Hot Gas Bypass Valve

An alternative approach could be to install a close-coupled Hot Gas Bypass Valve, as in the following Figure 26.

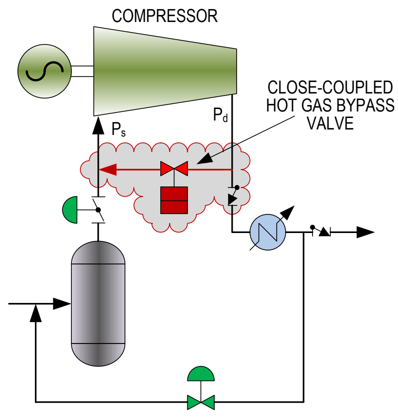

Figure 26 – Adding a Hot Gas Bypass Valve

In the event of an ESD scenario, the hot gas bypass valve provides an alternative path for the gas to recycle, thereby "killing" (dropping) the pressure rise of the compressor almost instantaneously, and thus preventing any potential surging, no matter how fast the compressor rotor is decelerating. In order to reduce the discharge volume that the hot gas bypass valve is handling, it must be located very close downstream of the compressor discharge flange and there must be an associated non-return (Check) valve just immediately of the hot gas take-off line.

Sizing Methodology for the Hot Gas Bypass Valve

Again, the mass of the gas in the discharge volume may be estimated as:

$$W_d = V_d \cdot \rho_d \,[\mathrm{kg}]$$

In this approach, however, we have considered that the hot gas bypass valve must be capable of dropping the pressure in the reduced discharge volume by a factor of 0.9 in one second. That implies that the ideal flowrate through it may be estimated as:

$$W = W_d \cdot 0.90 \cdot 60 \cdot 60 = 3{,}240 \cdot W_d \;\; [\mathrm{kg/h}]$$

Sizing Parameters for the Hot Gas Bypass Valve

P_1	= Valve inlet pressure =	75% of the compressor design discharge pressure (0.75 P_d)
P_2	= Valve outlet pressure =	Compressor design discharge pressure (P_s)
T_1	= Valve inlet temperature =	Compressor design discharge temperature (T_d)
Z_1	= Valve inlet compressibility =	Compressor design discharge compressibility (Z_d)
k_1	= Valve inlet specific heat ratio =	Compressor design discharge specific heat ratio (k_d)
MW	= Valve inlet molecular weight =	Compressor molecular weight
Y1	= Gas expansion factor =	0.67

Example

For the same sample compressor performance curve as per Figure 24, let us assume that the reduced discharge volume in a hot gas bypass arrangement is 0.150 m^3. The capacity of the hot gas bypass valve will then be approx. 21.8. This can be accommodated by a 2 inch globe valve suitable for the operating pressures and temperatures, and equipped with a fast-acting on/off actuator, such as a solenoid.

PART 4: QUENCH CONTROL VALVES

General

In most refrigeration compressors, hot gas from the compressor discharge is used as the recycle gas. This is due to the fact that the discharge gas pressure of the compressor is usually just above the condensing temperature of the refrigerant gas, and thus it is necessary to utilize the immediate discharge gas, prior to any cooling, to ensure that two-phase flow is avoided. However, allowing this hot recycle gas to circulate into the compressor inlet, especially when heavy or full recycle is necessary, would result in the inlet temperature of the compressor very quickly exceeding the trip, or safe value. Thus it is necessary to cool the recycle gas after it passes through the antisurge valve.

In many cases, the evaporative cooling effect of liquid quench is utilized. In this approach, an appropriate amount of liquid refrigerant is admitted at a sufficiently high pressure into a specially designed nozzle system, that ejects the liquid quench into the stream of hot recycle gas as multiple fine sprays as shown in Figure 27.

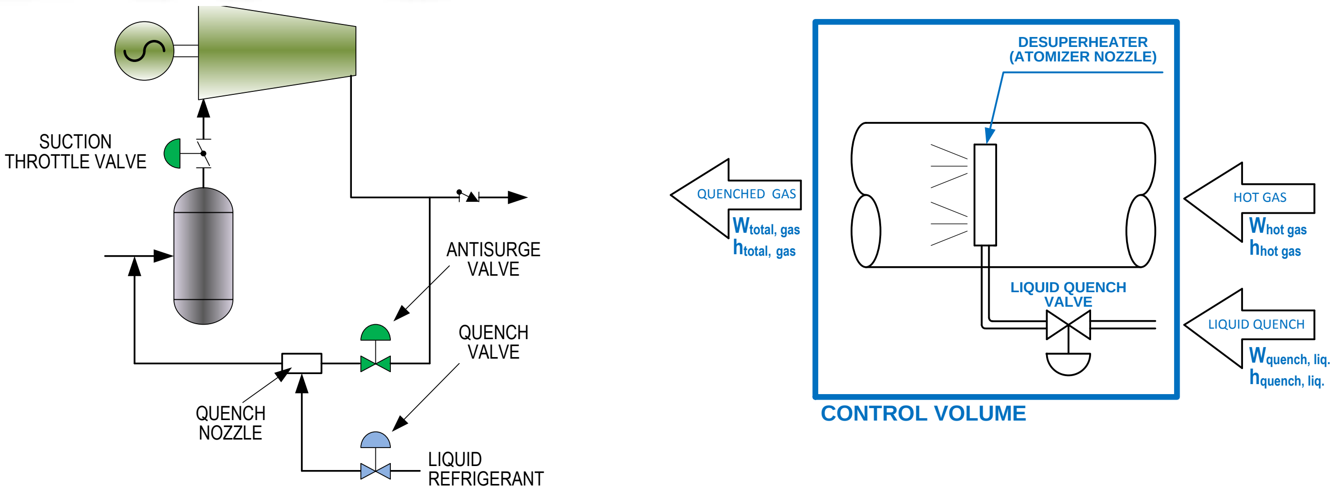

Figure 27 – Quench Valve and Quench Nozzle Arrangement

The fine sprays of liquid immediately evaporate, and thereby cools the combined stream. Crucial in the design of an effective liquid quench system is the actual design of the spray nozzles. In a poorly designed system, the liquid quench will not be introduced into the hot recycle gas stream as multiple fine sprays, and therefore there will be a significantly reduced cooling effect. Also there will be the additional risk of sending excessive amounts of liquid (the un-evaporated quench) into the compressor suction drum, and thereby tripping the train on excessive liquid level in that drum. Care must be taken to ensure that the quench fluid remains in its liquid state all through the quench liquid system, from the source through the quench valve and up to the nozzles; which implies that the liquid quench pressures must remain above the medium's vapor pressure throughout the quench system.

Establishing the Amount of Liquid Quench Needed

The right pane of Figure 27, illustrates the mass and energy balance around a quench nozzle used as an evaporative cooling system to reduce the temperature of the hot recycle gas. This can be stated as follows:

$$W_{total,gas} \bullet h_{total,gas} = \left(W_{hot,gas} \bullet h_{hot,gas}\right) + \left(W_{quench,liq.} \bullet h_{quench,liq.}\right) \quad \text{..........eqn. 1}$$

$$W_{total,gas} = W_{hot,gas} + W_{quench,liq.} \quad \text{..........eqn. 2}$$

Where:

$W_{total,gas}$	=	mass flowrate of the quenched (cooled) recycle gas [kg/h]
$W_{hot,gas}$	=	mass flowrate of the hot recycle gas [kg/h]
$W_{quench,liq.}$	=	mass flowrate of the liquid refrigerant used for quench [kg/h]
$h_{total,gas}$	=	enthalpy of the quenched (cooled) recycle gas [kJ/kg]
$h_{hot,gas}$	=	enthalpy of the hot recycle gas [kJ/kg]
$h_{quench,liq.}$	=	enthalpy of the liquid refrigerant used for quench [kJ/kg]

Equation 2 may be re-arranged as:

$$W_{hot,gas} = W_{total,gas} - W_{quench,liq.} \quad \text{..........eqn. 3}$$

Subsituting eqn. 3 into eqn. 1 yields:

$$W_{total,gas} \bullet h_{total,gas} = ((W_{total,gas} - W_{quench,liq.}) \bullet h_{hot,gas}) + (W_{quench,liq.} \bullet h_{quench,liq.}), \text{ or,}$$
$$W_{total,gas} \bullet h_{total,gas} = (W_{total,gas} \bullet h_{hot,gas}) - (W_{quench,liq.} \bullet h_{hot,gas}) + (W_{quench,liq.} \bullet h_{quench,liq.}), \text{ or}$$
$$(W_{quench,liq.} \bullet h_{hot,gas}) - (W_{quench,liq.} \bullet h_{quench,liq.}) = (W_{total,gas} \bullet h_{hot,gas}) - (W_{total,gas} \bullet h_{total,gas})$$

This yields:

$$W_{quench,liq.} = W_{total,gas} \bullet \frac{(h_{hot,gas} - h_{total,gas})}{(h_{hot,gas} - h_{quench,liq.})}$$

Example

Consider a propane refrigeration compressor. The pressure of the hot gas after the antisurge valve may be taken as the design suction pressure of the compressor. In our example this is 1.3 bara. The temperature of the hot gas is 90.0 degC. Liquid propane refrigerant is available upstream of the quench valve at a pressure of 7.0 bar and a temperature of -5.0 degC, and it is assumed that the quench valve will produce 0.25 bara pressure drop at the required flowrate, hence the pressure at the outlet of the quench valve is 6.75 bara. Assuming that the compressor is running on full recycle, we may consider that the compressor requires 40,000 ACMH (volumetric flow) of quenched gas at the surge control line, at -30.0 degC. We can establish the amount of liquid quench needed as follows:

The molecular weight of propane is taken as 44.1.
The density of the quenched gas (at -30.0 degC and 1.30 bara) is 2.9538 kg/m3.
Therefore required compressor quenched recycle volumetric flowrate is equivalent to $W_{total,gas}$ = 13,541 kg/h.
The enthalpy of the hot gas (at 90.0 degC and 1.30 bara), $h_{hot,gas}$ is 748.85 kJ/kg
The enthalpy of the quenched gas (at -30.0 degC and 1.30 bara), $h_{total,gas}$ is 540.76 kJ/kg
The enthalpy of the available liquid refrigerant (at -5.0 degC and 7.0 bara), $h_{quench,liq}$ is 187.72 kJ/kg
The amount of liquid quench needed would then be: $W_{quench,liq.} = 13{,}541 \bullet \frac{(748.85 - 540.76)}{(748.85 - 187.72)}$ = 5,022 kg/h

Sizing the Quench Valve

Since the quench valve is designed to handle a liquid refrigerant and care should be taken to avoid the liquid from flashing inside the quench valve and the downstream piping, then the sizing equations for incompressible non-vaporizing fluids should be used. The quench valve sizing criteria would then be:

P_1	= Valve inlet pressure =	7.0 bara
P_2	= Valve outlet pressure =	6.0 bara
T_1	= Valve inlet temperature =	-5.0 degC
MW	= Valve inlet molecular weight =	44.1
G	= Liquid specific gravity =	0.5359 (@ 7.0 bara and -5.0 degC)
μ	= Absolute viscosity of liquid =	0.1329
P_c	= Critical pressure =	42.477 bara
P_v	= vapor pressure @ -5.0 degC =	4.22 bara

The quench valve calculated capacity (C_v) would then be approx. 15.9. Providing a 25% oversizing margin would result in selecting a valve with a C_v of approx. 20.

NOMENCLATURE

ASV	= Antisurge valve
C_V	= Valve Flow Capacity
$C_{V,V}$	= Cv value of the valve at 100% open
D	= internal diameter of piping in [in.]
d	= nominal inlet diameter of the valve in [in.]
dPc	= Pressure differential across compressor.
dPo	= Pressure differential across flow measuring device (orifice typical), in WC or kPa
F_K	= ratio of the specific heat ratio of gas at the compressor discharge flange to the specific heat ratio of air.
Fp	= piping geometry factor
H_P	= Polytropic Head, ft or M
k	= Ratio of Specific Heats Cp and Cv of the gas
MW	= Molecular weight of the gas, lb/lb_{mole} or kg/kg_{mole}
P	= Pressure, psia or kPaA
Q	= Volumetric Flow Rate, actual cubic feet per minute, ACFM or M^3/hr
R	= Universal Gas Constant, 1545.3 ft *lbf/(lbmol.*ºR) or 8.3143 J/(mol.*ºK)
R_C	= Compression Ratio across the compressor (or compressor stage)
R_O	= Universal gas constant
ST	= Speed transmitter
T	= Temperature, degR or degK
TT	= Temperature transmitter
W	= mass flow
x	= ratio of actual pressure drop across the valve to absolute valve inlet pressure
x_T	= pressure drop ratio factor for the valve's particular internal geometry (obtained from the valve manufacturer).
Y	= gas expansion factor
Z	= Compressibility, non-dimensional
ρ	= gas density

Subscripts:

AC	= After Cooler
av	= Average
d, D	= Discharge
c, C	= Choke
fe	= Flow element
S	= Suction
SS	= Side Stream
SCL	= Surge Control Line
STV	= Suction Throttle Valve
LP	= Low Pressure Stage of the compressor
MP	= Medium Pressure Stage of the compressor
HP	= High Pressure Stage of the compressor
1	= Valve Inlet
2	= Valve Outlet

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