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COMPRESSOR LOADSHARING CONTROL AND SURGE DETECTION TECHNIQUES

Wayne Jacobson
Global Technology Manager
Compressor Controls Corporation
Des Moines, IA, USA

Medhat Zaghloul
Regional Manager – Abu Dhabi
Compressor Controls Corporation
Al Reem Island, Abu Dhabi, UAE

Michael Tolmatsky
Director – Research and Development
Compressor Controls Corporation
Des Moines, IA, USA

Serge Staroselsky
Chief Technology Officer
Compressor Controls Corporation
Des Moines, IA, USA

Jeff McWhirter, PE
Independent Consultant
Houston, TX, USA

Serge Staroselsky *is CCC's Chief Technology Officer in Des Moines, IA*
Serge has been with CCC in various roles for over 20 years, assuming his current position in 2014. He also spent 9 years at General Electric. Serge oversees product and technology development activities and has extensive experience in designing and implementing control systems for turbomachinery, specializing in antisurge control and protection and compressor capacity control, covering a variety of applications. He has B.S. degree in Mechanical Engineering from University of California at Berkeley and M.S., in Mechanical, Engineering from University of Minnesota.

Wayne Jacobson *is CCC's Global Technology Manager in Des Moines, IA.. Wayne joined CCC in 1996 as a Systems Engineer with a BS in Mechanical Engineering from the University of Wisconsin – Platteville. His responsibilities include providing technical guidance and support throughout the organization,develop and enhance control applications, maintain and improve standard engineering and overseeing the overall technical development of engineering staff.*

Medhat Zaghloul *is CCC's Regional Technology Manager for the Europe, Middle East and Africa regions with CCC, based in the Abu Dhabi Office. Medhat joined CCC in 1993, after a 15-year career in instrumentation and controls in the petrochemical industry. His responsibilities include providing technical guidance, supporting Sales, and developing technical solutions and control applications for CCC. Medhat has over 38 years of controls experience in a variety of up-, mid- and down-stream Oil & Gas facilities. Medhat holds a B.Sc. in Electrical Engineering from the Cairo Institute of Technology, Egypt.*

Jeff McWhirter *has extensive experience with turbomachinery controls encompasses many industries and extends to LNG plants - implementing both the APCI and COP processes, refineries utilizing fluidized catalytic cracking(FCCU) and power recovery, NGL facilities, oil and gas production facilities including separation, injection, gas lift and sales gas, as well ammonia and ethylene plants. Mr. McWhirter is a member of the working committee for the new Machinery Protection System API 670 5th edition specification and joint authored a patent entitled "Compressor-Driver Power Limiting in Consideration of Antisurge Control" in December 2012. Jeff received his B.S. degree in Mechanical Engineering from Texas A&M University in 1980 and is a member of the ASME*

Michael Tolmatsky *Graduated in 1981 from Moscow Oil and Gas Institute with Master degree in Electrical Engineering. Michael has worked as a control systems engineer for various chemical and petrochemical processes. He has been with Compressor Controls Corporation in various roles for 15 years. He also worked for GE (2007-2009) and Dresser-Rand (2009-2014). Currently, he leads the R&D department. Michael's main area of expertise is developing control systems.*

ABSTRACT

Centrifugal compressor trains operating in parallel or series arrangements are complex when it comes to turbomachinery control. Methodologies for controlling compressor trains in these arrangements piping layout considerations to be observed is discussed. Surge in centrifugal and axial compressors often manifests itself as oscillations of flow and pressure, as well as related compressor parameters, such as rotating speed. Many of the devices, which incorporate surge detection function, use either the location of the operating point relative to the Surge Limit Line (SLL) or physical manifestations of surge, such as high rates of changes of selected process signals for identifying surge. This paper analyzes surge signatures collected in the field and outlines several approaches to surge detection using a combination of parameters, such as flow and discharge or suction pressure, based on the type of application.

INTRODUCTION

Compressors operating in parallel or series are common in the oil and gas industry as well as in the petrochemical industry. Ensuring proper overall design for these systems is important for safe and efficient operation. Not only does the piping layout around these compressors effect the operation, the control system design can affect the efficiency and reliability of the compressors. There are several load sharing control methodologies for compressors operating in parallel arrangements, with some providing better operational efficiency and reliability than others.

API 670, 5th Edition, lists compressor flow, discharge pressure, and inlet temperature as the signals that should be used for surge detection. The surge detection function should be able to identify each surge cycle. An accepted approach in the industry consists of several actions (or combination of them) provided by the surge detector, such as issuing an alarm, opening the antisurge valve via a s trip solenoid, and actually tripping the unit if number of cycles exceeds a predefined threshold within a given time period, e.g., 3 cycles in 10 seconds. In case of flow, the standard recommends detecting surge when flow decreases below lowest possible surge flow or on a rapid decrease of the signal. For the inlet temperature method, the inlet temperature of the compressor rises considerably above the upstream gas temperature. There are no specific recommendations for discharge pressure, other than detecting rapid decrease and recovery. Detection of rapid decrease in the signal means that the surge detector should be able to calculate rate of change of the signal. In addition, the detector should also be able to discern between high rates of change during signal failures and surging.

In practice, reliable implementation of surge detection based on the rate of change presents a number of difficulties. First of all, detection function should be able to differentiate between process changes and surging. Surge signature of the compressor may be quite different in the field compared to shop testing. Therefore, surge testing should be conducted in the field in order to identify rates of change during surge. However, in many cases, particularly in new installations with large and critical machines with high potential forces acting on machine components, field surge testing is often not conducted due to risk of machine damage. Therefore, the surge signature remains unknown. Furthermore, the instrumentation used for measurement, such as differential pressure transmitters used for flow measurement, may limit the sensitivity of rate of change determination due to poor speed of response and excessive damping used for noise suppression. The location of the instrumentation also plays an important role. Pressure measurements, which are far away from the actual discharge and suction of the machine, may not exhibit significant changes during first several cycles.

REVIEW: INVARIANT COMPRESSOR MAP COORDINATES

In order to develop compressor network control strategies, it is necessary first to express compressor performance in an invariant coordinate system. Most compressor manufacturers express compressor performance curves in a coordinate system that is dependent on certain known and specified inlet conditions, and choose as the vertical axis polytropic head, discharge pressure or pressure ratio; and as the horizontal axis suction volumetric flow or mass flow. An invariant coordinate system removes this dependency on the inlet gas conditions (pressure, molecular weight, temperature, compressibility factor and specific heat ratio), and thus become dimensionless.

For the purposes of this tutorial, the vertical axis becomes “reduced head” (or $H_{p,red}$) and is equal to

$$H_p = \frac{{R_c}^{\sigma}-1}{\sigma} \cdot \frac{Z_{ave} \cdot R_o \cdot T_s}{MW} \quad \text{…… eqn (1)}$$

The horizontal axis becomes “reduced flow” (or ${Q_s}^2{}_{,red}$) and is equal to:

$$Q_s{}^2 = \frac{\Delta P_{o,s}}{P_s} \cdot \frac{Z_s \cdot R_o \cdot T_s}{MW} \quad \ldots\ldots \text{ eqn (2)}$$

If the proximity-to-surge of the compressor's operating point is expressed as the ratio of the slope of a line through the operating point relative to the slope of a line through the surge point, then:

The vertical axis reduces to: $H_{p,red} = \frac{R_c{}^{\sigma} - 1}{\sigma}$ …… eqn (3); and

The horizontal axis reduces to: $Q_s{}^2{}_{,red} = \frac{\Delta P_{o,s}}{P_s}$ ……. eqn (4)

If the gas composition remains constant then the vertical axis can be further reduced to: $H_{p,red} = R_c$ …… eqn (3-b).

The benefits of using such an invariant coordinate system are clear: it is possible to reduce all the different performance curves for a given compressor for all the different inlet gas conditions to a single "Universal" set of performance curves, that is independent of inlet conditions. In such an invariant co-ordinate system, the most widely used method of determining the proximity-to-surge (S_S) is ratio of the slope of a line passing through the operating point to the slope of a line passing to the surge point at that performance curve, as in the below Figure 1:

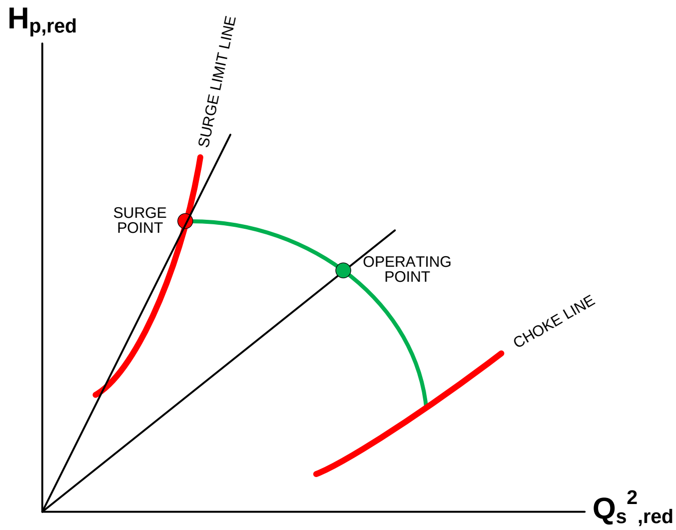

Figure 1

$$S_S = \frac{\left[H_{p,red} / Q^2_{s,red} \right]_{Operating\ Point}}{\left[H_{p,red} / Q^2_{s,red} \right]_{Surge\ Point}}$$

Under steady-state conditions, the proximity to the Surge Control Line can then be determined as:

DEV = 1 – (S_S + b1) where b1 is the surge control safety margin

The proximity-to-surge variable is termed "DEV" (abbreviation for DEViation) and when expressed in this invariant coordinate system looks like this:

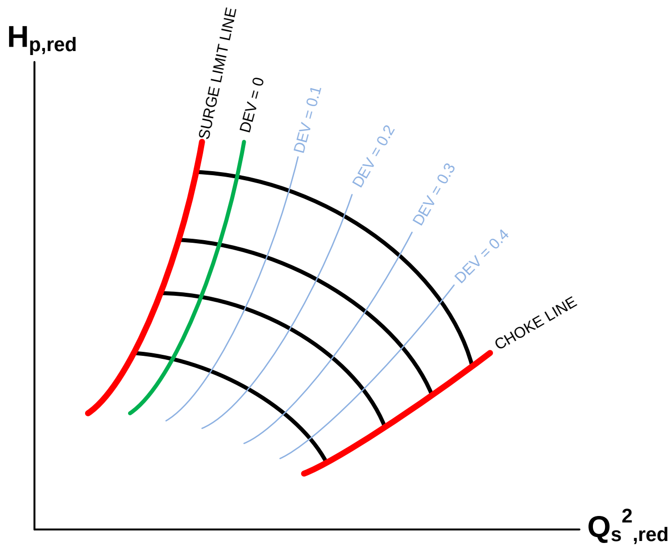

Figure 2

When expressed as above, the variable DEV will be equal to 0 when the operating point is on the surge control line. When it is to the right of the surge control line, DEV is a positive value and when it is to the left, it will have a negative value. A major benefit of using this invariant coordinate system is that the proximity-to-surge variable ("DEV") has the same numerical significance for all compressors, regardless of size or the gas being compressed. In other words, DEV is truly a universal expression of proximity-to-surge.

COMPRESSOR PERFORMANCE CONTROL

The ultimate goal of compressor performance control is to match the compressor delivery to the process demand. This is achieved by varying the compressor throughput, which is why the term "performance control" may be also be called "capacity control". There are two aspects of compressor performance control that must be considered:

- The selection of the most suitable process variable to represent the process performance demand side of the equation; and,
- The selection of the most suitable mechanism to vary the compressor delivery side of the equation.

Process Variable Selection for Performance Control

Some processes have wildly variable demand in a given period, while the compressors supplying that process have an unlimited, or almost unlimited supply sources. Typical of such a system would be a compressed air distribution network in a plant. In such cases the most suitable process variable is the pressure at the compressed air delivery manifold, which is the discharge pressure of the compressor(s). To smooth out any sudden and large variations in the discharge pressure measurement reading, it is advisable to install a buffer tank (holding capacity) where the discharge pressure is measured.

Another process/compressor configuration would be delivering compressed gas into an almost unlimited discharge volume, with a process demand that is "fixed" for long periods of time, but with the gas source experiencing fluctuations in supply. As this is almost the opposite of the previously described one, it should come as no surprise that the most suitable process variable for good performance control of the compressor would, in this case, be suction pressure. A typical example would be gas collection from offshore or onshore gas fields and compressing it into a long pipeline. Some processes require a certain supply of compressed gas for a specific reactive process, or to mix it in a certain proportion to another process fluid. An example would be compressed air in a blast furnace (steel plant) or air for a catalytic cracker (refinery). In cases such as these, then probably discharge mass flow of the compressor would be the most suitable process variable for compressor performance control.

The role of any centrifugal compressor used in a refrigeration cycle is to compress the refrigerant gas sufficiently so that it can be condensed with an economically available coolant, such as river or sea water, or using ambient air in air-cooled condensers. The ambient air or water temperature will dictate the pressure at which the compressor's discharge gas will condense (or change state to a liquid refrigerant), so controlling discharge pressure would be inappropriate. Compressor suction pressure control would be a much more representative primary process control variable.

Control Element Selection for Performance Control

Discharge Throttle Valve

Assume that it is required to control the pressure to a process using a discharge throttle valve on a single speed compressor is illustrated in the following Figure 3:

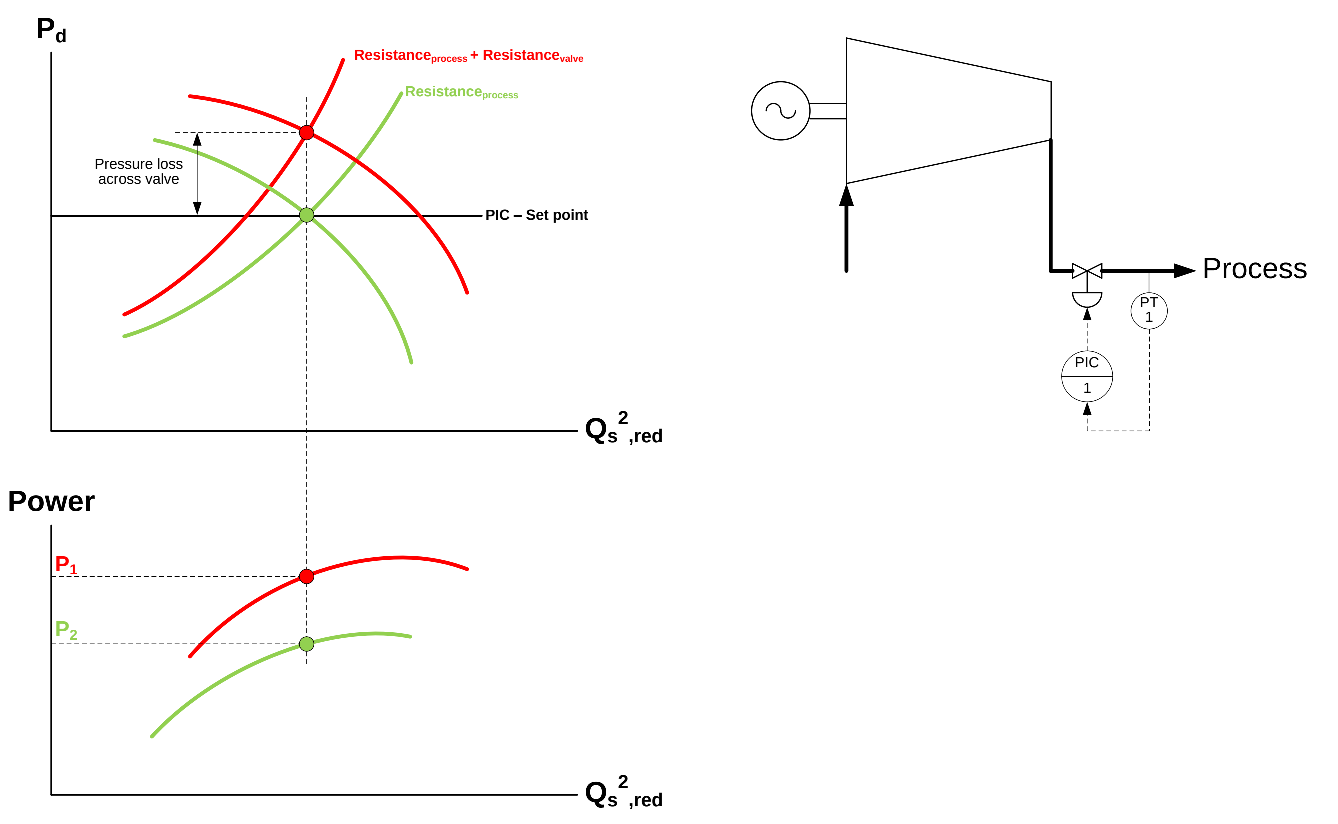

Figure 3

In order to accommodate process pressure set-point values that the process may require, it is necessary to force the compressor to operate at an even higher discharge pressure, and use the discharge throttle valve to drop the compressor discharge pressure to the value the process needs. So, even though the process feed requires only the expenditure of a compressor shaft power represented by P1, the compressor is required to operate with a shaft power equivalent to P2. This constant excess power consumed by the compressor makes this method of modulating compressor performance quite in-efficient.

Blow-off or Recycle Valve

Using the blow-off or recycle valve to control the process pressure, as illustrated in the following Figure 4 is even more energy inefficient than using a discharge throttle valve.

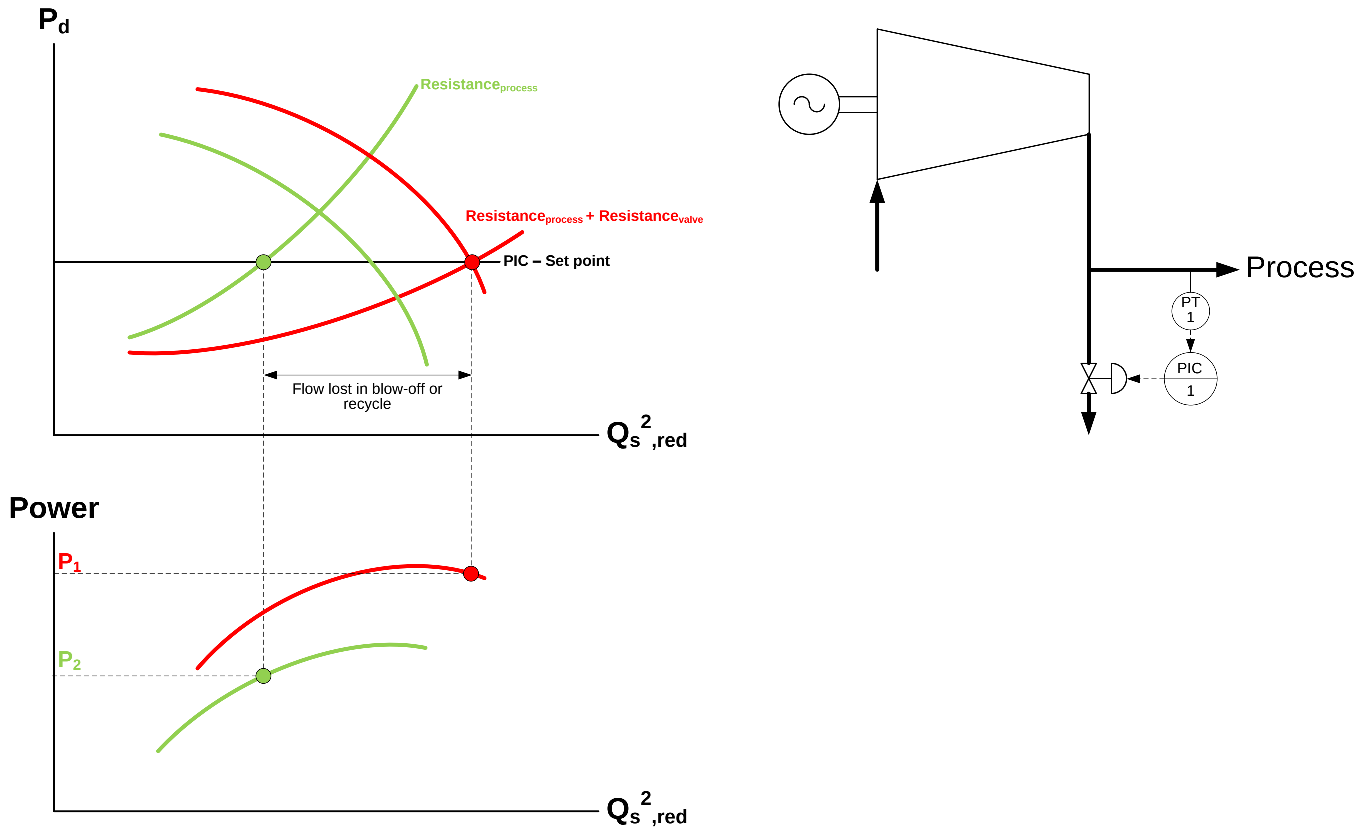

Figure 4

Here the compressor must operate all the time at close to its maximum flow capacity, with the excess to what is needed by the process recycled or blown-off.

Suction Throttle Valve

For single-speed compressors, it is a common practice to install a suction throttle valve that will, as it closes, lower the actual suction pressure of the compressor. If the discharge pressure remains relatively constant – as would be the case if the control intent was to control it – then as the suction throttle valve closes, the pressure ratio across the compressor rises. This, in turn causes the compressor to handle less flow.

On a compressor performance map using discharge pressure as the vertical coordinate, it is common practice to depict a "family" of performance curves for different suction throttle valve positions. It is important to keep in mind that a single speed compressor has only one performance curve – *at the stated suction conditions, including pressure*. As the mentioned earlier, when the suction throttle valve closes, an increasing pressure drop occurs across it, which means that the actual suction pressure of the compressor drops. However, it is convenient to depict the "family" of performance curves that represent the varying actual suction pressures on the original performance map at the "reference" suction pressure. Thus, as the suction throttle valve closes, it appears that the "reference" performance curve shifts to the left and downwards, for varying closing positions of the suction throttle valve.

This may be illustrated in the following Figure 5:

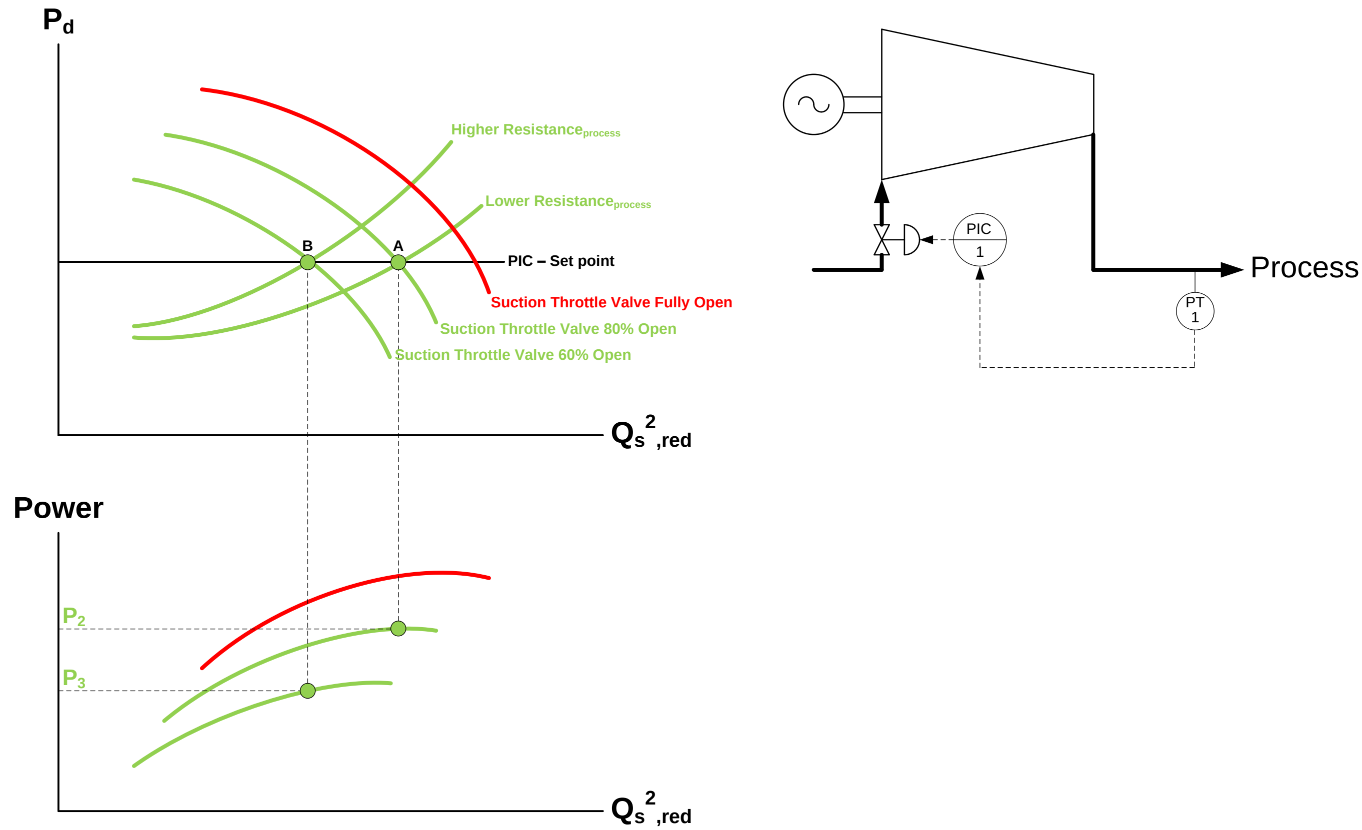

Figure 5

Using a suction throttle valve results in the power consumption of the compressor varying with the varying process load, which makes this method of compressor capacity control much more energy-efficient that the previous two methods (discharge throttling and using the blow-off or recycle valve). However, there is slight energy penalty to be paid in the quantity of power lost through the inevitable pressure drop across the suction throttle valve when it is not in the 100% open position. When selecting this method of controlling the performance of a compressor, the designer should be careful to ensure that the “reference” suction pressure – when the suction throttle valve is fully open – is sufficiently higher than atmospheric pressure, as any subsequent closure of the suction throttle valve would then produce sub-atmospheric pressures at the compressor immediate inlet flange.

Adjustable Inlet Guide Vanes (IGVs)

Another method to modulate the capacity (performance) of a single-speed compressor is to utilize adjustable or variable guide vanes located at the compressor inlet, upstream of the compressor’s first impeller. By modulating the IGVs, an angular deviation of the absolute velocity of the gas entering the leading edge of the first impeller is change, which causes a change in the head produced by the machine, the flow handled by it, and hence the power consumed by the machine. Using IGVs also results in the power consumption of the compressor varying with the varying process load. However, this method is even more energy efficient than the previous method (using an inlet throttle valve), because the energy losses associated with the suction throttle valve are eliminated. It is also possible to achieve a greater turndown of the capacity of the machine, while operating at high polytropic efficiency values.

The additional energy savings thus obtained justify the higher capital and operating costs of the IGVs, which are more complex to build and maintain than a suction throttle valve. This method may be illustrated in the following Figure 6:

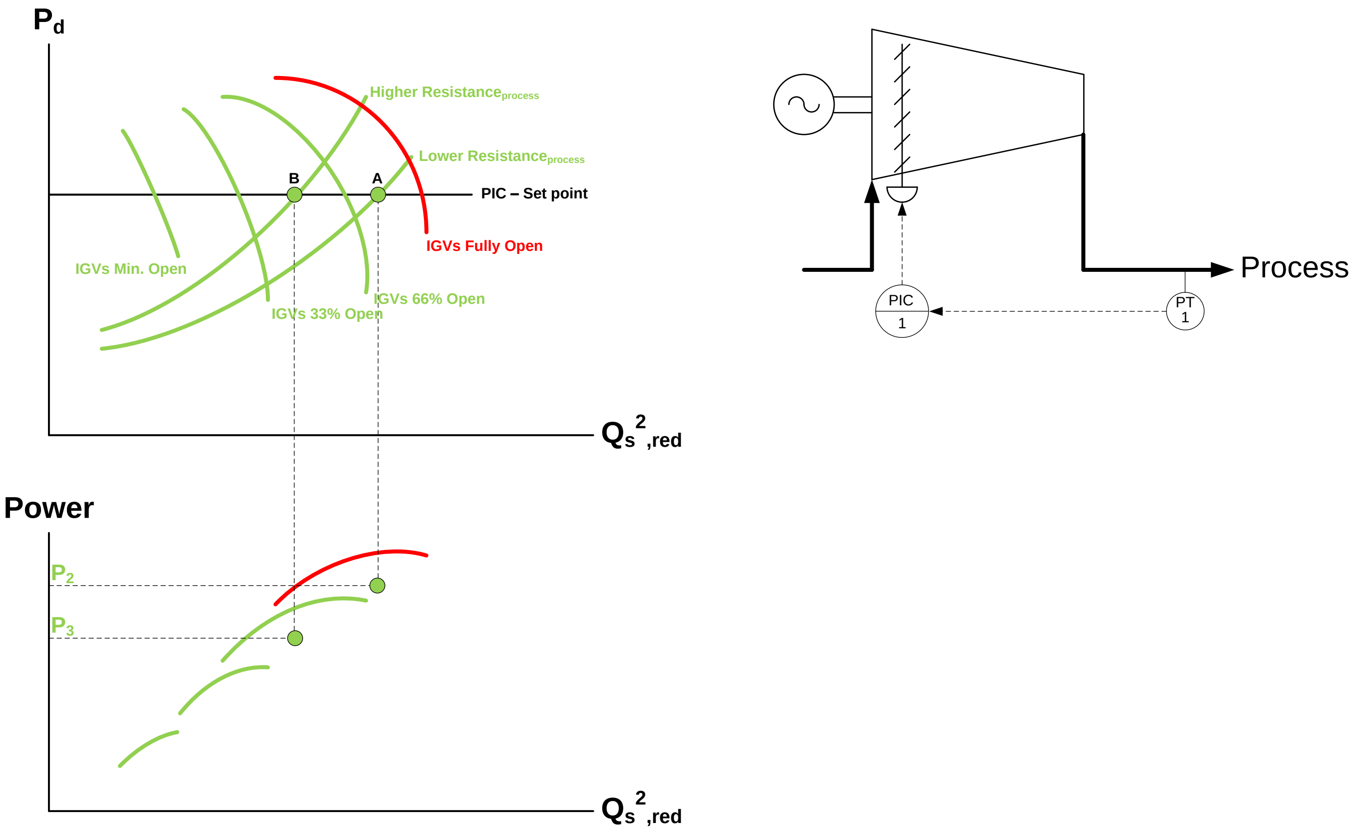

Figure 6

Speed Variation

The most efficient method to modulate the capacity (performance) of a compressor is to vary its speed. Speed modulation makes full use of the Fan Law (for a constant diameter impeller), which state that flow is proportional to the speed of the machine, pressure or head is proportional to the square of the speed and power is proportional to the cube of the speed.

Thus, the highest turndown is possible with speed variation, compared with other method of capacity control, and this allows for the most efficient energy reduction when the process load drops. This method may be illustrated in the following Figure 7:

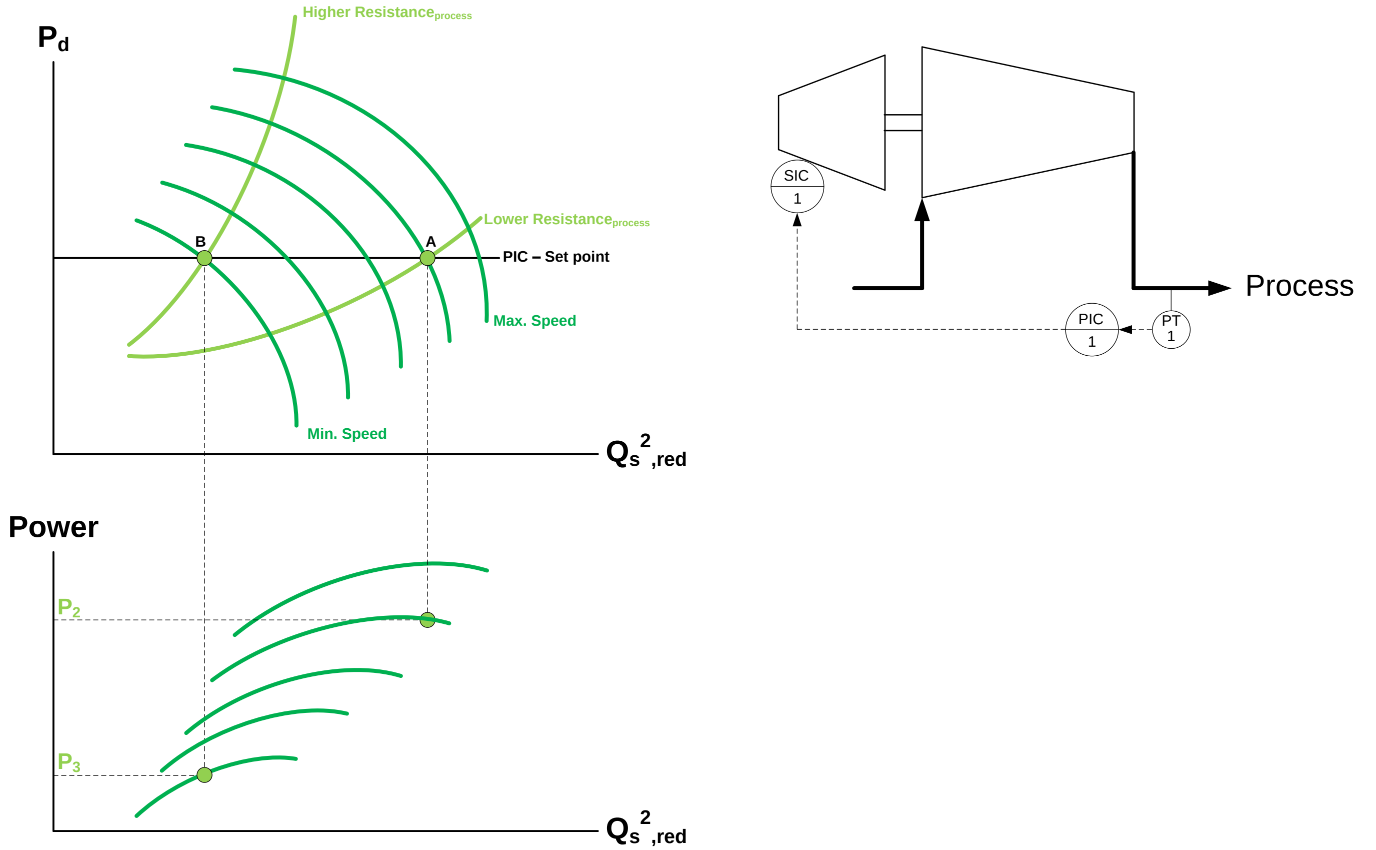

Figure 7

This method requires a much higher investment (capital costs) to provide the variable speed drive system for the compressor, and may involve higher maintenance costs than all other methods, but the energy savings over the life-cycle of the plant should easily justify this. Variable speed drive mechanisms include:

- Steam turbines
- Gas turbines
- Variable speed electric motors
- Single speed electric motors with a variable speed/torque hydraulic converter.
- Hot gas or cryogenic gas expander.

COMPRESSOR NETWORKS

Networks of compressors are installed for different reasons, which include:

- Redundancy – for example, one running 100% machine and one standby 100% machine in a parallel arrangement; or 3 x 50% machines, also in a parallel arrangement, with two running machines and a third, standby machine.
- Flexibility – for example, adding a compression train to allow running trains to be taken off-line for maintenance.
- Incremental capacity additions – for example adding a machine in series to an existing train to raise the overall pressure ratio, common when gas reservoir pressure drops due to depletion. Another example would be adding a second machine in parallel to an existing train in order to handle a higher gas compression requirement in terms of volume.

Parallel Compressor Networks

A parallel network is when two or more compressors are piped in an arrangement where there is a common suction header for all machines, as well as a common discharge header. As a result of this, every machine receives the same gas, and all running trains run at the same pressure ratio. An example of a parallel network would be:

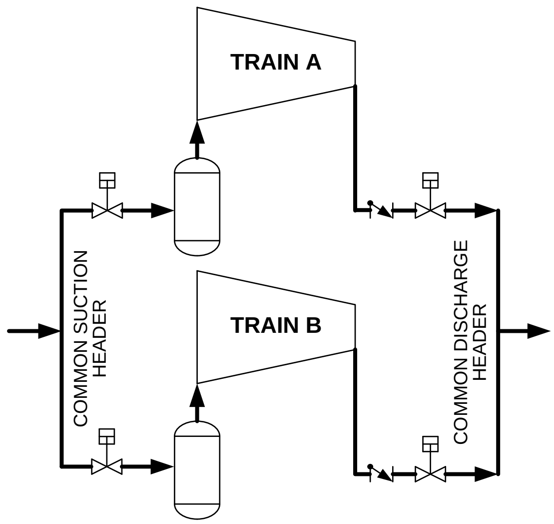

Figure 8

If two identical compressors are piped in a parallel arrangement, such as depicted in Figure 8, then ideally the total incoming gas feed would be split perfectly equally amongst the two running trains (A and B), with the result that the flow handled by each of the two trains would be the same, and hence the power consumption would be equal, also. This would represent the ideal load sharing between these two identical trains.

In real-life, such ideal conditions will never occur. There will probably be piping asymmetries between the two trains. Even if the two compressors were built to be identical, over time, their efficiencies and hence their performance at the same rotational speed value will cease to be identical. In brief, it is highly unlikely that these two supposedly identical trains will present exactly the same dynamic resistance to the gas in the common suction header. Hence more and more gas, over time, will tend to go through the path (train A or Train B) of lesser resistance, while the path presenting the slightly higher resistance will handle less and less gas, even to the point where recycle is required.

PARALLEL TRAIN OPERATION – HISTORICAL PRACTICES

Base Loading

Recognizing the implicit difficulty in achieving perfectly balanced trains, operators deliberately adopted the “base loading” approach to controlling parallel trains. Assume that the primary control objective was to maintain the common discharge header pressure at some desired set-point value. In a “base-loaded” arrangement, operators would manually run one of the two trains at a certain “fixed” performance, termed “base-loading” it; while allowing the other train to be modulated in response to the discharge pressure control requirements – thus becoming the “swing” train.

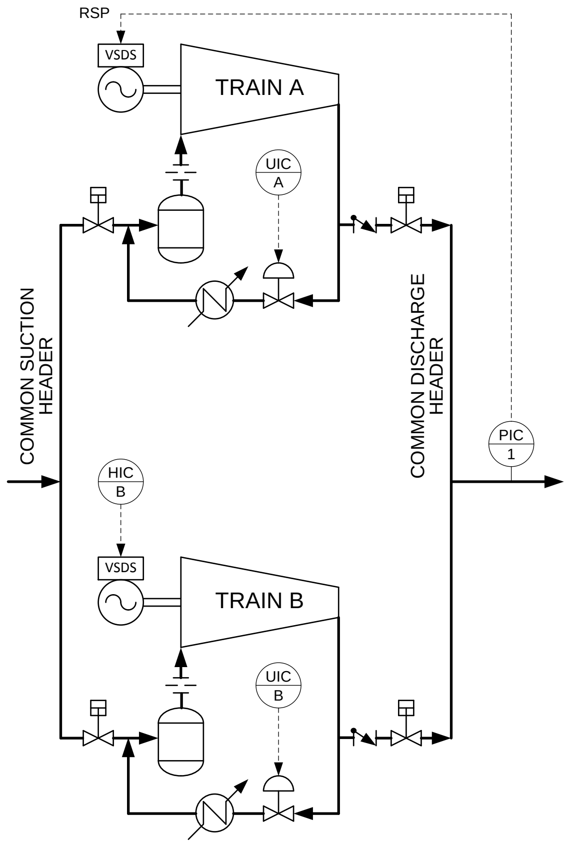

Figure 9

Typically, the operator would select the most efficient train and “fix” the speed of this “base-loaded” train either at the highest speed (to obtain its maximum flow), or at the highest efficiency, to optimize power consumption. The swing train would vary its speed in a cascade control arrangement with the common discharge line pressure (performance) controller. In other words, the swing train will absorb the load swings as the process demand changes.

Because of this arrangement, it is completely possible that the swing machine, during periods when the process load is slight, could be operating at low throughput and even requiring recycle; while the “base-loaded” train is still operating at high or maximum flowrates. This is illustrated in the following Figure 10. Base loading could result in easily being an inefficient manner of controlling the throughput of parallel trains, especially during periods of low process demand. Not only could the swing machine be recycling significantly, but is there is a surge-inducing incident, all of this disturbance will be borne by the swing machine, which is operating very close to its surge-limit line to begin with.

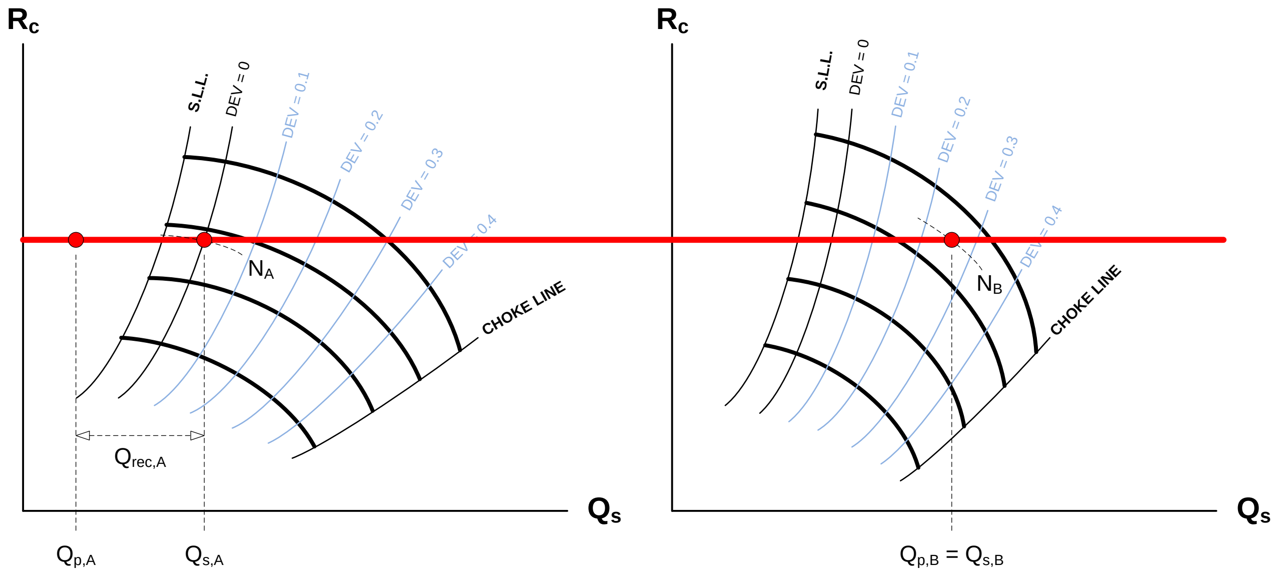

Figure 10

$Q_{p,A}$ & $Q_{s,A}$ are the process and compressor inlet volumetric flow rates, respectively, for Train A, the “swing” machine.

$Q_{p,B}$ & $Q_{s,B}$ are the process and compressor inlet volumetric flow rates, respectively, for Train B, the “base-loaded” machine.

It is, of course, possible for the operator to frequently conduct a study on the short-term process load, compare it to the capacities of the two compressor trains and, based on the results, re-fix the “base-loaded” machine’s speed (and hence throughput) in order to re-distribute the load amongst the two trains such that the “swing” train is no longer operating with recycle, as illustrated in the following Figure 11:

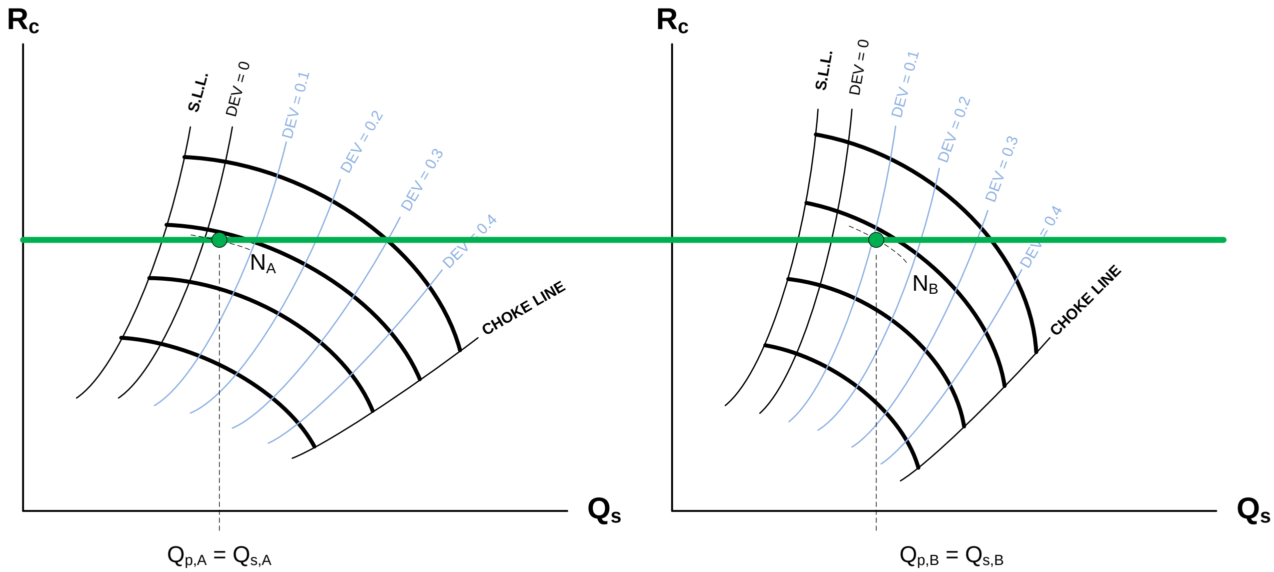

Figure 11

This, however, could hardly be termed “automatic load sharing”.

Flow Balancing

Recognizing the deficiencies of the “base loading” approach, designers introduced a more complex control arrangement designed to induce equal throughputs in the two trains, called Flow Balancing. In the Flow Balancing approach, the common discharge header pressure (performance) controller generates the total control demand to the two running trains simultaneously. A local train discharge flow controller receives this common control demand signal and scales it to use as its local train flow set-point. The other train is equipped with its own local discharge flow controller and receives the same common control demand signal that is scaled appropriate to its own capacity range. This is illustrated in the following Figure 12.

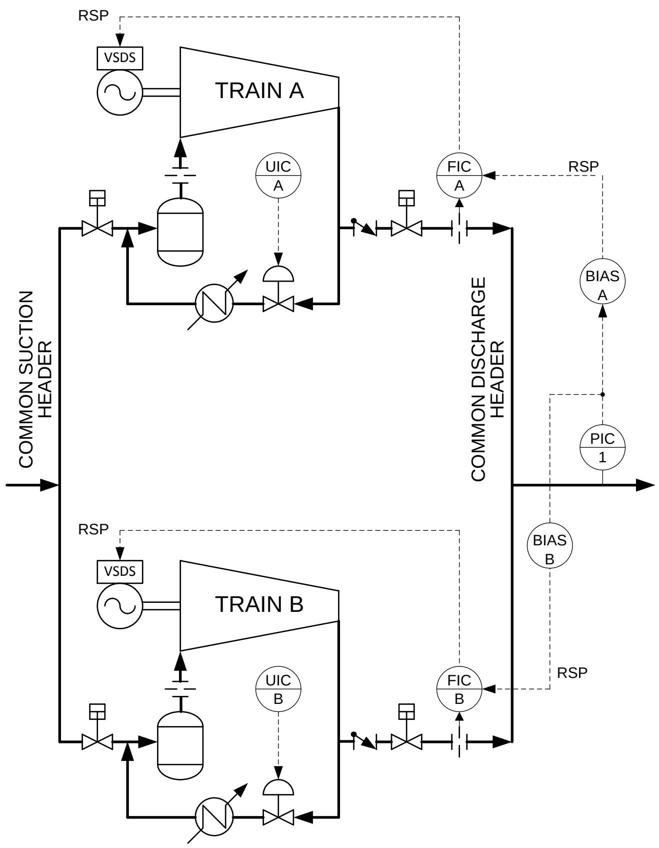

Figure 12

The Flow Balancing approach requires additional flow elements, which represents additional capital expenditure and additional energy losses.

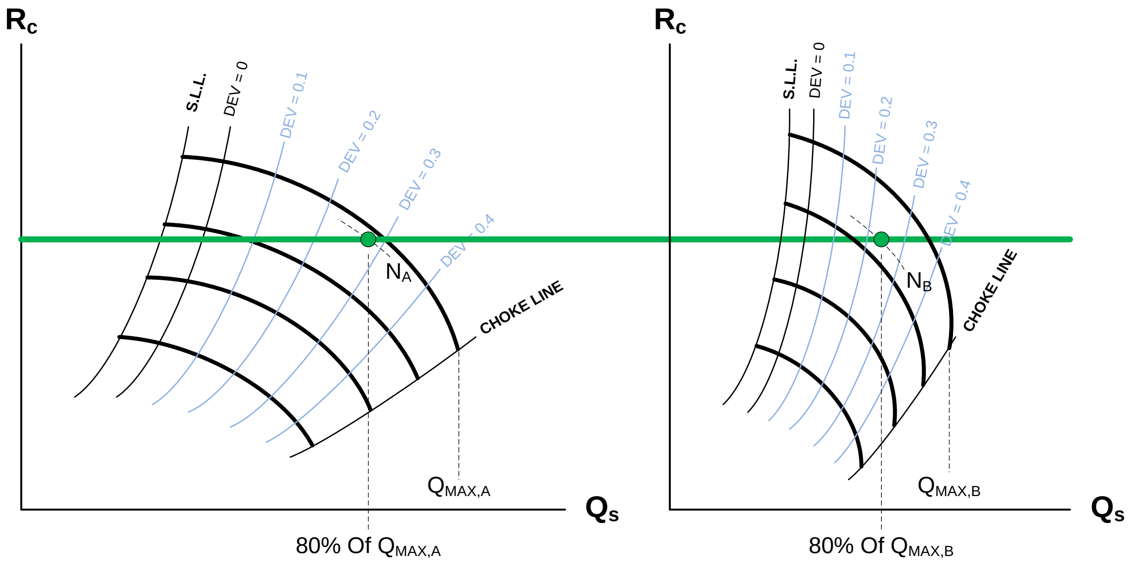

Figure 13

In addition, while it is possible to accommodate compressors with dissimilar performance maps using remote flow set-point bias modules for each train (as illustrated in Figure 123) this does not address the fact that dissimilar trains may have significantly different surge points, so that even while equal flows (or percentage of maximum train flow) is achieved, the resulting operating points could be quite different in terms of proximity-to-surge, which could be problematic for low process load scenario as shown in Figure 14.

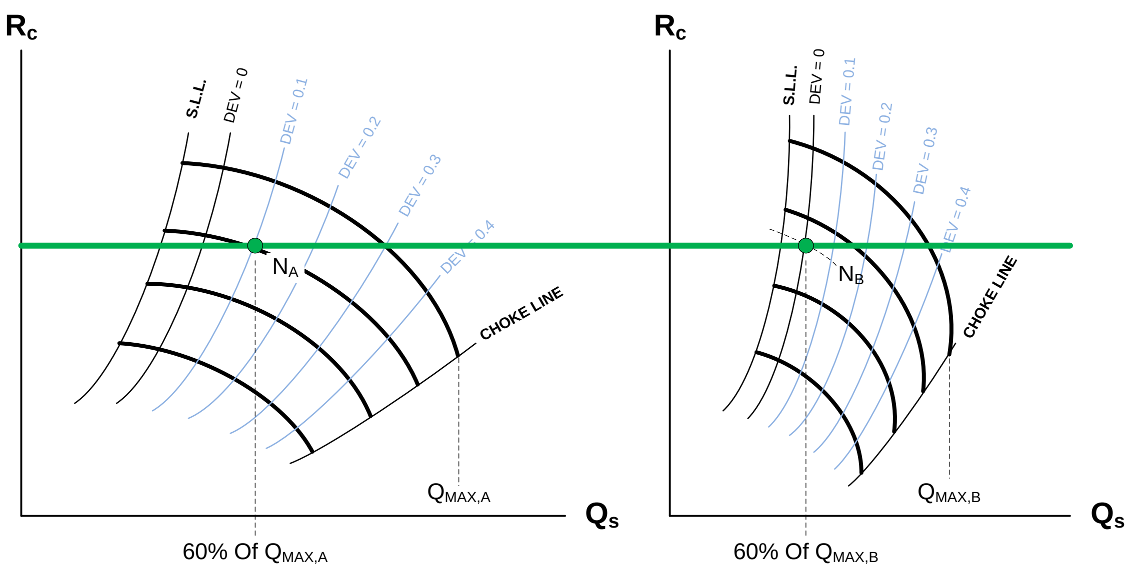

Figure 14

Positive Feedback System

In addition to the problems mentioned above, the depicted Flow Balancing scheme creates a "positive feedback system". From the perspective of each of the parallel trains, and referring to Figure 12, there are several cascaded control loops: The master controller provides a common pressure control response that is used by the train's discharge flow controller, in an "outer" cascade control loop arrangement; and the train's discharge flow controller provides a flow control response that is used by the train's speed controller in an "inner", and second, cascade control loop arrangement. See Figure 15.

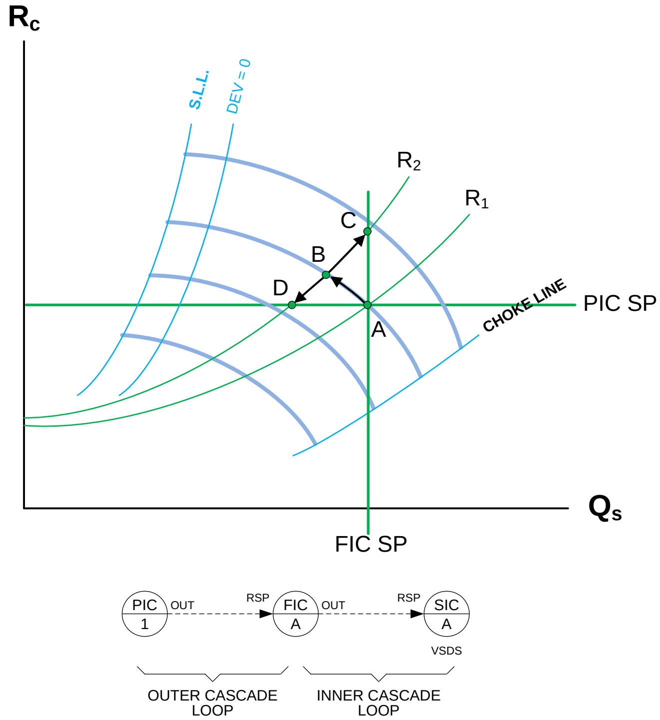

Figure 15

If we consider that the operating point initially, and during steady-state operations, lies at the intersection of two curves: the speed curve corresponding to the output of the SIC-A; and the curve representing the process resistance R1 imposed on the compressor performance envelope. The operating point will, at the same time, also lie on the intersection of two control lines: a line representing the set-point of the FIC-A and a line representing the set-point of the PIC-1. Now imagine that the process resistance line changes (increases) from a value R_1 to a value R_2.

The compressor's operating point will initially move up the initial speed curve from point A to point B, where the initial speed curve intersects with the higher process resistance curve R_2. The operating point lying at point B creates an error in *both controllers* PIC-1 and FIC-A. In a cascade control arrangement, the inner loop must be several times faster (typically five times) than the outer loop. Hence the error in FIC-A will generate a much larger control response (output) than the control response (output) of PIC-1.

This means that the operating point of the compressor will tend to increase initially – from point B to point C – in response to the faster change in the FIC-A output, before moving to its final steady state position of point D. This translates into the compressor – in response to a process disturbance that requires the driver to slow down, will initially speed up before ultimately slowing down, thus creating a Positive Feedback behavior.

This positive feedback initial response to a process upset could result in the train unnecessarily reaching maximum speed, and will initially magnify the pressure disturbance before restoring it the set-point. This could prove quite problematic for the safe operation of the process.

The only remedy to attenuate the "positive feedback" behavior is to deliberately slow the control response of the "inner" controller (the FIC-A) to a degree that the initial speed excursion in the opposite direction is rendered tolerable, which means that, by the rules of cascade control tuning, the "outer" loop, the PIC-1 will have to be tuned at least 5 times slower than the before the FIC-A. Thus, the positive feedback behavior can only be mitigated by forcing the PIC to be very sluggish. This is not conducive to good process control.

Equidistant-to-surge Load Balancing

A better approach to balancing the load of parallel trains is to equalize their proximity-to-surge variables or DEV values. Refer to the following Figure 16:

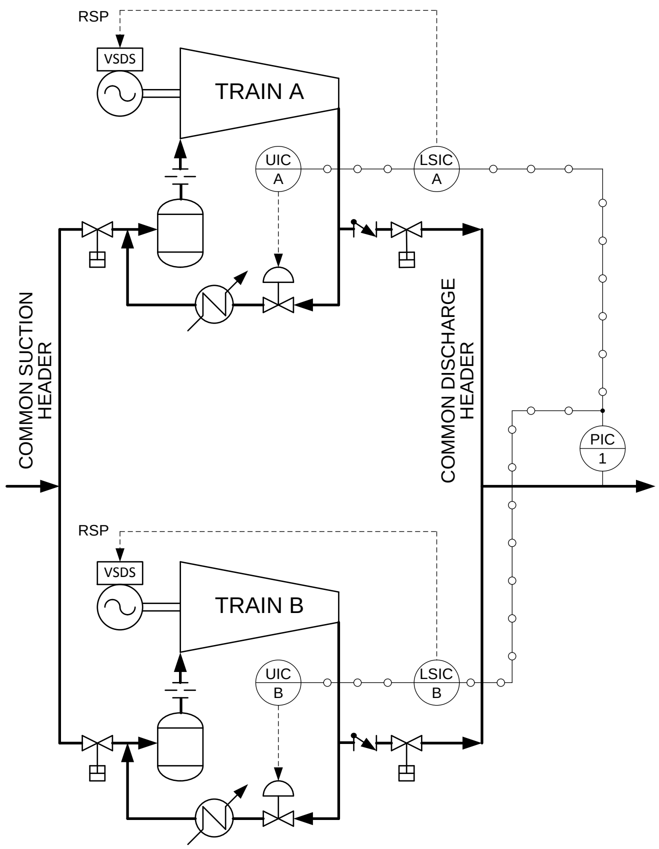

Figure 16

The antisurge controller UIC-A calculates and sends its train's proximity-to-surge (DEV) value to train A's Load Sharing Controller LSIC-A. Likewise, the antisurge controller UIC-B calculates and sends its train's proximity-to-surge (DEV) value to train B's Load Sharing Controller LSIC-B. If train A or train B have multiple stages with separate antisurge controllers, then each train should select the stage with the lowest DEV value (i.e. closest to the Surge Control Line) to represent the DEV value for the whole train. Each train's Load Sharing Controller will then send its selected DEV value to the Master Performance Controller (PIC-1) which will calculate the arithmetic average of these values (DEV´). Each Load Sharing controller will use the train's DEV value as the process variable in its Load Balancing PID loop and compare the DEV value with the average DEV´ received from the Master Performance controller and used as the Load Balancing loop's set-point. The Load Sharing controller will then raise or lower the performance of the train so as to bring its DEV value equivalent to the DEV´ value.

This Equidistant Load Balancing method ensures that each train operates the same distance to its own Surge Control Line, as illustrated in the following Figure 17:

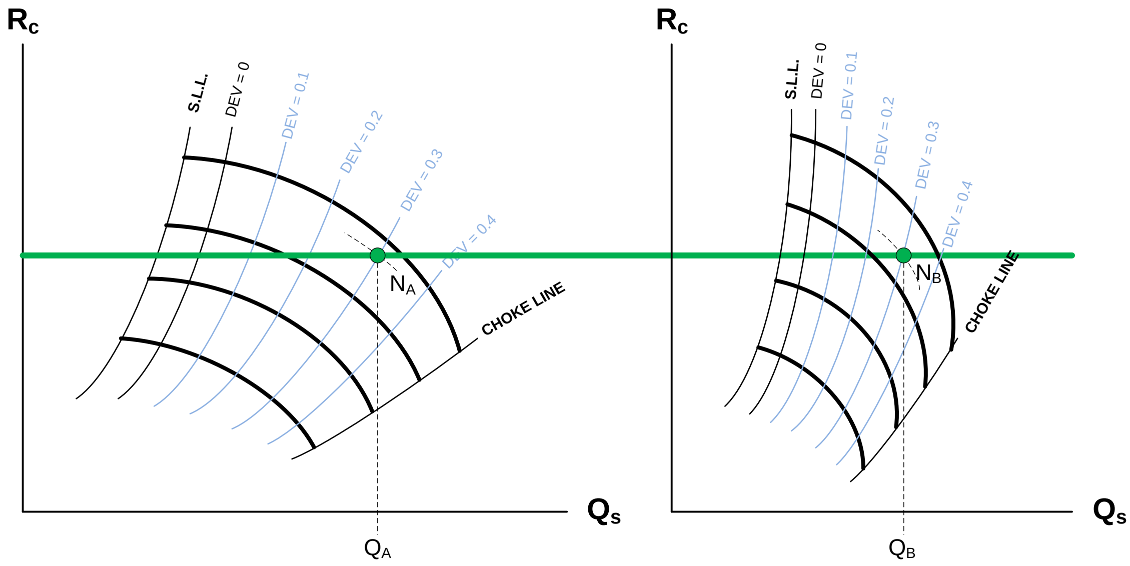

Figure 17

When the trains are operated in this Equidistant-from-surge arrangement, the flow rates (Q_A & Q_B) do not necessarily have to be the same, as the performance curves of the trains do not have to be identical, nor do the speeds of the trains have to be identical, as efficiencies will vary between the trains, and piping may not be exactly symmetrical. What this Equidistant-to-surge load balancing scheme ensures is that, for decreasing overall load, each train's operating point will reach its Surge Control Line at the same time, thus significantly decreasing the risk of surge for all trains. This is illustrated in the following Figure 18 :

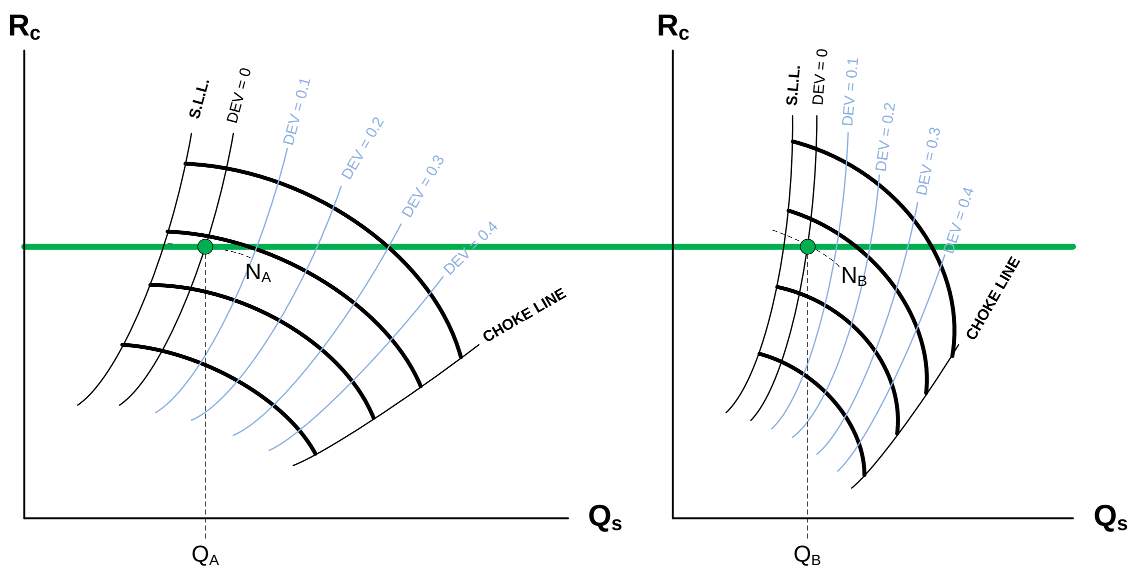

Figure 18

Note that this approach may be implemented when the running trains are operating to the right of their respective Surge Control Lines.

Recycle Balancing

Once the total process load is small enough that all trains start to recycle, the recycle rates for each train may differ, and even differ strongly, from the companion trains, even if the trains have identical performance maps. If this poses a problem for compressor operations, then the control system needs to be able to equalize the recycle rates for all running trains.

Load Sharing for Parallel Trains

The primary process variable that represents the combines capacity (throughput) of the compressor trains is processed in the Master Performance Controller as its PV. The Master Performance Controller compares this PV to the assigned set-point and generates a PID control response, called the Primary Response. A rising Primary Response signifies that all running trains must increase their individual capacities so that the PV equals the set-point, and vice versa.

Each train's Loadsharing Controller will allow this Primary Response to pass through in its appropriate block if that train's proximity-to-surge variable (DEV) is positive, i.e. if its operating point is to the right of its Surge Control Line. If that trains DEV value is zero or negative, the primary response coming from the Master Performance Controller will be frozen. This approach ensures that, if the Master Performance Controller requires a total decrease in throughput (capacity), this train will allow that request to lower its performance until its operating point reaches its Surge Control Line. Lowering the performance further (speed, IGV opening, …etc.) while the operating point is so close to the Surge Limit Line is likely to force the compressor to surge, and so is not recommended.

At the same time, that train's Antisurge Controller is calculating and broadcasting that train's proximity-to-surge value (DEV). If the DEV value is zero or negative, i.e. the operating point is on or to the left of the Surge Control Line, then the Primary Control Response coming from the Master Performance Controller is allowed to go through, after applying a suitable gain, and added to the Antisurge Controller's other control responses. This forces the antisurge valve to open more than needed for surge control, so that extra recycle is used to help reduce the train's throughput (capacity) and thereby help the Master Controller to restore its PV to the assigned SP value; when the train is operating close to its Surge Limit Line. By this means, the control system can safely demand a reduction of train throughput while it is operating close to its Surge Limit Line without the risk of surging. This is illustrated in the following Figure 19:

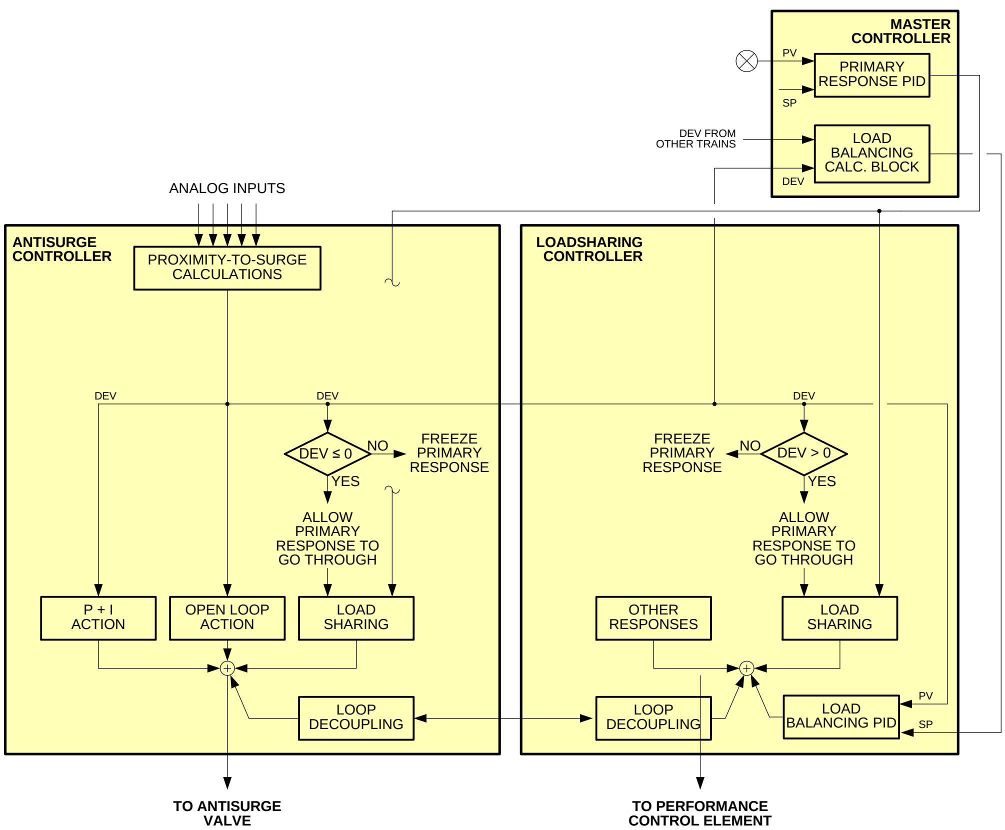

Figure 19

PIPING LAYOUT DESIGNS FOR PARALLEL COMPRESSOR TRAINS

A shared recycle path will reduce the effectiveness of the antisurge control system during normal operation, but the main issue for this layout is bringing units on and offline while other units are in operation. When one compressor is running and a second compressor needs to be brought online, the second compressor will most certainly surge. This is due to the running compressor taking all of the recycled flow of the starting compressor until it comes on line. The same is true for when one unit is taken off line.

See the following Figure 20. Train A is running and this is illustrated by the green piping, showing the gas being sucked from the shared suction drum, compressed and delivered to the shared discharge header. As train A is feeding forward with no recycle, its antisurge valve is closed. Train B is shut down and ready to start, with its antisurge valve fully open to facilitate startup. But, when train B is started up, as long as its discharge check valve (CV2) is closed, then it will present a higher resistance to gas flow than the path through Train A. Therefore, despite that the antisurge valve of Train B being wide open, there is a strong probability that Train B will surge, and continue surging, until its discharge check valve opens.

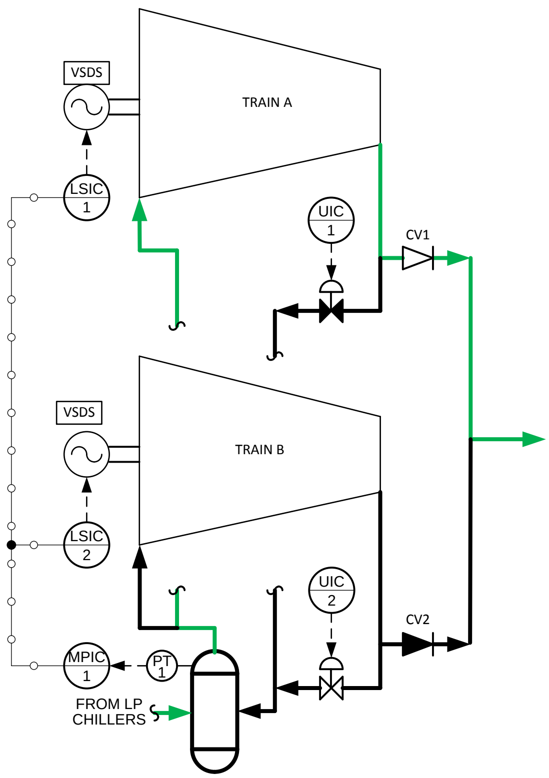

Figure 20

Having individual recycle paths but sharing a common suction drum can have similar results. This is especially true with closed loop parallel compressor systems, such as refrigeration cycles. Figure 20 illustrates a piping layout that incorporates common suction drums between two parallel refrigeration compressors. With this piping configuration, the effectiveness of the antisurge control system is reduced while both compressors are in operation. This is due to having part of the flow through an opening recycle valve absorbed by the other compressor(s). In addition, each antisurge valve will face the back pressure from the other running compressors. Another issue with this piping design is the difficulty starting and stopping an offline compressor when a parallel unit is in operation. A compressor that is being brought online or offline while the other is in operation has a high potential of surging since the running compressor will take the majority of the flow. An emergency shutdown (ESD) will result in the most severe effect, surging the machine from operating point, down to zero speed, and if not enough check valves/ block valves are present in the system, possibly, resulting additionally in reverse rotation of the compressor.

Therefore, piping designs for parallel compressor systems need to account for isolation of the individual compressor trains to allow for standalone operation, as well as successful startup and shutdown of each parallel compressor train. Block valves are used to isolate an offline compressor while discharge check valves are used to bring the compressor offline as well as online. When there are common suction and discharge headers, the compressors need to have individual recycle paths with individual suction knockout drums.

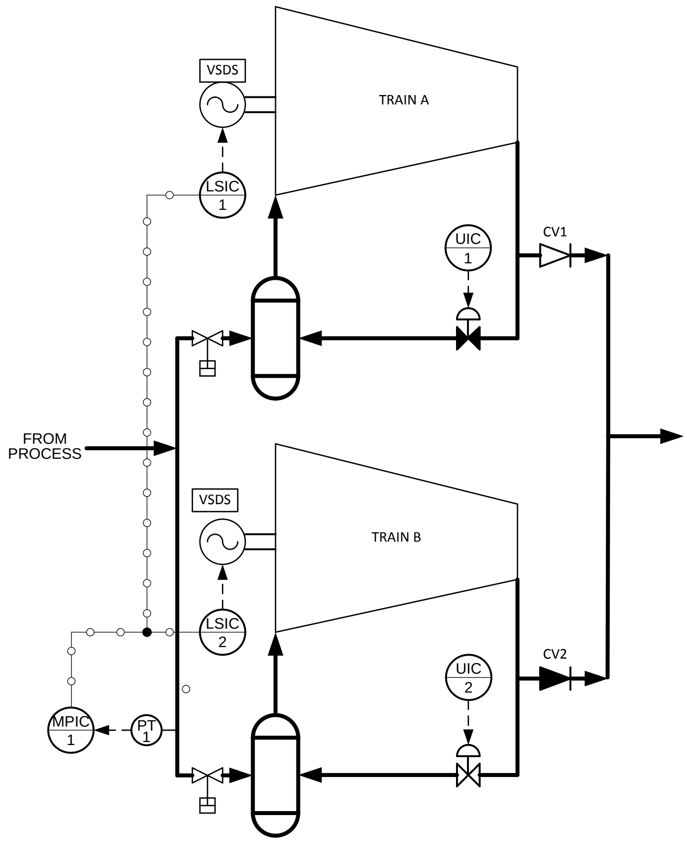

Figure 21

Having separate suction drums for each compressor eliminates the above potential risks associated with common suction drums. This is illustrated in the above Figure 21.

SERIES TRAIN OPERATION

Centrifugal compressors may be piped to operate in series. This is usually done to achieve a higher overall pressure ratio than can be produced by a single train. The primary characteristic of series trains is that each train has its own dedicated driver. In contrast, arranging several compressor stages on the same driven shaft only produces a multi-stage train. Please see Figure 22:

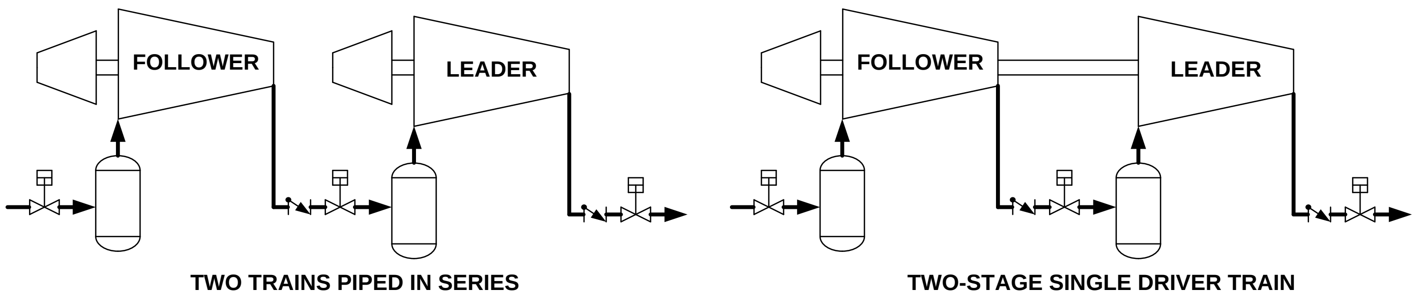

Figure 22

It is common practice in, for example, the gas collection and compression stations close to gas reservoirs, to begin production of the reservoir with a set of single stage trains, and then, as reservoir depletion takes place over time, to add upstream or downstream trains – piped in series, with their own dedicated drivers – to make the same pressure ratio as suction manifold pressure drops.

Load Balancing for Series Trains

When balancing series trains, it is not desirable to apply the same approach as for parallel trains; i.e. equalizing DEViation values is not the best approach. Instead, it is recommended to balance a Load Value (L), when the operating point of the compressor is to the right of the Surge Control Line. The most common Load Value is compressor pressure ratio (Rc), but for special cases, other variables may be more appropriate (such as proximity-to-power limit; speed; or inlet guide vane position, ….etc.).

To illustrate series Load Balancing based on compressor pressure ratios, consider two dissimilar trains, piped in series. In the absence of any sidestreams coming in or going out, then one may consider that the mass flow through the two series trains to be equal. In the following Figure we are considering a Leader train (i.e. with the higher operating pressure) and a Follower train (i.e. with a lower operating pressure). To represent a real-life scenario, the two trains' performance envelopes are significantly different. Consider first a steady-state situation where each train is running at an Rc value approximating 2.75, as illustrated in Figure 25:

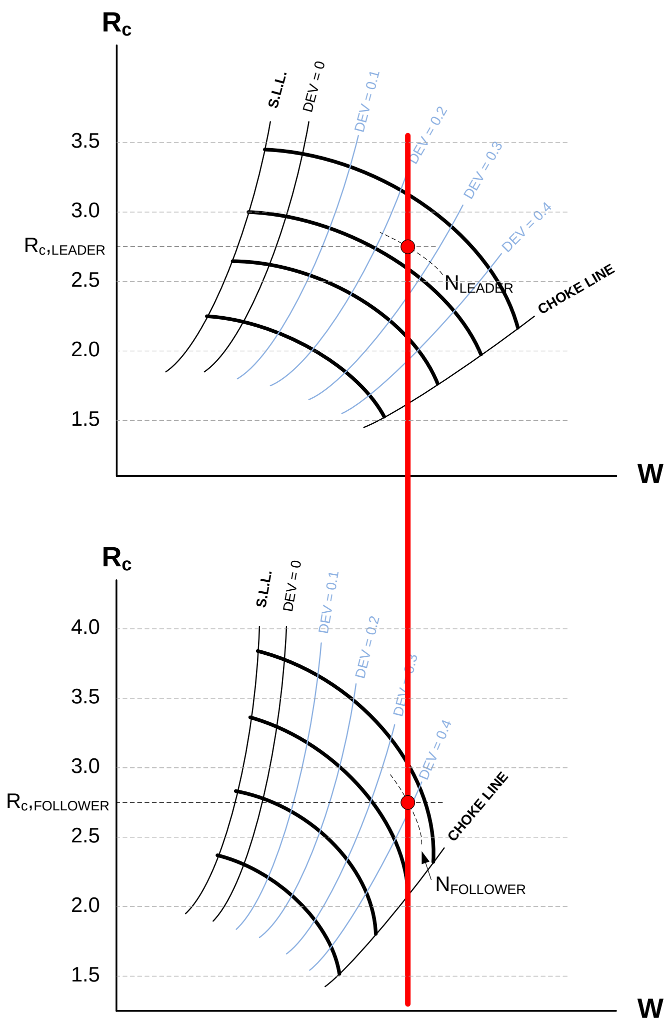

Figure 23

If the process conditions change, and it is necessary to operate with a lower mass flow through each train, and, at the same time, a higher pressure ratio, then the new operating condition may be illustrated with Figure 24, where the pressure ratio of each train rises to 3.75 using an R_C-based Series Load Balancing control scheme:

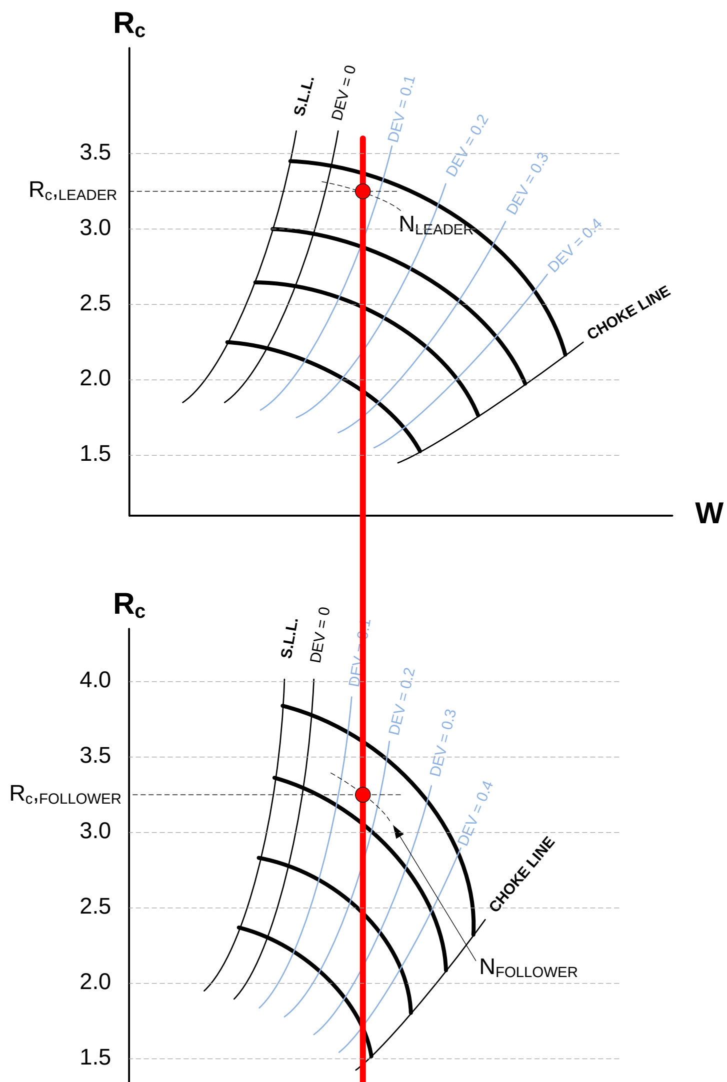

Figure 24

If process conditions force either (or both) the series trains to recycle, then it is recommended to implement Series Load Balancing based on equalizing recycle rates (the ratio of the recycle rate compared to the maximum capacity of the compressor).

Finally, it is desirable to have a transition area where the Series Load Balancing response transitions from L to the recycle rate. This is achieved by designating a low S threshold (called S_1) and a high S threshold (called S_2) around the Surge Control Line, as illustrated in the following Figure 25:

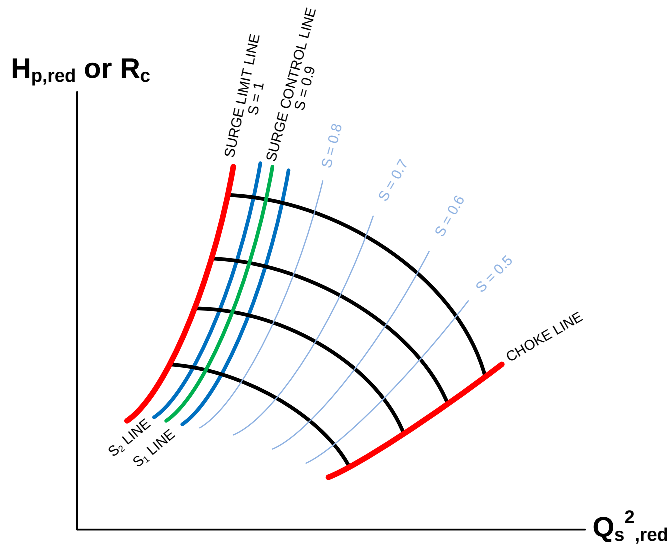

Figure 25

Load Sharing for Series Trains

The Load Sharing response for series trains is identical to that for parallel trains. The only difference is in the Load Balancing approach, and this is illustrated in the following Figure 26:

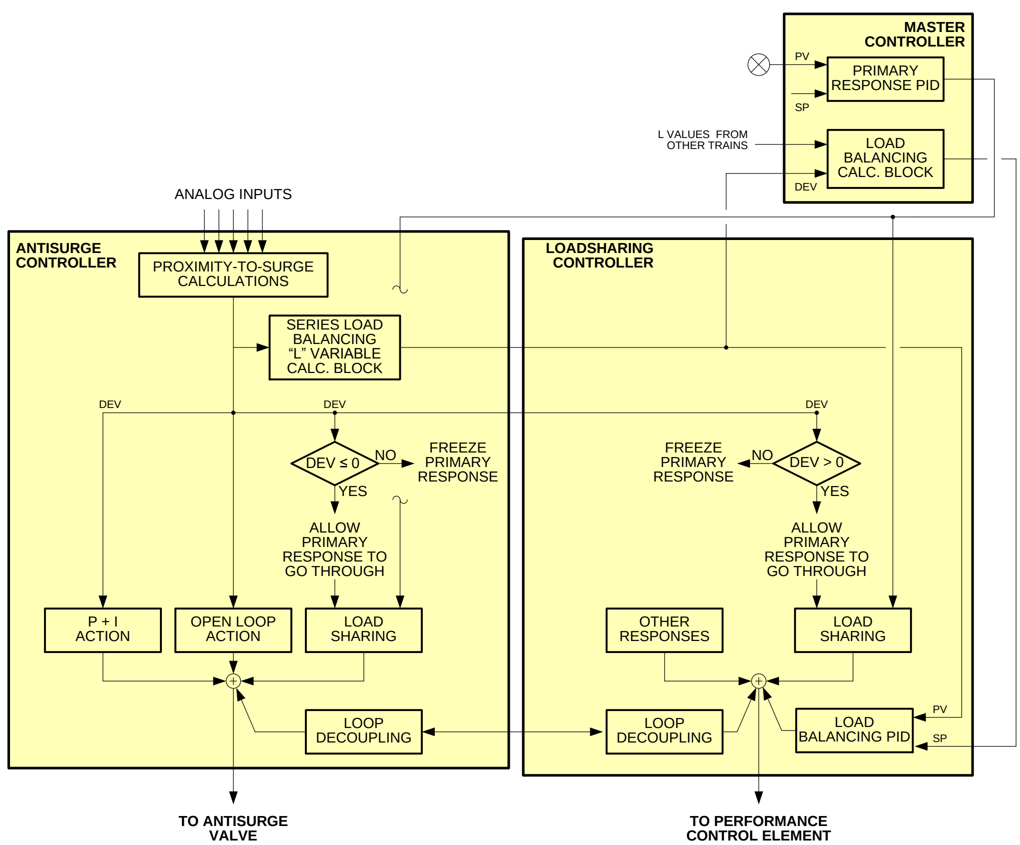

Figure 26

DEFINING SURGE

Many definitions of surge are used, but they all settle on stating that surge is an event in which the energy contained in the gas being compressed becomes equal, or larger than the energy imparted by the rotating impeller (or, set of blades) of the compressor. Essentially, energy accumulated in the compressed gas becomes greater than the energy level the rotating impeller of the compressor can sustain, and, as a result, the gas rapidly expands in an abnormal direction back through the compressor. Stall is another well-known phenomenon; at times it precedes the surge event and it may lead to the onset of surge. Surge is a very rapidly occurring event of gas expansion through the compressor, in most cases a loud sound can be heard by the observer. Once the gas energy in the volume associated with the discharge of the compressor is dissipated by flowing back to the suction of the compressor, the operation of the compressor is returning back to normal, but if preventive measures are not taken, the compressor eventually proceeds back into the next surge. As a result, surge that is not aborted, becomes a cyclical event, with frequency dependent on the design of the compressor and, on piping and volumes of the process system (typically observed, surge cycle times are 1-5 sec. for centrifugal compressors, and 3-20 sec, for axial compressors). Figure 27a illustrates trajectory of the operating point of the typical compressor during surge cycle in the Compressor Map, while changes in discharge pressure, flow and discharge temperature for a typical compressor running in a surge cycle are shown in Figure 27b.

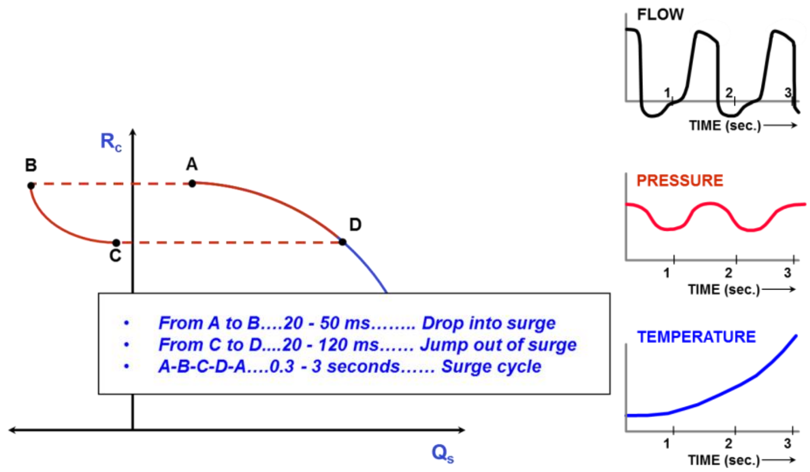

Figure 27a and 27b. Surge cycle and signal traces during surge cycling

Consequences of surge are: complete defeat and reversal of the balancing design of the rotating bundle of the compressor leading to abnormally high loads on bearings (especially thrust bearings) and seals; exceptionally high levels of vibration; rapid rise in the internal temperature of gas, and consequently, a rise in the internal temperature of the compressor; inability to meet flow and pressure demands of the process. For the majority of modern compressors surge is a highly damaging, sometimes a destructive event. The degree of damage to compressors during surge varies depending on design and application, and statistics are incomplete for obvious reasons. However, it is a common industry belief that a ½ hour of accumulated surge is detrimental enough for the compressor to become damaged beyond acceptable level. Operation of the compressor to the left of the SLL must be avoided and a surge event that evolved into surge cycling must be stopped immediately.

LOCATION OF SURGE DETECTION INSTRUMENTATION

Instruments used for surge detection need to be located as close as possible to the compressor flanges, as shown below in Figure 28. Locating the points of process sensing as close to the compressor as possible provides the quickest and most accurate information about the condition of the compressor. When selecting the location for the transmitter, vibration at the point of installation must also be considered. Installation on or near rotating machinery, or on a segment of pipe affected by vibration may produce adverse effects on the transmitter and its signal. Flow measurement in particular is susceptible to noise. Often, poor installation such as not allowing enough straight run piping upstream or downstream of the flow measuring device results in signal that is too noisy. Figure 29 is an actual recording of a noisy flow dP signal that caused false surge detection on rate of change.

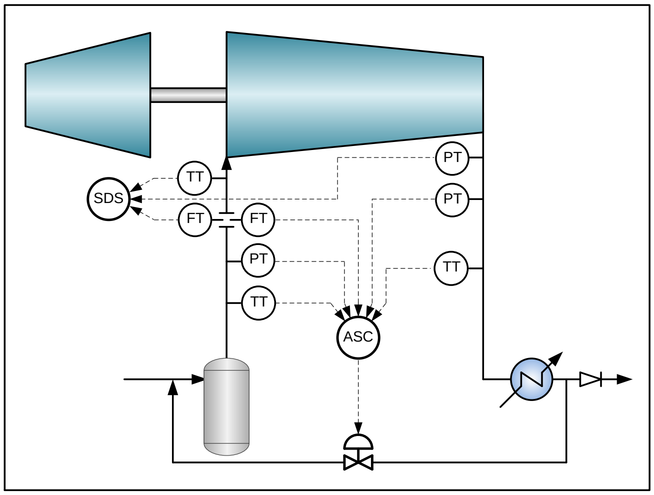

Figure 28. Transmitter tap locations

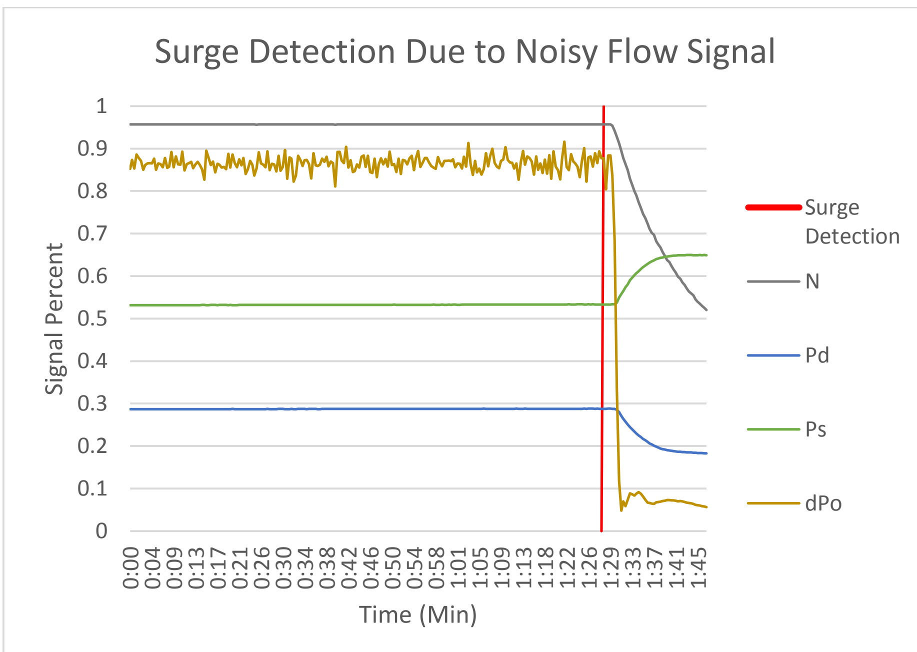

Figure 29. False surge detection

SURGE EXAMPLES

Reactor Effluent Compressor

Figure 30 below is a recording of a two stage Reactor Effluent compressor surging. It is a variable speed axial compressor with fixed vane angle. The molecular weight is 30.1 with a makeup of 50% H2, 25% Butane and 25% Butylene. The compression ratio (Rc) prior to surge was 3.6. This particular compressor has two flow measuring devices. One is measuring the dP across an inlet eye and is for low flow trip. The other is measuring the flow dP across a venturi located in the discharge of the compressor and is used by the antisurge control system. The compressor surges five times before it is tripped. The first surge event is severe due to the stored energy of the system and is followed by a series of less severe surge events. Flow dropping to zero on the first surge event is typical for this type of compressor. Both the suction and discharge flow measurements have similar rates of change. The discharge flow rate of change in the negative direction is -115%/sec. In the positive direction, the rate of change is +196%/sec. The suction flow rate of change in the negative direction is -235%/sec. In the positive direction, the rate of change is +137%/sec. The determination of rates or change have some error due to the limitation of a 200msec recording rate.

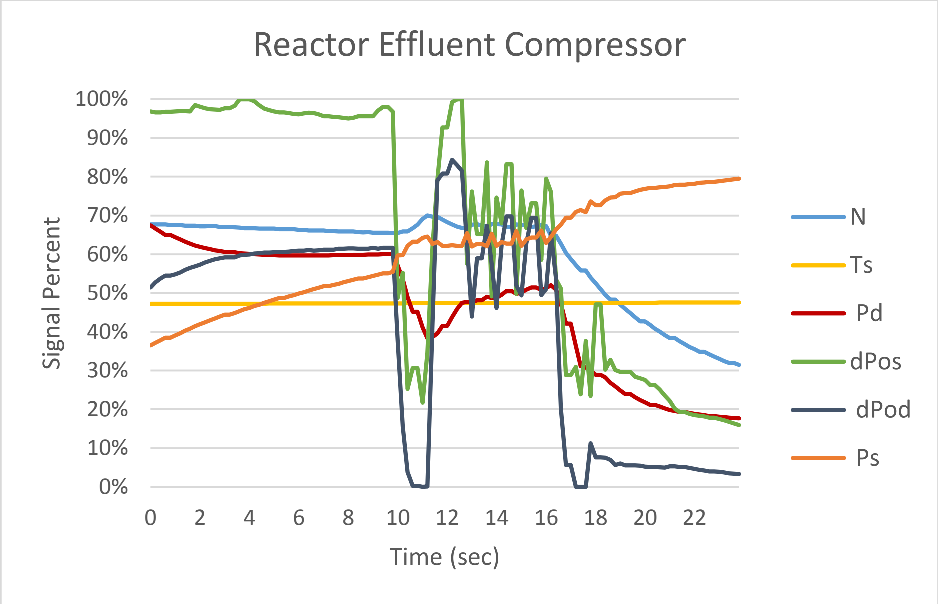

Figure 30. Surging of two stage Reactor Effluent compressor

There is also a drop in discharge pressure with a rate of change equal to -23%/sec. Small spikes in suction pressure were also recorded with rates of change +21%.

Two Section Nitric Acid Air Compressor

Figure 31 is a trend of a surge event from a Nitric Acid variable speed air compressor. The centrifugal compressor consists of two section bodies with the fist being a single stage and the second having two stages. The overall compression ratio at the time of the surge event was 2.5. The compressor is equipped with two flow measurements, an inlet eye in the suction of the first section and an inlet eye in the suction of the second section. The first section inlet eye flow dP measurement is used for surge detection as well as antisurge control. The trend of the surge event shows the 1st section flow dP initially increasing due to surge whereas the 2nd section flow dP decreases. Figure 32 shows the recorded rates of change of 1st section's flow dP measurement from the Surge Detector. The recorded rates where +49%/sec.

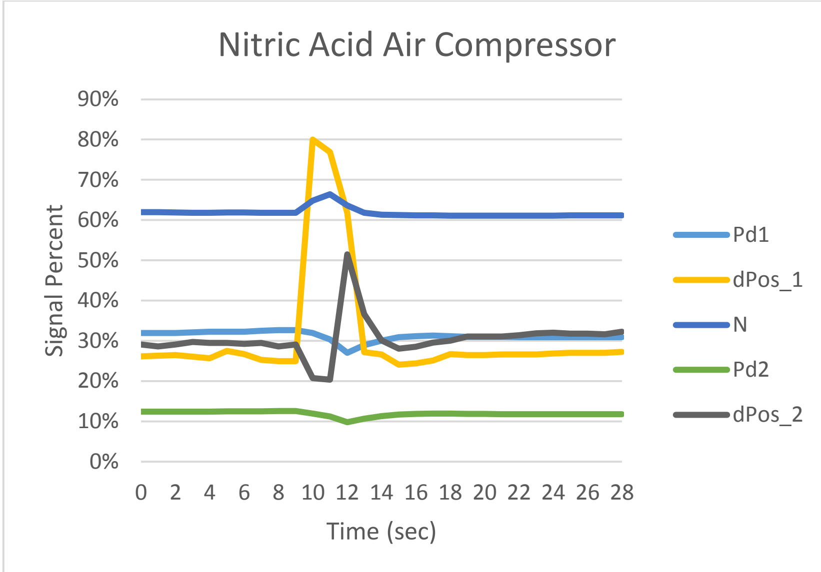

Figure 31. Surge event from a two section Nitric Acid air compressor

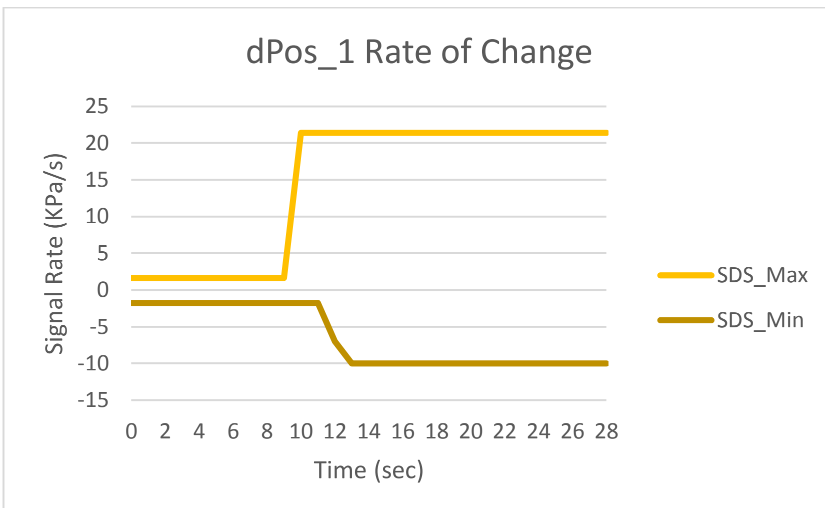

Figure 32. Rate of change recording from Surge Detector

Two Section Natural Gas Boil Off Compressor

Figure 33 is a trend of a surge event from a constant speed Natural Gas Boil Off compressor. The compressor consists of an axial section with inlet guide vanes and a centrifugal section. The overall compression ratio at the time of the surge event was 45. Flow measurement is in the discharge of the second section. In this example, both motor current and suction pressure could be used for surge detection in addition for flow dP. The rate of change for motor current was -39%/sec and the rate of change for suction pressure was +29%/sec. The rate of change for discharge flow was -38%/sec.

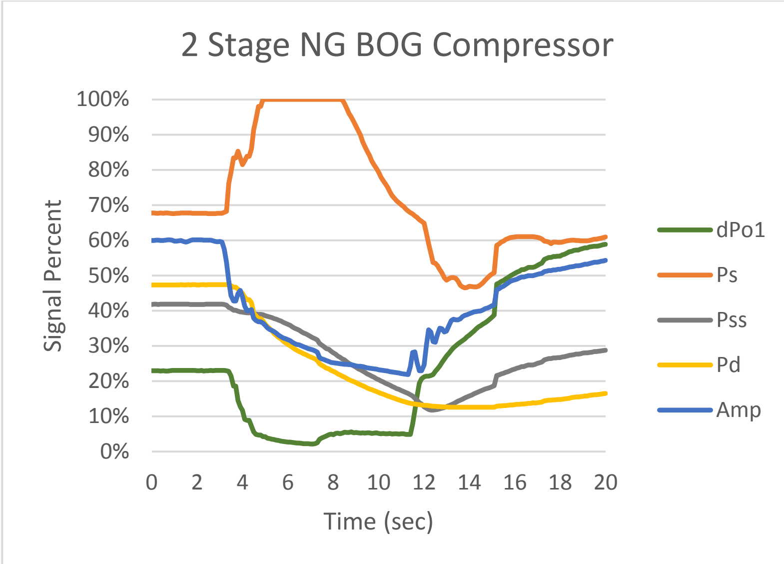

Figure 33. Surge event from NG Boil Off compressor

Three Stage Mixed Refrigerant Compressor

The below example is from a three stage Mixed Refrigerant (MR) compressor. Each stage has flow measurement in the discharge. The compression ratio of the LP stage is approximately 7 and the around 2 for both the MP and HP stages. The calculated rates of change are -126%/sec for the LP stage, -36%/sec for the MP stage and -92%/sec for the HP stage. Rapid changes in suction and discharge pressure for the LP compressor stage are not seen. This is primarily due to the large volumes associated with this application. Small spikes in discharge pressure for the MP and HP stages are seen.

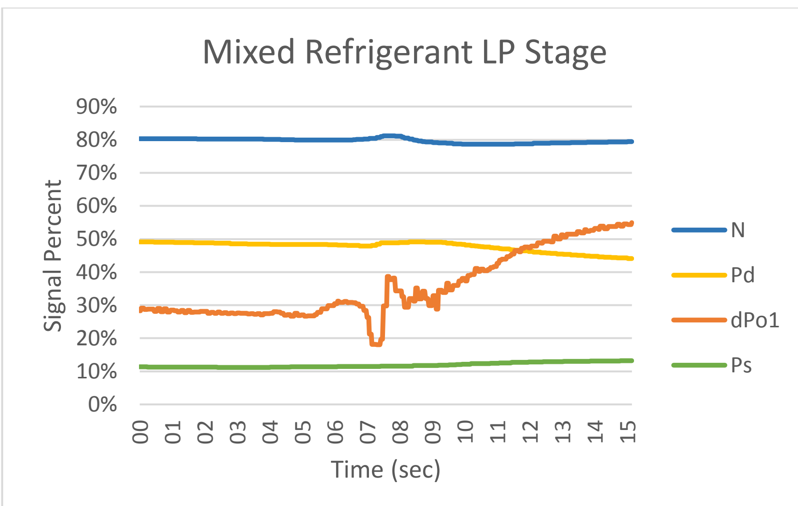

Figure 34. Surge event from MR compressor LP stage

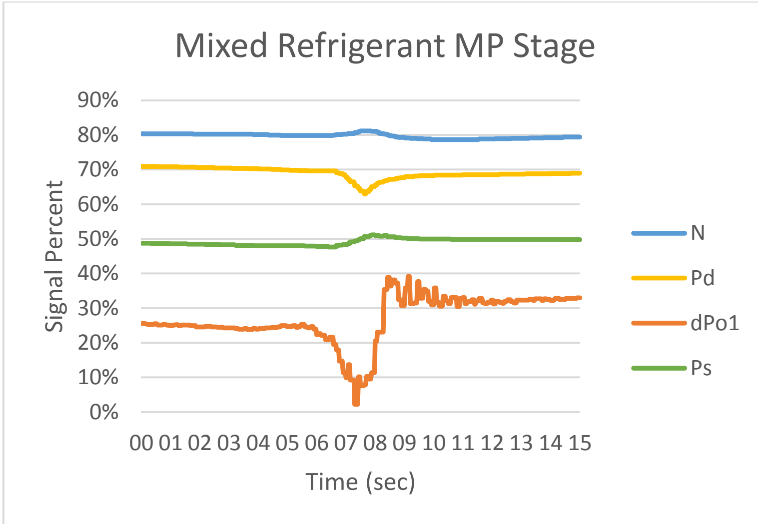

Figure 35. Surge event from MR compressor MP stage

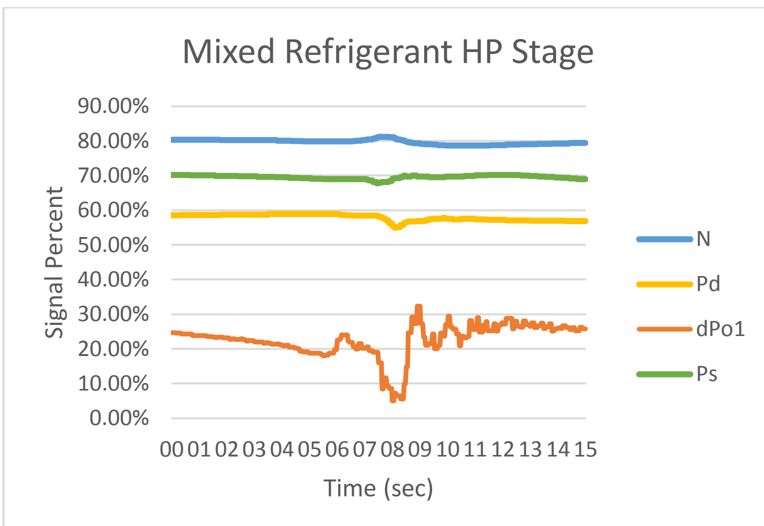

Figure 36. Surge event from MR compressor HP stage

Four Stage Propane Refrigeration Compressor

The last example is from a four stage Propane Refrigeration (PR) compressor. There are flow measurements in the suction, in the first two inlet sidestreams and one in the discharge. The compression ratio for the first three stages is approximately 1.7 and the last stage is 2.7. For this surge event, the third stage surges first and the 2nd and 4th stages quickly follow. Although the 1st stage is far from its surge limit line, flow oscillations are still seen in the suction of the compressor. The fourth stage remains in deep surge for 4.5 seconds while we see oscillations and spikes from the sidestream flow measurements. The calculated rates of change are -33%/sec for the 2nd stage, -78%kPa/sec for the 3rd stage and -58%/sec for the 4th stage.

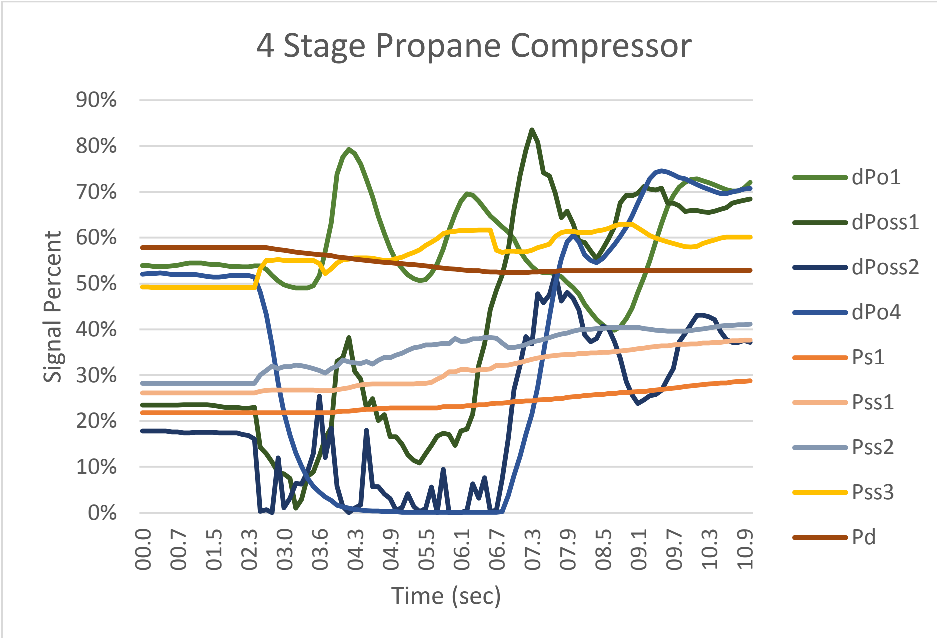

Figure 37. Surge event from a Propane refrigeration compressor

The sidestream flow measurements exhibit high frequency noise after the initial surge event and will need to be filtered. Rapid changes for the suction and discharge pressure of the compressor are not present. Similar to the MR compressor example above, this is due to the large volumes associated with this application. For this type of compressor, using rate of change in pressure for surge detection may not be a viable option.

From the above surge event data, it can be concluded that the flow dP signal will exhibit rapid rates of change in both the negative and positive directions for most compressor applications. The necessary rate of change signature may not always be present for other signals such as discharge pressure and suction pressure. These signals need to be evaluated per individual application to see which signal(s) may be appropriate.

Discussion of a specific example of compressor surge

Figure 38 shows another example of a surge event on a single section variable speed Residue Gas Compressor.

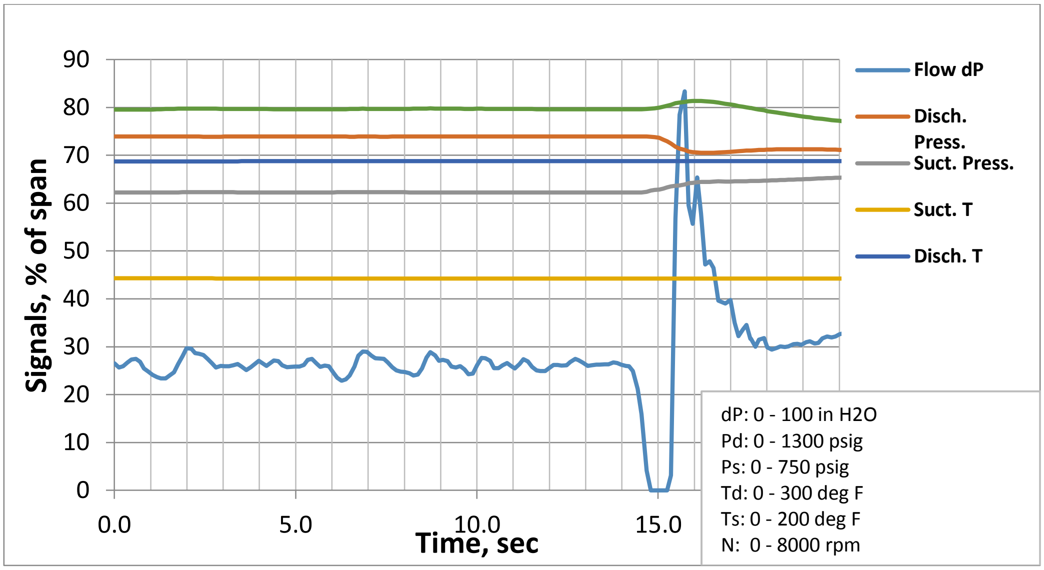

Figure 38. Surge event on a Residue Gas Compressor

During surge flow decreases rapidly, followed by a decrease in discharge pressure, a rise in suction pressure, as well as a rise in compressor rotating speed. Figure 39 shows the flow dP (d_dt_dPo) and discharge pressure (d_dt_Pd) derivatives. The dP signal right after the surge event contains high frequency components that can trigger false detection. Therefore, using combination of the signals, for example, both the flow dP and discharge pressure signals exceeding their derivative thresholds within certain time of each other, will provide a more robust surge detection method. In addition, the derivative of the flow dP signal should be calculated using a low pass filter in order to decrease the amplitude of the high frequency components. An example of the filtered derivative is shown in Figure 39 as d_dt_dPo_f.

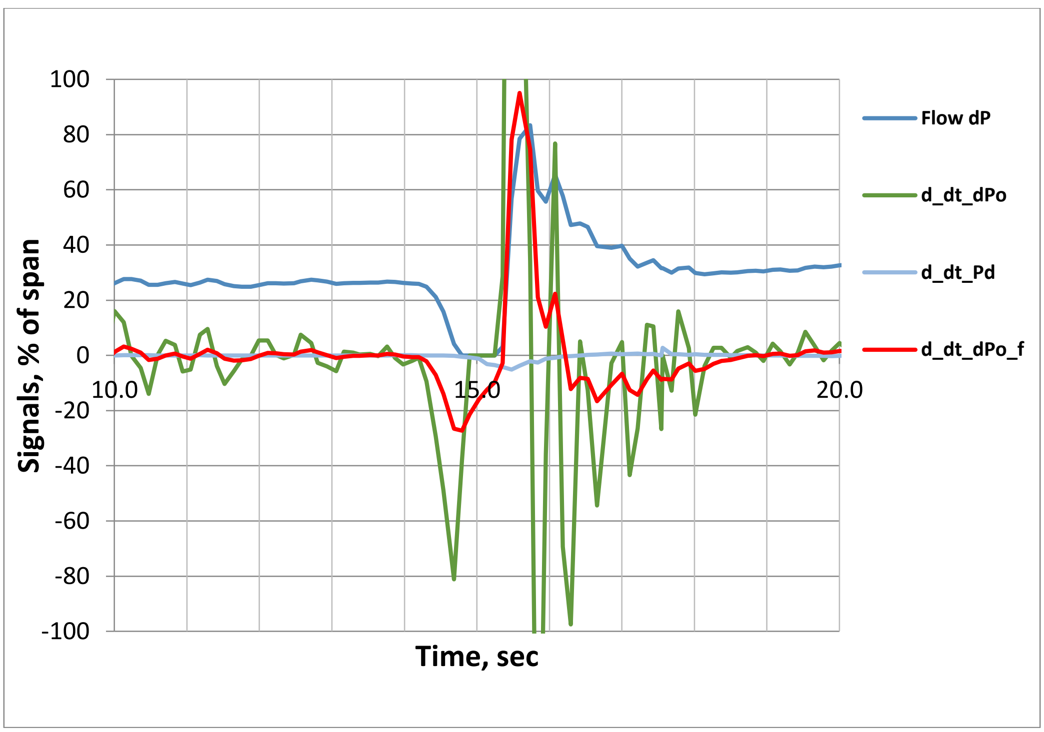

Figure 39. Flow dP and discharge pressure derivatives in % of span/sec

SURGE DETECTOR REQUIREMENTS

Generally, the occurrence of surge is inferred from a rapid change in any measurement that is a direct function of compressor flow or head. As mentioned previously, compressor flow, discharge pressure, and inlet temperature are signals that can be used for surge detection. However, based on experience, suction pressure can also exhibit surge induced oscillations. The Surge Detection System (SDS) should have several analog input channels available that can accept any of the above measurements. Since compressor flow dP will exhibit the most telling surge signature in almost all applications, the flow measurement should always be considered for surge detection. Other measurements such as discharge and suction pressure or suction temperature are selected based on the evaluation of the specific installation. In some instances, flow measurement may not be available. In these cases, motor current or power may be used instead of flow dP, provided that the signal produces a measurable surge signature. The SDS needs to accommodate any combination of these signals.

Functionality

The SDS must be able to measure the rate of change for each analog input channel. Each channel will have a rate of change threshold that represents a surge condition. The rate-of-change thresholds could be negative or positive to depending on the observed surge signature. In some instances, a compressor may not recover after a surge cycle, due to, for example, failure of the check valve, or mechanical damage. In order to protect against such eventuality, the SDS should also include minimum flow protection. If flow dP remains below configured minimum level after the first cycle, the surge counter is incremented every configured time period. The time period is an estimate of the surge cycle duration.

The SDS should be flexible on how it uses the different analog inputs for triggering surge detection. Depending on selected surge detection mode, detection may be triggered by monitoring either one or a combination of its analog input signals. When a combination of inputs is used, surge can be detected when any one input exceeds their rates of change threshold or only when all inputs exceed their individual rate of change thresholds within a specified time period. When using the mode where any one input is used for surge detection, a specified time period between surge counts is needed in order to avoid duplicate surge counts from the different inputs. If more than two inputs are used, more complex arrangements can be designed into the system. Detection should be disabled during compressor startup and shutdown in order to avoid false detection at low signal levels, where the machine characteristics are not well defined.

It is imperative then that SDS can differentiate between normal process changes, signal failures, and surge. For this purpose, using a combination of signals listed above can increase reliability of the detection. For example, concurrent reduction of flow and discharge pressure on a running machine signifies surge with a larger degree of certainty than just a flow reduction. The SDS shall be equipped with two surge detection counters. Each time a surge is detected by the SDS, two counters are incremented. One for counting the number of surges during a particular surge event and one for accumulated surge counts over time. The event surge counter resets to zero when the surge alarms are reset. The accumulated surge counter is not reset when the surge alarms are reset and is the total number of detected surges.

A surge alarm is required on the detection of the first surge cycle. Additionally, a Surge Event should be generated every time the surge count is incremented. It is recommended that an additional alarm for excessive surge is generated if the event surge counter exceeds a configurable threshold within a configurable time (for example 3 surges detected in 10 sec). The excessive surge discrete output is typically used as a condition to trip the compressor. For centrifugal compressors, the excessive surge threshold is typically set to three surge counts within 10 to 20 seconds. Axial compressors may be set to trip on the first or second surge detected depending on the compressor manufacturer's requirements.

Surge and Excessive Surge alarms as well as Surge Events are captured in a log that is viewable by operators. In addition, the log needs to include parameter values and their rates of change at the time of the alarm/event. It is desirable to have surge event data accessible for external analysis. An Event Recorder is necessary to archive each surge event and store the data in a buffer. At a minimum, the data consists of input channel values and their rates of change as well as the Surge and Excessive Surge alarms. The data should be recorded with a sampling rate of no more than 20ms. The recorder should capture the data at least 10 seconds before the event and 10 seconds after the event. For each recorded event archived, a file is generated which allows the data to be viewed in a standard format, such as a csv file and can be uploaded to a computer.

Analog Inputs

The SDS should have a minimum of two analog inputs available for surge detection. The scan rate for the inputs needs to at least 5ms. A low-pass filter, such as an adjustable first order filter should be implemented for each input. This is very important particularly for the flow dP. Flow dP may contain significant noise component due the nature of the typical measuring techniques. The elimination of the noise component is one the critical issues in surge detection. SDS should also provide a signal validity check, based on the signal level, before signals are used for detection. A failed input needs to be automatically excluded from detection with associated alarm and failure flag.

Discrete Inputs

The following discrete input commands are recommended as a minimum;

- A Bypass or Disable input to enable or disable surge detection functionality based on compressor operating conditions and external command.
- A Reset command to reset the Surge and Excessive Surge alarms and the associated discrete outputs. The Event Surge counter and the highest positive and negative rates-of-change are also reset to zero values.

Discrete Outputs

The discrete outputs should be solid-state relays with at least one failsafe output for ESD. The outputs should be programmable. The following discrete outputs are recommended as a minimum;

- A latched discrete output that is set if Surge is detected. This output is reset by either the Reset discrete input or Reset command button from the SDS local interface.
- A latched discrete output that is set if Excessive Surge is detected. This output is also reset by either the Reset discrete input or Reset command button from the SDS local interface.
- A discrete output that indicates whether the surge detection function is Online (i.e. surge detection is active) or Disabled/Bypassed. This output is set by the Bypass or Disable discrete input.

CONCLUSIONS

Due to the limited nature of this Tutorial, only a general coverage of the topics of interest have been addressed, and much material remains beyond the scope of this paper. However, this Tutorial does provide general guidance, and in the authors' view, satisfies the objective that real life systems must be designed with a much higher attention to details than was previously thought. The control systems for parallel or series compressors are very demanding systems. Compressor surge is a rapid event and therefore it is important to ensure proper selection and location of devices and equipment associated with surge detection. Surge testing is the most reliable means of determining measurements and settings for surge detection. Flow measurement in the form of differential pressure across a flow measuring device should always be considered for surge detection based on rate of change of signal. Other measurements or methods may not prove to be viable for some applications and will need to be verified during surge testing of the compressor.

NOMENCLATURE

DEV = Deviation of the operating point from the SCL, DEV=0 when the operating point is on the SCL
H_p = Polytropic head (kJ/kg)
Q_s^2 = Suction volumetric flow, squared (ACMH)
dPo = Pressure differential across flow measuring device (orifice typical) , (kPa)
P = Absolute pressure (kPa)
T = Absolute Temperature (K)
Z = Gas compressibility factor
Rc = Compressor pressure ratio
σ = Polytropic exponent
R_o = Universal gas constant (8.3144 N*m/(kg*mole K))
MW = Gas molecular weight
$H_{p,red}$ = Reduced head (dimensionless)
$Q_s^2{}_{,red}$ = Reduced suction flow (dimensionless)
SDS = Surge Detection System
SLL = Surge Limit Line
LP = Low Pressure
MP = Medium Pressure
HP = High Pressure
MR = Mixed Refrigerant
PR = Propane Refrigerant
ESD = Emergency Shutdown

Subscripts:

s = Suction
d = Discharge

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