16

COMPRESSOR CONTROL*

16.1 INTRODUCTION

Every centrifugal or axial compressor has (at a given rotational speed and inlet conditions) a characteristic combination of maximum head and minimum flow beyond which it will surge. Preventing this damaging phenomenon is one of the most important tasks of a compressor control system.

The most common way to prevent surge is to recycle or blow off a portion of the flow to keep the compressor away from its surge limit. Unfortunately, such recycling extracts an economic penalty due to the cost of compressing this extra flow. So the control system must be able to determine accurately how close the compressor is to surging so it can maintain an adequate—but not excessive—recycle flow rate.

This task is complicated by the fact that the surge limit, in general, is not fixed with respect to a single variable such as pressure ratio or the pressure drop across a flowmeter. Instead, it is a complex function that also depends on gas composition, suction temperature and pressure, rotational speed, and guide vane angle. An understanding of the principles of integrated control and protection systems is thus extremely important to companies and industries operating turbocompressors.

16.2 CONTROL SYSTEM OBJECTIVES

An integrated compressor control system may have any or all of the following objectives:

- *Performance control:* maintaining a primary process variable (e.g., discharge pressure or mass flow rate) at a desired set point level

* Developed and contributed by Compressor Controls Corporation, Des Moines, Iowa.

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- *Surge protection:* preventing surge-induced compressor damage and process upsets without sacrificing energy efficiency or system capacity
- *Limiting control:* maintaining any process-limiting variables (such as drive motor current and fluid temperature) within safe or acceptable ranges
- *Loop decoupling:* minimizing adverse interactions between control functions
- *Load balancing:* distributing the overall compression load among several compressors in a multiple-compressor network
- *Event sequencing:* automating changes in process status by controlling such events as startup, shutdown, and purging operations
- *Online redundancy:* providing uninterrupted control in the event of a hardware failure
- *Host communication:* integrating the compressor control system into higher-level distributed or supervisory control systems.

The purpose of this segment is to highlight the conceptual framework within which these control objectives and methods of meeting them can be understood.

16.3 COMPRESSOR MAPS

An axial or centrifugal compressor raises the pressure of a gas via energy added to it. If inlet conditions, rotational speed, and guide vane angle are held constant, the amount of energy added per unit mass (polytropic head) of gas will depend only on the inlet volumetric flow rate. Thus, we can construct compressor characteristics in terms of polytropic head and volumetric flow in suction.

The specific mechanical energy of a fluid is defined as

$$e_m = \frac{p}{\rho} + \alpha \frac{V^2}{2} + gz \tag{16.1}$$

where p = static pressure
ρ = density
V = average (or bulk) velocity
g = gravitational constant
z = elevation

In most applications dealing with gases, the elevation component is insignificant. Thus, we need only be concerned with the first two terms of Eq. (16.1).

Because each of these forms of energy can be expressed as an equivalent elevation (head), they are often referred to as the *pressure head and velocity head.* As explained earlier, they will have units such as ft-lb_f/lb_m (kJ/kg) if expressed as specific energies, or simply ft (m) when expressed as elevations.

A compressor uses a two-stage process to increase the pressure of its process stream. First, the mechanical energy of the rotor is transferred to the fluid, resulting in its acceleration—and increasing its kinetic energy. Most of the velocity head is then converted to an increase in pressure head by decelerating the fluid through a diffuser.

The work added to the fluid in the compressor depends on the path that the state of the gas takes as it passes from suction to discharge. However, this energy can be characterized by choosing a particular path having the same end points as that of the actual fluid. The path often chosen is a *polytropic compression* path. The overall increase in fluid specific mechanical energy is referred to as the *polytropic head* (H_p). The ratio of the change in mechanical energy divided by the change in stagnation enthalpy (mechanical plus internal energy) is defined as the *polytropic efficiency.*

Unfortunately, it is not possible to measure polytropic head directly—it must be calculated as a function of fluid properties and several measurable process variables. By integrating the thermodynamic relationship for work over a polytropic path, it can be shown that

$$H_p = \frac{Z_a R_u T_s}{\mathrm{MW}} \frac{R_c^\sigma - 1}{\sigma} \tag{16.2}$$

where Z_a = average compressibility factor
R_u = universal gas constant
T_s = suction temperature (absolute)
MW = molecular weight
R_c = pressure ratio (p_d/p_s)
σ = exponent, $(k - 1)/k\eta_p$
k = ratio of specific heats (c_p/c_v)
η_p = polytropic efficiency

The polytropic head developed by a specific compressor will vary as a function of the inlet volumetric flow rate, rotational speed, position of the guide vanes, and suction conditions.

Like polytropic head, volumetric flow in suction must be calculated as a function of fluid properties and process variables that can be measured directly. For orifice plates and venturi meters (which are, perhaps, the most commonly used flow-measuring instruments), it can be shown that

$$Q_s^2 = \frac{Z_s R_u T_s}{\mathrm{MW}} \frac{\Delta p_{o,s}}{p_s} \tag{16.3}$$

where Z_s = suction compressibility factor
R_u = universal gas constant
T_s = suction temperature (absolute)
MW = molecular weight
$\Delta p_{o,s}$ = pressure drop across an orifice plate
p_s = suction pressure (absolute)

Thus, compressor performance is often illustrated by plotting characteristic curves in H_p vs. Q_s. For a given speed and inlet guide vane angle, this will produce a single performance curve at constant inlet conditions. By allowing either the rotational speed or guide vane angle to take a series of discrete values, we can generate a family of performance curves, which is called a *compressor map*. It is important to note that in the coordinates (Q_s,H_p), the performance curves are valid only for the given inlet conditions.

In a similar fashion, constant-resistance curves could be used to plot the amount of energy that must be added to the fluid to sustain a given flow rate. Each of these curves represents some possible combination of gas properties, inlet and outlet piping, valve positions, back pressures, and operating devices.

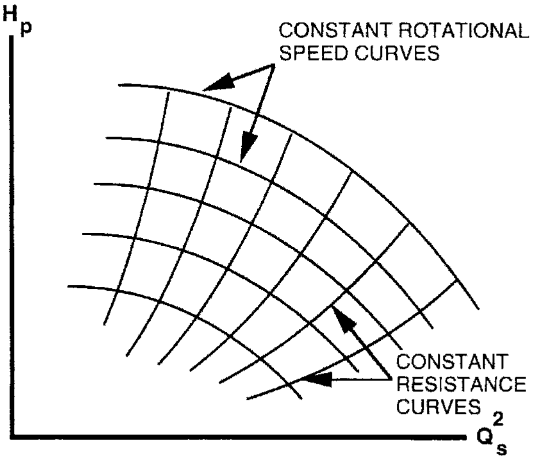

FIGURE 16.1 Typical compressor performance map. *(Compressor Controls Corporation, Des Moines, Iowa)*

The shape of the resistance curves depends on the characteristics of each specific application. For flow-through pipes, energy required will be approximately proportional to volumetric flow squared. Plotting resistance curves in the coordinates H_p vs. Q_s^2 would then yield a series of straight lines radiating from the origin. In general, however, the exact shape of the resistance curves will defy simple analysis. Fortunately, it is rarely necessary to know their exact shape, so we will represent them as a series of generic curves.

The performance map for a compressor system can be illustrated by superimposing both performance and constant resistance curves (see Fig. 16.1). At any given instant, the value of the compressor's independent variable (such as rotational speed) will determine which performance curve it operates along. Similarly, the resistance of the network at that instant will determine the position of the current line of constant resistance. The intersection of these two curves is called the *operating point.*

The flow rate at this point is such that the amount of energy added by the compressor is equal to that required to overcome the network resistance. The coordinates of the operating point thus represent the volumetric flow rate through, and polytropic head developed by, the compressor.

16.3.1 Invariant Coordinates

As noted above, the compressor performance curves in the coordinate system (Q_s, H_p) are unique for the suction conditions given. In practice, the inlet conditions are not constant, so for the purposes of control, the coordinates used must be invariant to changes in inlet conditions. Performing dimensional analysis on the parameters important to compressors results in several possible coordinates suitable for control. Two of these are presented here.

The first is *reduced polytropic head* (h_r) vs. *reduced flow rate* in suction (q_s). (For the same reasons as stated earlier, it is convenient to square the reduced flow rate.) These coordinates are defined as

$$h_r \equiv \frac{R_c^{\sigma} - 1}{\sigma} \tag{16.4}$$

$$q_s^2 \equiv \frac{Q_s^2(\mathrm{MW})}{Z_s R_u T_s} \propto \frac{\Delta p_{o,s}}{p_s} \tag{16.5}$$

where R_c = pressure ratio (p_d/p_s)
σ = exponent $(k - 1)/k\eta_p$
k = ratio of specific heats (c_p/c_v)
η_p = polytropic efficiency
MW = molecular weight
Z_s = suction compressibility factor
R_u = universal gas constant
T_s = suction temperature (absolute)
$\Delta p_{o,s}$ = pressure drop across an orifice plate in suction
p_s = suction pressure (absolute)

The second coordinate system is *pressure ratio* (R_c) vs. *reduced flow rate* in suction (q_s). This combination has been in use for many years and remains a basis for many surge control systems. This has the advantage of requiring only pressure and flow measurements. Often, however, using reduced head, h_r, results in a more invariant surge line.

From the (q_s^2,R_c) coordinate system comes the common surge control system based on Δp_c and $\Delta p_{o,s}$. This is constructed by assuming a surge control line that satisfies

$$R_c - 1 = C\frac{\Delta p_{o,s}}{p_s} \tag{16.6}$$

The derivation is

$$\begin{aligned} R_c - 1 &= C\frac{\Delta p_{o,s}}{p_s} \\ \frac{p_d - p_s}{p_s} &= C\frac{\Delta p_{o,s}}{p_s} \\ p_d - p_s &= C\Delta p_{o,s} \\ \Delta p_c &= C\Delta p_{o,s} \end{aligned} \tag{16.7}$$

Note that the gas composition is not required to calculate any of these coordinates. In application, reduced flow rate is calculated using the differential pressure, $\Delta p_{o,s}$, from the flow measurement device divided by suction pressure, p_s so temperature, gas molecular weight, and compressibility are not required.

It should be pointed out that these coordinates are not invariant to variations in isentropic exponent k. However, since k does not vary considerably in most applications, this does not present a problem. Therefore, although the term *invariant* is used, it should be remembered that these coordinates are *nearly invariant.*

Using either of these coordinates for a compressor without inlet guide vanes, the surge limit line is represented by a single curve that is stationary with changing inlet conditions. The rotational speed is not required to determine the relative distance between the operating point and the surge control line.

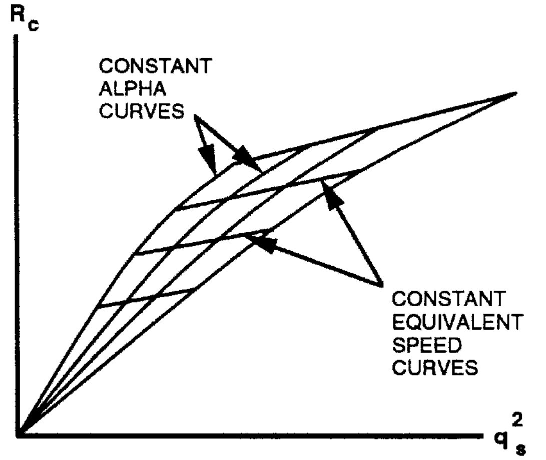

FIGURE 16.2 Compressor performance map for a compressor with variable inlet guide vanes. *(Compressor Controls Corporation, Des Moines, Iowa)*

For compressors with inlet guide vanes, the surge limit is represented by a family of curves that do not depend on suction conditions (Fig. 16.2). In this case, along with the two basic coordinates (reduced polytropic head *or* pressure ratio, and reduced flow rate), another coordinate may be required. This additional coordinate could be either guide vane position, α, or *equivalent speed,* N_e, defined as

$$N_e \equiv \frac{N\sqrt{\mathrm{MW}}}{\sqrt{Z_s R_u T_s}} \tag{16.8}$$

where N = rotational speed
MW = molecular weight
Z_s = suction compressibility factor
R_u = universal gas constant
T_s = suction temperature (absolute)

Note that the gas composition must be known to calculate equivalent speed. This is due to the appearance of molecular weight in the definition.

Although compressor maps are rarely presented in these invariant coordinates, these are the preferred systems for control. This is due, not only to their invariance, but also to the fact that gas composition is not required for their calculation (with the exception of equivalent speed).

Some variations and simplifications of the invariant coordinates are possible. Some other signals that are possible to use to produce invariant coordinates are the differential pressure across the compressor, Δp_c, and the flow measurement in discharge represented by the differential pressure, $\Delta p_{o,d}$.

16.4 PERFORMANCE CONTROL

One of the basic objectives of a compressor control system is to regulate the output of the compressor to meet the varying needs of the overall process. That objective is accomplished by

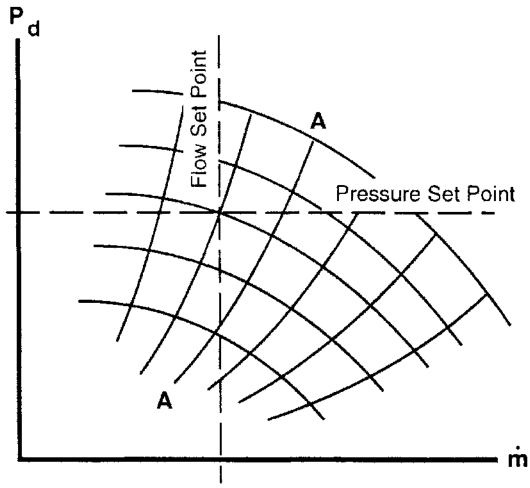

FIGURE 16.3 Compressor map showing typical performance control set points. *(Compressor Controls Corporation, Des Moines, Iowa)*

manipulating a *performance control element.* This control element might be a suction or discharge control valve, guide vane positioner, or rotational speed governor. It would serve to maintain a process pressure or flow rate at a *set point* value.

When illustrating performance control, it is helpful to redefine the compressor map coordinate system. For example, we might plot discharge header pressure against mass flow through the compressor (see Fig. 16.3). The set point will then appear as either a horizontal or a vertical line, depending on whether we want to maintain constant pressure or constant mass flow, respectively.

To achieve this coordinate system transformation, we must define the control element to be part of the compressor system rather than part of the network. Thus, repositioning a control valve moves the performance curve rather than the network resistance curve.

Defining the coordinate system in this way implies that any independent process variables (such as fluid composition) have constant values, so that each point in the H_p vs. Q_s coordinate system corresponds to a unique point in the new coordinate system. Changes in any of these variables would change the shape and position of the performance and resistance curves.

Within this transformed coordinate system, the discharge pressure and flow rate are determined by the intersection of the performance and resistance curves, corresponding to the current control element position and network resistance. The control system must manipulate the control element in response to changes in network resistance and gas properties, so that the operating point is kept as close to the set point line as possible.

For example, assume that the compressor of Fig. 16.4 is operating at point 1, which is on the set point line. If the network resistance decreases, the operating point will move along the performance curve to point 1′, where it intersects the new resistance curve. To restore set point operation, the control system must reposition the control element to select the performance curve that intersects the new resistance curve at the new operating point 1″.

Consider a hypothetical situation in which all of the assumptions made in transforming the coordinate system were valid, the process and control system were free of dynamic effects, and the equations of the performance and resistance curves were known. Then the control system might (at least theoretically) be able to deduce changes in network resistance from

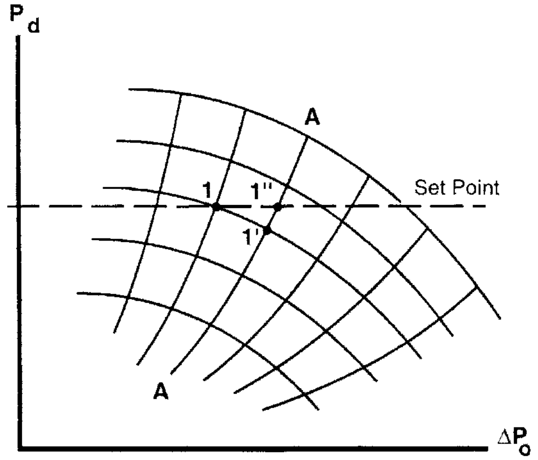

FIGURE 16.4 Compressor map illustrating typical performance control response. *(Compressor Controls Corporation, Des Moines, Iowa)*

the position of the control element and the value of the controlled variable. At that point the controller could calculate the exact position required by the system to satisfy the control objectives.

In general, none of these conditions can be met. Performance control algorithms must therefore be able to adapt to changes in process parameters that are not (or cannot) be measured. They must be able to overcome dynamic effects and time delays while dealing with the uncertainties in our knowledge of compressor performance and network resistance characteristics. They have to circumvent the difficulties associated with solving even approximate models of these imperfectly understood relationships in real time (usually by *not* solving them at all).

So our hypothetical situation doesn't exist in the real world. As a solution to these problems, we must construct a control system in another way. Very simply, the alternative of choice is a control scheme that always acts in a fashion that reduces the error between the operating point and the set point. This is explained in the following section.

16.4.1 PI and PID Control Algorithms

The commonly used proportional–integral (PI) and proportional–integral–derivative (PID) control algorithms provide the adaptable method needed to control imperfectly known compression systems. The basic premise of these algorithms is that the controller output should be a function of the difference (which is called the *error,* ϵ) between the value of the controlled variable (or *process variable,* PV) and its *set point* (SP).

$$\epsilon = \text{SP} - \text{PV} \tag{16.9}$$

The controller output is calculated as the sum of a constant steady-state value and several bias terms that are functions of the error:

$$\text{OUT}_{\text{PI}} = \text{SS} + P + I \tag{16.10}$$

$$\mathrm{OUT_{PID}} = \mathrm{SS} + P + I + D \tag{16.11}$$

where OUT = value of controller output signal
SS = steady-state component
$P \propto \epsilon$ (instantaneous error)
$I \propto \epsilon\, dt$ (integral error)
$D \propto d\epsilon/dt$ (derivative error)

To understand how these algorithms work, imagine an application in which the goal is to maintain a steady 100-psig discharge pressure. Assume the compressor is initially operating at steady state with its suction throttle valve open 75%. Because the compressor is in steady state and the valve is not at a limit, the error must be zero. Therefore, the instantaneous and derivative errors are zero, and the integral error is constant. Thus, the controller output will not change until the compression system is perturbed or the set point is changed.

Now assume that the downstream process system resistance decreases so that the control valve must be increased to 80% open to maintain the desired set point of 100 psig pressure. A 75% open valve would then produce a lower-than-desired pressure. Because the controller is unable to calculate the required new valve setting directly, an intelligent trial-and-error approach must be taken: Simply open the valve a specified amount, monitor the result, and proceed as required from there. This type of approach is known as *closed-loop control*.

The relationship between valve opening and discharge pressure is unknown. However, a good starting point is to assume that the additional valve opening should be approximately proportional to the pressure change: A large pressure variation is countered by a large change in the valve position. This assumption leads to the proportional term of the PID algorithm.

Returning to the example above, note what happens if the control were proportional *only* (the integral and derivative terms both set to zero). With the suction throttle valve at 75% open, the discharge pressure would decrease, causing the error (and thus the output) to rise. Resetting the output to 80% causes the pressure to increase, reducing the error (and thus the output). Note that if the pressure error were to go to zero, the output would go to SS since the control system is proportional only. Therefore, there is an intermediate level at which the error and the output (both nonzero) stabilize, leaving the pressure below its set point.

For example, assume that a valve opening of 78% would yield a steady 98-psig discharge pressure. If a 2-psig error produced a 3% proportional response (starting at a valve position of 75%), the output of the controller at 98 psig would be 78%—the exact valve setting needed to maintain that pressure. The discharge pressure would thus stabilize at 98 (instead of the desired 100) psig. The resulting 2-psig disparity is known as a *proportional offset.*

Proportional offsets are eliminated by adding an integral term to the control algorithm. Because this integral will accumulate whenever the error is nonzero, the control action cannot stabilize unless the controlled variable is at its set point. At steady state, then, the proportional part is zero (because the error is zero), but the integral term equals the value required to keep the error at zero.

Therefore, the steady-state controller output is actually the sum of the steady-state and integral error terms. As a result, the steady-state term loses its special significance and is usually merged into the integral term.

In the PID algorithm only, another term is added that is proportional to the first time derivative of the error. The basic premise behind including this term is that the magnitude of the control response should be modulated according to how fast the error is changing.

For example, a fairly large response would be appropriate if the pressure was too low and still falling. In contrast, a smaller response would be warranted if the pressure was too low but rising.

The practical effect of including the derivative term is that it often allows the control response to be accelerated without increasing the risk of instability. However, derivative control will also make the system more sensitive to signal noise. Thus, the simpler PI algorithm is sometimes more appropriate than full PID control.

16.4.2 Stability Considerations

Improper tuning can render either the PI or PID algorithm (or even simple proportional control) unstable (see Fig. 16.5). To gain a qualitative appreciation of how this can occur, consider the PI response to a step change in set point for a process initially operating at steady state.

Initially, the integral of the error is constant. So the output changes proportionately to the magnitude of the step change, countering its effect and thus decreasing the error. With purely proportional control, the system will tend toward a new steady state characterized by a proportional offset of the controlled variable relative to its set point.

However, because of the natural inertia of the system, changes in the process will lag behind the control signal. Because that signal will reach its new value before the controlled variable does, the control action will overshoot. As a result, the controlled variable will also overshoot, causing the control action to reverse. The overall process will therefore oscillate about the new steady-state conditions. Proper tuning of the control response will damp out these oscillations.

The magnitude of the overshoot will depend on the value of the control system's proportionality constant (which is known as the *proportional gain*). Increasing this gain will accelerate the control response, reduce the proportional offset, and increase the overshoot. If the proportional gain is set too high, each successive overshoot will be larger than its predecessor, resulting in an unstable system.

Adding an integral control action has the effect of increasing the system inertia and thus exacerbates the risk of instability. The reason is fairly simple—both the error and its integral will lag behind changes in the controller output. The proportional and integral gains must both be compromised to maintain system stability. As mentioned earlier, however, the integral term allows the control system to reduce the error to zero.

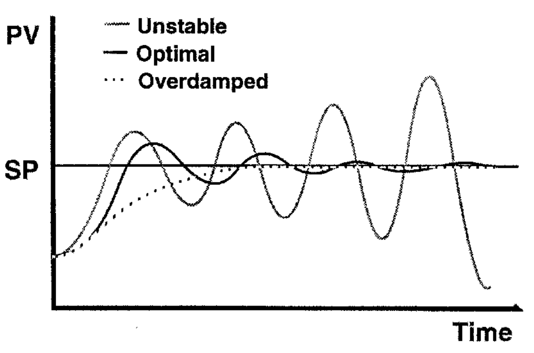

FIGURE 16.5 System response to improper tuning of the PID algorithm. *(Compressor Controls Corporation, Des Moines, Iowa)*

On the other hand, adding a derivative term can reduce the risk of instability. This is because the derivative of the error is a measure of how fast the system is responding. If the system becomes unstable, the derivative action will tend to counter the oscillatory action.

16.4.3 Integral or Reset Windup

A PID controller may encounter situations in which changing the output signal cannot reduce the error to zero. For example, suppose that our control objective is to maintain a constant fluid level in a storage tank by manipulating an inlet valve to balance the inlet and outlet flows. Even with the valve fully open, the feed rate might prove insufficient to maintain the desired level. The control system cannot manipulate the controlled variable to eliminate this problem.

In such a situation, the integral error would continue growing indefinitely, eventually reaching a saturation value. Then if the flow rates change so that the level becomes too high, the PID loop must integrate the error (which is now opposite in sign) for some time to reduce its value from the saturation level to a magnitude that allows the valve position to come off 100% open. So the control system is unable to respond rapidly enough to sudden decreases in outflow, possibly causing the tank to overflow.

The breakdown of control under such circumstances is referred to as *integral* or *reset windup.* Compressor control systems that employ PI or PID algorithms must be able to recognize and respond to this condition by disabling the integral portion of the algorithm, setting it to a constant (generally nonzero) value.

16.5 PERFORMANCE LIMITATIONS

As shown previously, a compressor increases the pressure of the process gas by first accelerating the fluid and then converting the resulting kinetic energy into an increased pressure head. Thus, the maximum polytropic head that can be developed is limited by the tip speed of the impeller blades. Polytropic head is observed to be a decreasing function of volumetric flow.

At any given rotational speed and inlet conditions, the performance of the compressor is limited not only by this maximum polytropic head but also by a maximum flow rate. These limitations are illustrated on the typical performance curve shown in Fig. 16.6.

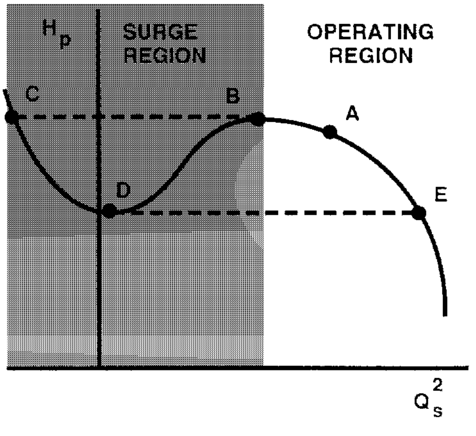

FIGURE 16.6 Performance curve illustrating typical performance control response. *(Compressor Controls Corporation, Des Moines, Iowa)*

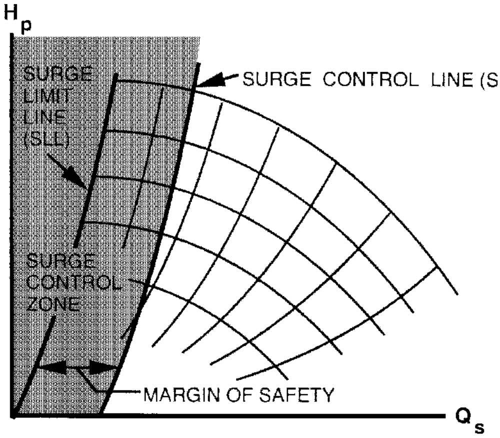

FIGURE 16.7 Compressor map showing the surge limit line, surge control line, and surge control zone. *(Compressor Controls Corporation, Des Moines, Iowa)*

16.5.1 Surge Limit

Consider the scenario of a constant-speed compressor with fixed suction conditions. This compressor is initially at steady state, so the operating point is stationary. If the network resistance increases, the operating point will move along the performance curve to the left. Eventually, a point of minimum stable flow and maximum polytropic head is encountered. Operating the compressor to the left of this point can induce a potentially destructive phenomenon known as *surge*. This point is known as the *surge limit point.* The locus of all such points defines a curve known as the *surge limit line* (SLL), shown in Fig. 16.7.

The onset of surge can be visualized by considering a simple system consisting of a compressor and discharge valve. If this system were operating in steady state, there would be no accumulation of the process gas (over time) in the plenum between the two elements. In Fig. 16.6, this point is represented by *A*.

If the valve were then closed slightly, the flow rate through it would drop. Process gas would accumulate in the plenum, causing a pressure increase. This would, in turn, cause a decrease in the flow through the compressor and an increase in the flow through the valve, until they were once again equal. The new steady-state flow rate would be lower than it was originally, and the new plenum pressure would be higher.

If the valve continued to close in small steps, the compressor would approach its surge limit (point *B* in Fig. 16.6). This approach would be noticeable when a relatively large drop in flow produced a small increase in the final plenum pressure. In other words, the slope of the compressor characteristics tends to reduce in magnitude near the surge limit point.

At the surge limit point, closing the valve further would cause the flow to decrease. As shown in Fig. 16.6, the compressor could not achieve the pressure required to maintain the flow less than that of the surge point. The result is that the flow through the compressor would drop precipitously, typically reversing in approximately 20 to 80 ms (see Fig. 16.8). This is shown as the dashed line *B–C* in Fig. 16.6. The operating point cannot reside on the portion of the curve between *B* and *D.* Note that the flow drops off before the pressure. This can be seen in the top plot in Fig. 16.8.

At point *C*, the flow would be negative, but energy is still being added to the fluid. The plenum would begin to depressurize due to the net efflux of fluid. The trajectory of the

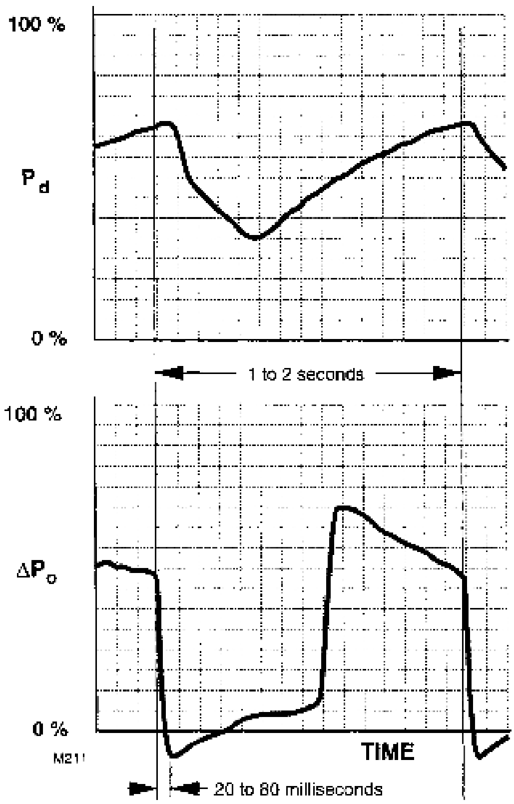

FIGURE 16.8 Pressure and flow variations during typical surge cycles. *(Compressor Controls Corporation, Des Moines, Iowa)*

operating point would be from *C* to *D* in Fig. 16.6. At point *D*, the compressor, once again, reaches the region of the curve on which it cannot reside. At this point the flow quickly jumps to point *E*, where the compressor is in the "safe region."

With the discharge valve at the position that caused the surge in the first place, if no other action is taken, the compressor will travel from *E* to *A* and then repeat the cycle. The oscillatory pattern will continue until some action occurs to stop it (e.g., opening the discharge valve) or the compressor fails.

During surge, the severe oscillations of flow and pressure create heavy thrust bearing and impeller loads, vibration and rising gas temperatures. If more than a very few surge cycles are experienced, process upsets and severe compressor damage are likely to result. The past decade has brought on a better understanding of the various modes of instability associated with compressors.

16.5.2 Stonewall

Again, consider a constant-speed compressor with fixed suction conditions. As network resistance decreases, the operating point will move along the performance curve to the right.

Eventually, a point of maximum flow and minimum polytropic head is encountered, beyond which further decreases in network resistance will not increase the flow rate. This is known as the *choke point,* or *stonewall.*

The physical phenomenon is that the gas velocity has increased to the local acoustic velocity (therefore, Mach 1) at some point in the compressor. When choke occurs, the flow rate cannot increase unless conditions change in the choked region.

Stonewall is not particularly damaging to single-stage centrifugal compressors but can cause serious damage to the rotors and blades of multistage centrifugal and axial compressors. In such situations, a suitably designed antichoke controller can be used to manipulate an antichoke control valve, thus maintaining sufficient system resistance to prevent choke.

16.6 PREVENTING SURGE

Surge occurs when the network resistance becomes too high for the compressor to overcome. The obvious way to prevent surge is to decrease network resistance whenever the operating point moves too close to the surge limit line. This is accomplished by opening an antisurge valve to recycle or discharge a portion of the total flow.

The chief drawback to this approach is the efficiency penalty that it entails—the energy that was used to compress the recycled gas goes to waste. Thus, the control system should be tailored to open the antisurge valve only when—and only as far as is—necessary. On the other hand, if we do not provide adequate protection against surge, we risk prohibitive repair and downtime costs.

Therefore, accurate and dependable methods of determining the surge limit are required. Antisurge control entails measuring the distance between this surge limit and the operating point and then maintaining an adequate margin of safety without sacrificing efficiency or stability.

The solution is to maintain the operating point on or to the right of a line known as the *surge control line* (SCL; see Fig. 16.7). The distance between the surge control and surge limit lines (the margin of safety) should be just enough to allow the chosen control algorithms to counteract an impending surge.

Whenever the operating point moves into the *surge control zone* (i.e., to the left of the SCL), the antisurge valve must be opened fast enough to keep the operating point from reaching the surge limit line and far enough to return it to the surge control line. On the other hand, when the operating point moves to the right of the SCL, the antisurge valve should be closed as far as possible without moving the operating point into the surge control zone.

16.6.1 Antisurge Control Variables

Like H_p and Q_s, the distance between the operating point and the surge limit line cannot be measured directly. Nor is there a standard definition relating it to parameters that can be measured. Thus, antisurge protection algorithms can be based on any function of measurable process variables that satisfies the following criteria:

- It should vary monotonically as the operating point approaches the surge limit so that the required control action is never ambiguous (if not, there must be a means of resolving any ambiguities).

- It must be invariant to any aspect of the process that might change so that the compressor is adequately protected in all possible situations.
- It must be easily calculated from process variables that can be accurately measured or assumed constant.
- It should be most sensitive to changes that occur when the operating point is near the surge limit.

The obvious possibilities include combinations of the coordinates presented in Section 16.3. Other, less obvious candidates include functions of the compressor drive power or rotational speed, all of which meet these criteria for at least some applications.

The existence of guide vanes may complicate antisurge control. In general, the guide vane angle, α, must be included as part of the variable used for determining the distance between the operating point and the surge control line.

One variable that has advantages of control on flow as well as pressure is a ratio of a function of the ordinate to the abscissa. From Section 16.3, choosing the (q_s^2, R_c) coordinate system, the antisurge control variable, S_s, would be calculated as

$$S_s = \frac{f(R_c)}{q_s^2} \tag{16.12}$$

where $f(\cdot) = q_s^2|\text{surge}$
R_c = compressor pressure ratio, p_d/p_s
$q_s^2 = \Delta p_o/p_s$

The surge limit line is therefore a line of $S_s = 1$. As shown in Fig. 16.9, lines of constant S_s comprise a family of curves that emanate from the origin. They are similar in shape to the surge limit line and diverge from the surge limit line. Using this method, comparing the value of S_s to unity gives a relative distance to surge.

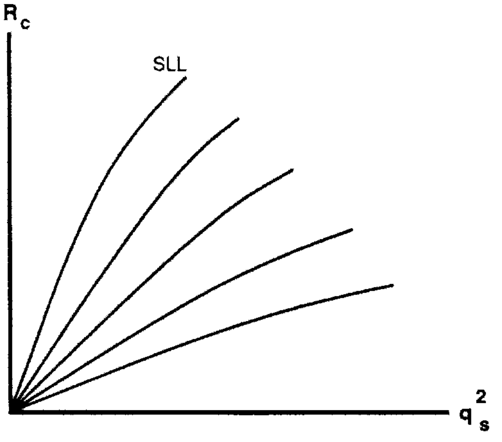

FIGURE 16.9 Compressor performance map showing lines of constant surge control variables S_s. *(Compressor Controls Corporation, Des Moines, Iowa)*

16.6.2 Antisurge Control Algorithms

In performance control, the goal is to minimize deviations of the controlled variable from its set point; variations to either side are of approximately equal concern. In contrast, the goal of an antisurge controller is to maintain the controlled variable to one side of an absolute limit; deviations to the other side must be prevented at any cost.

Thus, good antisurge control uses a combination of the following types of control responses:

- Opening and closing the control valve to maintain the operating point on or to the right of the surge control line without allowing deviations to the left of the surge limit line.
- Moving the surge control line (relative to the surge limit line) to adapt the margin of safety to changing process conditions.

The first of these points may be accomplished by simple closed-loop (PI or PID) control if the safety margin is great enough. To reduce the safety margin (thereby increasing the operating range and efficiency of the compressor), rapid increase of the recycle valve set point is expedient.

Another aspect of this issue was noted earlier. The safety margin should be increased if surge occurs under a large disturbance. This reduces the chance that surge will recur under a similarly large disturbance.

Another aspect of this issue is to increase the safety margin *dynamically* when surge is threatening. Basing this adjustment on the derivative of the approach (from the right of the surge control line *only*; see Ref. 1) results in a control scheme that does not compromise stability. It also provides for earlier opening of the recycle valve under fast disturbances.

16.6.3 Controlling Limiting Variables

The third major responsibility of an integrated compressor control system is to counteract undesirable changes in any process-limiting variables. The limitation involved might be required to protect the compressor or other process equipment (e.g., preventing excessive motor current, bearing temperatures, or excessive pressures). Or it may be necessary to protect the process gas from conditions, such as an excessive discharge temperature, that could chemically degrade or otherwise damage its quality.

In an integrated control system, we are concerned with two categories of process-limiting variables: those that must be controlled by opening the antisurge valve and those that must be controlled by manipulating the performance control element.

Limiting variables that fall into the first category can be controlled by the antisurge controller, provided that it has such a capability. Otherwise, an additional controller must be used, along with a switching device that can dynamically assign control of the antisurge valve to the appropriate controller.

Similarly, if there are limiting variables that can only be controlled by manipulating the performance control element, it is necessary to use either a performance controller with multivariable capabilities or multiple controllers along with an appropriate switching circuit.

Regardless of which approach(es) are chosen, it is necessary to provide protection against integral windup in any control loop using a PI or PID algorithm. When multivariable controllers are used, the operator may or may not be allowed to override automatic control of any or all of those variables.

16.7 LOOP DECOUPLING

The action of the antisurge control system can upset the performance control operation, and vice versa. In networks of compressors, the antisurge control actions between the various cases may also require decoupling. The potentially conflicting effects of interacting control loops can be counteracted by implementing a loop decoupling algorithm.

For example, if the performance controller (PIC) in Fig. 16.10 needs to reduce the downstream flow rate, the rate at which it closes its control valve may need to be compromised to avoid destabilizing the system. The optimum response will depend on the proximity of the compressor's operating point to the surge line.

If the compressor was operating on its surge control line, reducing the flow rate would move the operating point into the surge control zone. The antisurge controller (UIC) would respond by opening the recycle valve, which would have the side effect of further reducing the downstream flow rate. The performance controller would then need to increase the flow rate, which would (in turn) cause the antisurge controller to reduce the opening of its valve.

These interactions would render both loops more oscillatory and therefore less stable. Stability could be restored only by tuning the controllers less aggressively, thus making them less effective when operating well away from the surge limit. Unless some provision was made for coordinating the two loops, neither could be optimally tuned for all situations.

This problem can be overcome by having the controllers monitor and compensate for changes in each other's outputs. This feedforward control action would allow the performance controller to moderate its actions when the antisurge valve was opening, and vice versa. The loop interactions would thus be decoupled: The actions taken by each controller would then be correct regardless of what the other was doing.

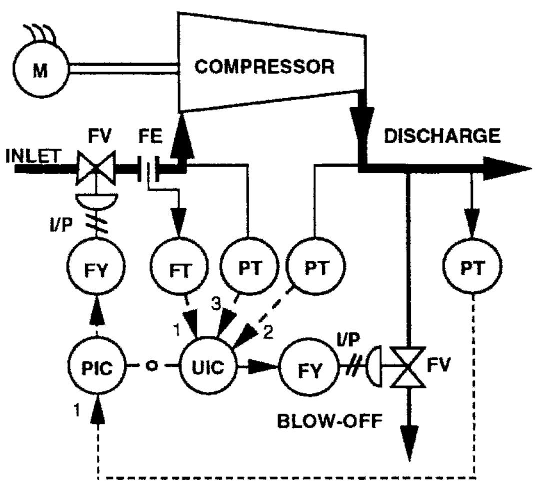

FIGURE 16.10 Interacting performance and antisurge control loops. *(Compressor Controls Corporation, Des Moines, Iowa)*

16.8 CONCLUSIONS

Compressor control is not only critical (from an economic standpoint), it is a challenging controls problem. Compression systems are typically dynamic, with lags and delays that make it difficult—most of the time impossible—to control them with a simple PID approach. The antisurge variable used for control should be selected carefully to provide accurate prediction regardless of the inlet conditions of the compressor.

It is important to integrate the various control tasks surrounding the compressor: antisurge, performance, limiting, and so on. When the compression system involves multiple compressors, a method of balancing the load between compressors may be required. This method should combine maintaining the performance variable at its set point while not compromising surge control.

REFERENCE

1. Marketing Literature, Compressor Controls Corporation, Des Moines, Iowa, 1992 and 2004.