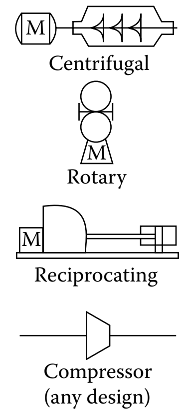
8.15 Compressor Control and Optimization



Flow sheet symbol

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INTRODUCTION

The transportation of vapors and gases is an important portion of the total operating cost of processing plants. The optimization of this unit operation can substantially lower the total operating cost of the plant. The goal of this section is to describe the strategies available to increase the safety and energy efficiency in these systems. Prior to the discussion of state-of-the-art advanced controls, the basic equipment and the conventional control strategies will be described.

Three types of compressors will be discussed: rotary, reciprocating, and centrifugal compressors. The emphasis will be on the centrifugal units, whose load, surge, and override controls will be described in some detail. Multiple compressor installations will also be described.

Compressors are gas transportation machines that perform the function of increasing the gas pressure by confinement or by kinetic energy conversion. Methods of capacity control for the principal types of compressors are listed in Table 8.15a.

TABLE 8.15a
Capacity Control Methods Used on the Different Compressor Types

Compressor Type	*Capacity Control Method*
Centrifugal	Suction throttling Discharge throttling Variable inlet guide vanes Speed control
Rotary	Bypassing Speed control
Reciprocating	On/off control Constant-speed unloading Speed control Speed control and unloading

The method of control to be used is a function of process requirements, type of the driver, and cost considerations.

Because the driver constitutes half of the cost of the compressor installation, careful selection must be made in order to ensure trouble-free performance. Variable speed control can be accomplished by the use of steam turbines, gas turbines, or gasoline or diesel engines. Electric motors are well suited for both constant- and variable-speed applications.

The flow and discharge pressure ranges of the various compressor designs are shown in Figure 8.15b.

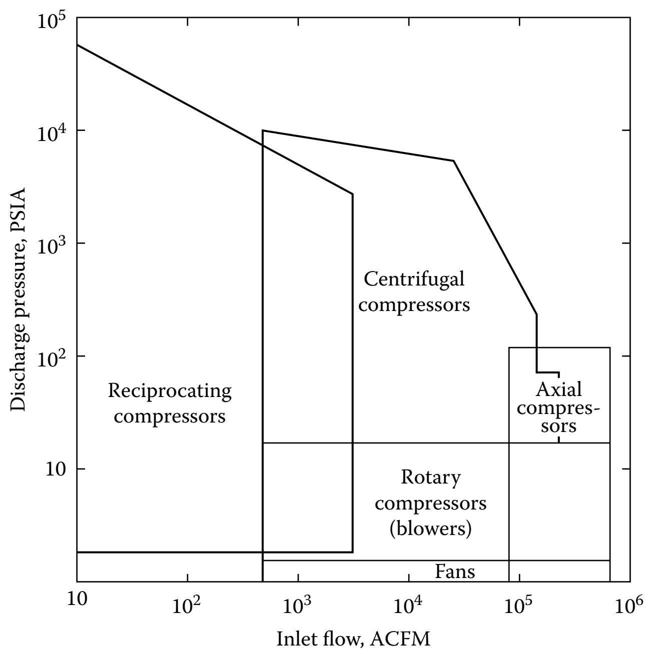
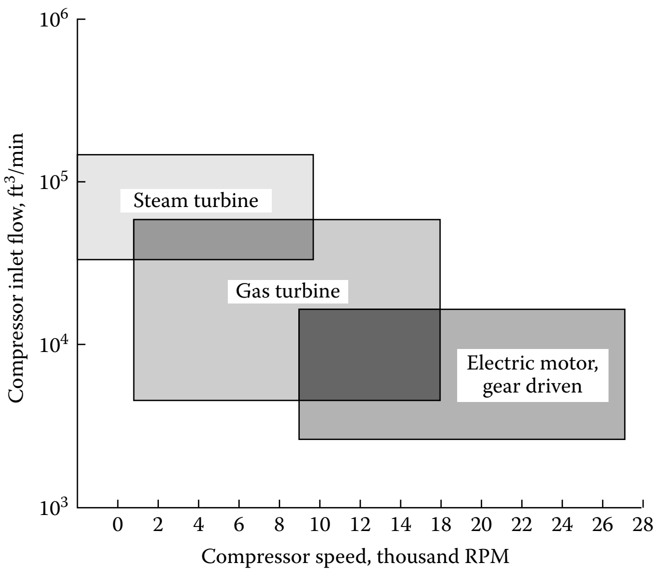


FIG. 8.15b
The range of flow capacities and discharge pressures that the different compressor designs can handle.[1]



Speed Governor Classification

Governor class	*Regulation*	*Variation*
A	10%	¾%
B	6%	½%
C	4%	¼%
D	½%	¼%

FIG. 8.15c
The speed and flow capacity ranges of standard compressor drives (top) and the classification of speed governors (bottom).

The operating ranges of the most popular drives are shown in Figure 8.15c. Also given in the lower part of the figure is the definition of some two-speed governors. Regulation error is the percentage of the difference in speed between the speed at zero power output and at the rated output. Variation error is the percentage variation that can be expected around the speed set point.

CENTRIFUGAL COMPRESSORS

The Compression Process

The centrifugal compressor is a machine that converts the momentum of gas into a pressure head.

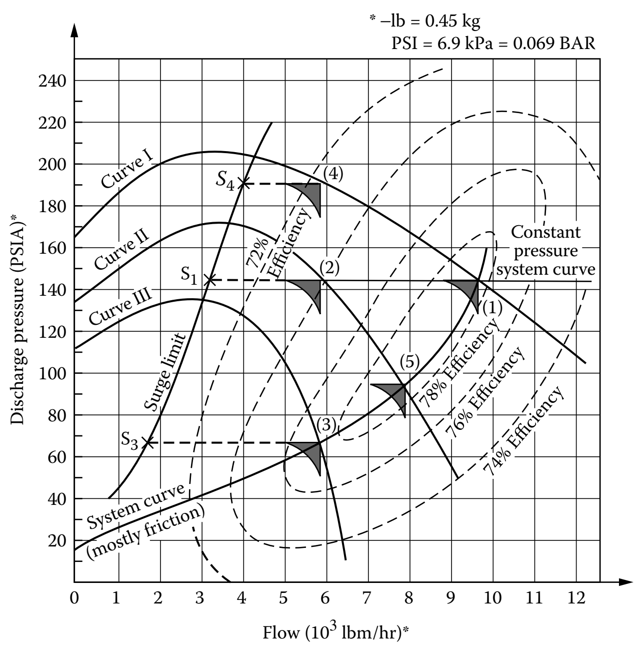
$$H = \frac{\tau\omega}{w} = \frac{n}{n-1} ZRT_I, \qquad [(P_D/P_I)^{\frac{n}{n-1}} - 1] \qquad \textbf{8.15(1)}$$

Equation 8.15(1) is the basis for plotting the compressor curves and for understanding the operation of capacity controls. The nomenclature for the symbols in the equation is as follows:

h = differential head (ft or m)
H = polytrophic compressor head (ft or m)
$K_{1,2,3}$ = flow constant
m = mass flow (lb/hr or kg/hr) = $c\sqrt{h\zeta}$
n = polytrophic coefficient
P_D = discharge pressure (psia or Pa)
P_I = inlet pressure (psia or Pa)
Q = volume flow rate (ACFH or m^3/hr) = $c\sqrt{h\zeta}$
R = gas constant
T_1 = inlet temperature
u = rotor tip speed (ft/s or m/s)
W = weight flow (lbm/hr or kg/hr)
Z = gas compressibility factor
τ = motor torque (ft lb$_f$ or J)
ψ = head coefficient
ω = angular velocity (radians/hr)
ζ = density of gas (lb/ft^3 or kg/m^3)

Characteristic Curves

In Equation 8.15(1), the pressure ratio (P_D/P_I) varies inversely with mass flow (W). For a compressor running at constant speed (ω), constant inlet temperature (T_1), constant molecular weight (implicit in R), and constant n, τ, and Z, the discharge pressure may be plotted against weight flow as in Figure 8.15d



*For SI units see Section A.1

FIG. 8.15d
The operating point is located where the centrifugal compressor curves cross the system curve of the process. The system curve can be a constant pressure one (horizontal line), a mostly friction one, or any other.

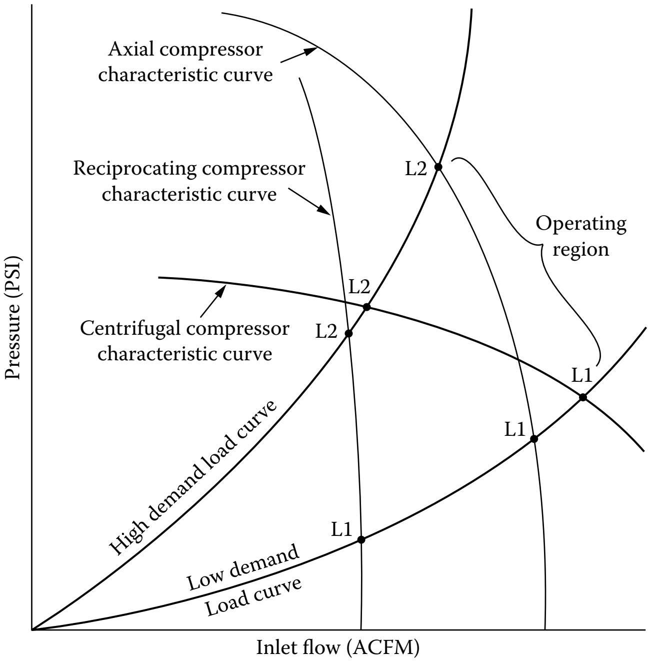


FIG. 8.15e
The characteristic curve of the axial compressor is steep, while that of the centrifugal compressor is flat. (Adapted from Reference 1.)

(curve I). The design point (1) is located in the maximum efficiency range at design flow and pressure.

Positive-displacement compressors pressurize gases through confinement. Dynamic compressors pressurize them by acceleration. The axial compressor moves the gas parallel to the shaft. In the case of the centrifugal compressor, the gas receives a radial thrust toward the wall of the casing where it is discharged.

The axial compressor is better suited for constant flow applications, whereas the centrifugal design is more applicable for constant pressure applications. This is because the characteristic curve of the axial design is steep, and that of the centrifugal design is flat (Figure 8.15e). The characteristic curve of a compressor plots its discharge pressure as a function of flow, and the load curve relates the system pressure to the system flow. The operating points (*L*1 or *L*2 in Figure 8.15e) are the intersections of these curves. The normal operating region falls between the low and the high demand load curves in Figure 8.15e.

Axial compressors are more efficient; centrifugal ones are better suited for dirty or corrosive services.

Compressor Throttling

Compressor loading can be reduced by throttling a discharge or a suction valve, by modulating a prerotation vane, or by reducing the speed. As is shown in Figure 8.15f, discharge throttling is the least energy efficient and speed modulation is the most energy efficient method of turndown. Suction throttling is a little more efficient and gives a little better turndown than discharge throttling, but it is still a means of wasting that transporting energy that should not have been introduced in the first place.

Guide vane positioning, which provides prerotation or counter-rotation to the gas, is not as efficient as speed modulation, but it does provide the greatest turndown. As is shown in Figure 8.15f, speed control is the most efficient, as small

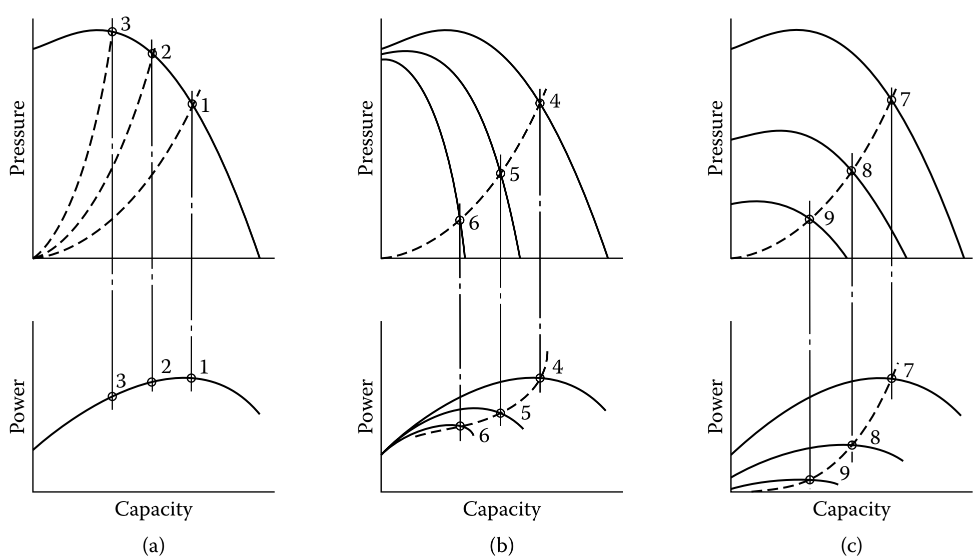


FIG. 8.15f
The efficiencies of discharge throttling (left), suction throttling (center), and variable speed control (right).

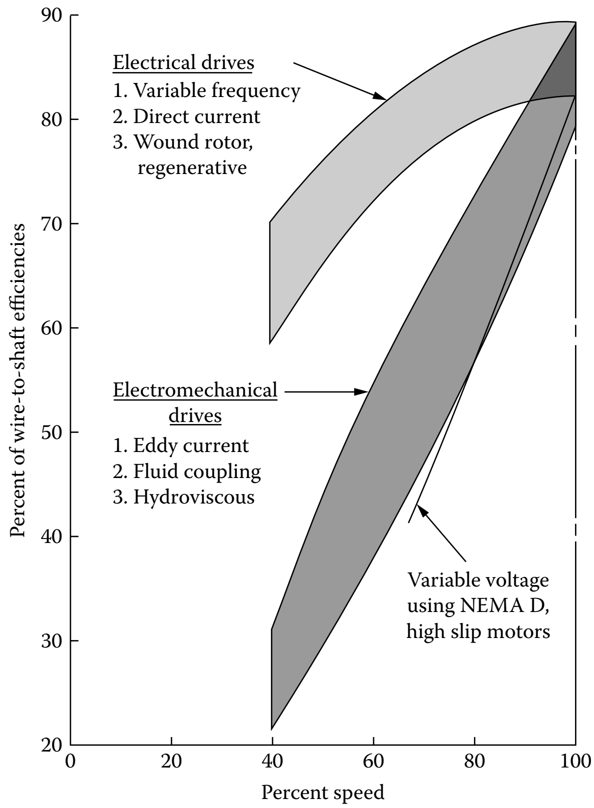


FIG. 8.15g
Wire-to-shaft efficiencies of electrical drives (top) are higher than those of the electromechanical drives.[2]

speed reductions result in large power savings because of the cubic relationship between speed and power.

If the discharge pressure is constant, flow tends to vary linearly with speed. If the discharge head is allowed to vary, it will change with the square of flow and, therefore, with the square of speed as well. This square relationship between speed and pressure tends to limit the speed range of compressors to the upper 30% of their range.[3]

Constant-speed steam turbine governors can be converted into variable-speed governors by revising the quick-opening characteristics of the steam valve into a linear one. The efficiency of variable-speed drives varies substantially with their design (Figure 8.15g). Electric governors tend to eliminate the dead bands that are present in mechanical designs. They also require less maintenance because of the elimination of mechanical parts. Electric governors also give better turndowns and are quicker and simpler to interface with surge or computer controls.

The error in following the load increases as the speed of process disturbances increase or as the speed control loop speed is reduced. Therefore, it is desirable to make the speed loop response as fast as possible. On turbines, this goal is served by the use of hydraulic actuators, and the motor response is usually increased by the use of tachometer feedback.

For the purposes of the control systems shown in this chapter, it will be assumed that the compressor throughput is controlled through speed modulation with tachometer feedback.

Suction Throttling One can control the capacity of a centrifugal compressor by throttling a control valve in the suction line, thereby altering the inlet pressure (P_I). From Equation 8.15(1) it can be seen that the discharge pressure will be altered for a given flow when P_I is changed, and a new compressor curve will be generated. This is illustrated in Figure 8.15d (curves II and III).

Consider first that the compressor is operating at its normal inlet pressure (following curve I) and is intersecting the "constant pressure system" curve at point (1) with a design flow of 9600 lbm/hr (4320 kg/hr) at a discharge pressure of 144 psia (1 MPa) and 78% efficiency. If it is desired to change the flow to 5900 lbm/hr (2655 kg/hr) while maintaining the same discharge pressure, it would be necessary to shift the compressor from curve I to curve II.

The new intersection with the "constant pressure system" curve is at the new operating point (2), at 74% efficiency. In order to shift from curve I to curve II, one must change the discharge pressure of 190 psia (1.3 MPa) at the 5900 lbm/hr (2655 kg/hr) flow on curve I to 144 psia (1 MPa) on curve II. If the pressure ratio is 10 (P_D/P_I), then it would be necessary to throttle the suction by only $\Delta P_I = 46/10 = 4.6$ psi (32 kPa) to achieve this shift.

It is also important to consider how close the operating point (2) is to the surge line. The surge line represents the low-flow limit for the compressor, below which its operation will become unstable as a result of momentary flow reversals. Methods of surge control will be discussed later in this section. At point (2) the flow is 5900 lbm/hr (2655 kg/hr), and at the surge limit (S_1) it is 3200 lbm/hr (1440 kg/hr). Thus, the compressor is operating at 5900/3200 = 184% of surge flow. This may be compared with curve I at point (1), where prior to suction throttling the machine was operating at 9600/3200 = 300% of surge flow.

The same method of suction throttling may be applied in a "mostly friction system" also shown in Figure 8.15d. In order to reduce the flow from 9600 lbm/hr (4320 kg/hr) to 5900 lbm/hr (2655 kg/hr), it is necessary to alter the compressor curve from curve II to III, so that the intersection with the "mostly friction system curve" is at the new operating point (3), at 77% efficiency.

In order to do this, one must change the discharge pressure from 190 psia (1.3 MPa—on curve I) to 68 psia (0.5 MPa—on curve III). Thus, $\Delta P_D = 190 - 68 = 122$ psi (0.8 MPa), and the amount of inlet pressure throttling for a machine with a compression ratio of 10 is $\Delta P_I = 122/10 = 12.2$ psi (84 kPa). The corresponding surge flow is at 1700 lbm/hr (765 kg/hr), which means that the compressor is operating at 5900/1700 = 347% of surge flow. Therefore,

TABLE 8.15h
Compressor Performance Parameters as a Function of Throttling Method

	Control Valve ΔP(PSI)	Compressor Efficiency	Operation Above Surge By
Suction throttling "constant pressure system"	4.6	74%	184%
Suction throttling "mostly friction system"	12.2	77%	347%
Discharge throttling "mostly friction system"	122	72%	148%

surge is less likely in a "mostly friction system" than in a "constant-pressure system" under suction throttling control.

Discharge Throttling A control valve on the discharge of the centrifugal compressor may also be used to control its capacity. In Figure 8.15d, if the flow is to be reduced from 9600 lbm/hr (4320 kg/hr) at point (1) to 5900 lbm/hr (2655 kg/hr), the compressor must follow curve I and therefore operate at point (4), at 190 psia (1.3 MPa) discharge pressure and 72% efficiency. However, the "mostly friction system" curve at this capacity requires only 68 psia (0.5 MPa) and the rest of the pressure is wasted through the discharge control valve. The surge flow (at S_4) is 4000 lbm/hr (1800 kg/hr), and the compressor is therefore operating at 5900/4000 = 148% of surge. Thus, surge is more likely to occur in a mostly friction system when discharge throttling is used than when suction throttling is used.

Table 8.15h compares the valve pressure drops, efficiencies, and surge margins of suction and discharge throttling.

Inlet Guide Vanes This method of control uses a set of adjustable guide vanes on the inlet to one or more of the compressor stages. By prerotation or counter-rotation of the gas stream relative to the impeller rotation, the stage is unloaded or loaded, thus lowering or raising the discharge head. The effect is similar to suction throttling as illustrated in Figure 8.15d (curves II and III), but less power is wasted because pressure is not throttled directly. Also, the control is two-directional, since it may be used to raise as well as to lower the band. It is more complex and expensive than throttling valves but may save 10 to 15% on power and is well suited for use on constant-speed machines in applications involving wide flow variations.

The guide vane effect on flow is more pronounced in constant discharge pressure systems. This can be seen in Figure 8.15d (curve II), where the intersection with the "constant pressure system" at point (2) represents a flow change from the normal design point (1) of 9600 − 5900 = 3700 lbm/hr (1665 kg/hr). The intersection with the "mostly friction system" at point (5) represents a flow change of only 9600 − 7800 = 1800 lbm/hr (810 kg/hr).

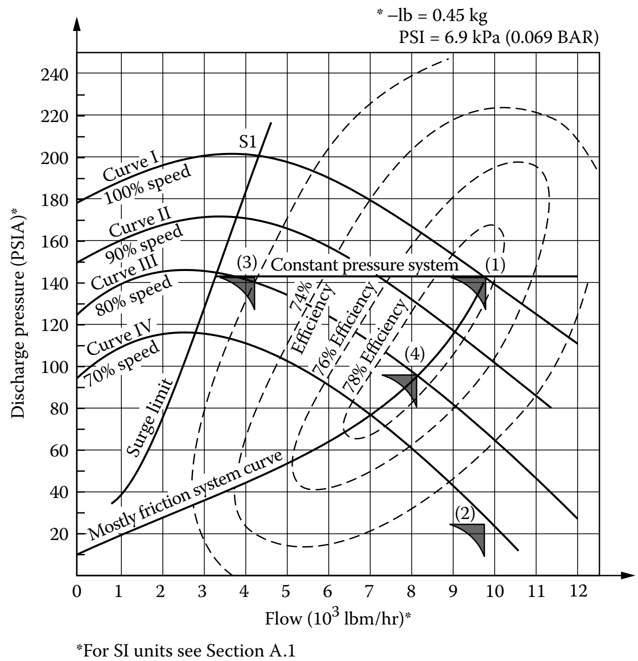


FIG. 8.15i
Control of centrifugal compressor capacity by speed variation.

Variable Speed The pressure ratio developed by a centrifugal compressor is related to the tip speed in the following manner:

$$\psi u^2/2g = \frac{ZRT_I}{(n-1/n)}[(P_D/P_I)^{(n-1/n)} - 1] \qquad \mathbf{8.15(2)}$$

From this relation the variation of discharge pressure with speed may be plotted for various percentages of design speed, as shown in Figure 8.15i. The obvious advantage of speed control from a process viewpoint is that both suction and discharge pressures can be specified independently of the flow.

The normal flow is shown at point (1) for 9700 lbm/hr (4365 kg/hr) at 142 psia (0.98 MPa). If the same flow is desired at a discharge pressure of 25 psia (173 kPa), the speed is reduced to 70% of design, shown at point (2). In order to achieve the same result through suction throttling with a pressure ratio of 10:1, the pressure drop across the valve would have to be (142 – 25)/10 = 11.7 psi (81 kPa), with the attendant waste of power, as a result of throttling. This is in contrast with a power saving accomplished with speed control, because power input is reduced as the square of the speed.

One disadvantage of speed control is apparent in constant-pressure systems, in which the change in capacity may be overlay sensitive to relatively small speed changes. This is shown at point (3), where a 20% speed change gives a flow change of (9600 − 4300)/9600 = 55%. The effect is less pronounced in a "mostly friction system," in which the flow change that results from a 20% speed change at point (4) is (9600 − 8100)/9600 = 16%.

Surge Control

The design of compressor control systems is not complete without consideration of surge control, because it affects the stability of the machine. Surging begins at the positively sloped section of the compressor curve. In Figure 8.15i this occurs at S_1 on the 100% speed curve at 4400 lbm/hr (1980 kg/hr). If the flow never drops below this limit, that will ensure safe operation for all speeds, but some power will be wasted at speeds below 100% because the surge limit decreases at reduced speed.

Even for a compressor running at a constant speed, the surge point changes as the thermodynamic properties vary at the inlet. This is shown in Figure 8.15j. Although inaccurate control of the surge point can put the compressor into deep surge, a conservatively set surge point results in useless recycling and wasted energy.

Various schemes to control surge are outlined in the following paragraphs. These include:

1. Compressor pressure rise ($\Delta P = P_D - P_I$) vs. differential across suction flow meter (h)
2. Pressure ratio (P_D/P_I) vs. actual volumetric flow (Q)
3. Break horsepower vs. mass flow (m)
4. Pressure ratio (P_D/P_I) vs. Mach number squared
5. Incipient surge
6. Surge spike detection

The Phenomenon of Surge In axial or centrifugal compressors, the phenomenon of momentary flow reversal is called surge. During surging, the compressor discharge pressure drops off and then is reestablished on a fast cycle. This cycling, or surging, can vary in intensity from an audible rattle to a violent shock. Intense surges are capable of causing complete destruction of compressor parts, such as blades and seals.

The characteristic curves of compressors are such that at each speed they reach a maximum discharge pressure as the flow drops (Figure 8.15k). A line connecting these points (A to F) is the surge line. If flow is further reduced, the pressure generated by the compressor drops below that which is already existing in the pipe, and momentary flow reversals occur. The frequency of these oscillations is between 0.5 and 10 Hz. The surge frequency of most compressor installations in the processing industries is slightly less than 1 Hz.[5] Surge

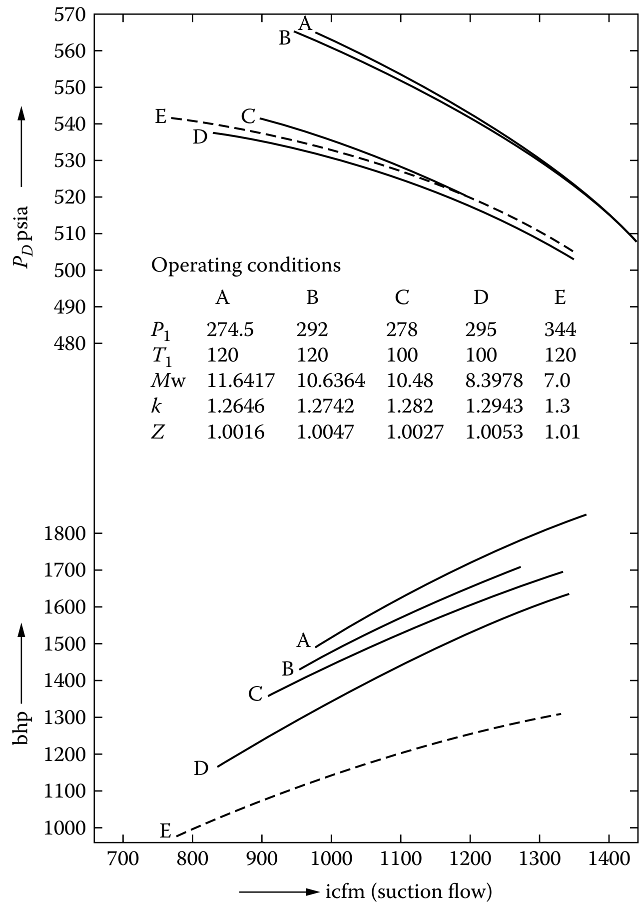


FIG. 8.15j
Discharge pressure generated and break horsepower used at various suction flows by a typical centrifugal compressor, under five different operating conditions.

is usually preceded by a stall condition, which is caused by localized flow oscillations around the rotor at frequencies of 50–100 Hz.

At the beginning of surge, the total flow drops off within 0.05 seconds, and then it starts cycling rapidly at a period of less than 2 sec.[1] This period is usually shorter than that of the flow control loop, which controls the capacity of the compressor. If the flow cycles occur faster than the control loop can respond to them, this cycling will pass through undetected as uncontrollable noise. Therefore, fast sensors and instruments are essential for this loop.

As is shown in Figure 8.15k, the surge line is a parabolic curve on a plot of pressure rise (discharge pressure minus suction pressure) vs. flow. This function shows as increasing nonlinearity as the compression ratio increases. If the surge line is plotted as $Pd - Pi$ vs. the square of flow (orifice differential = h), it becomes a straight line (Figure 8.15l) if the compression ratio is low (less than 4:1).

On a plot of ΔP vs. volumetric flow (Q), the following changes will reduce the safety margin between the operating

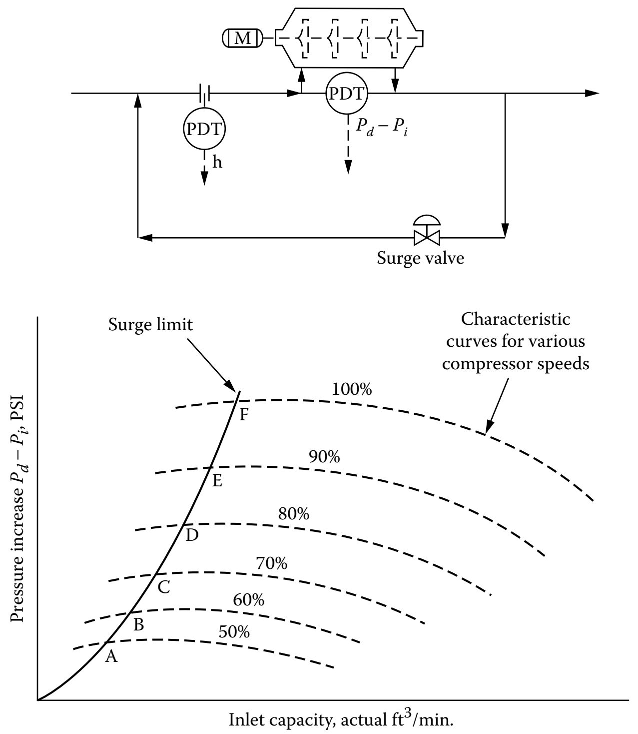


FIG. 8.15k
The location of the surge control valve (top) and the parabolic surge line of a speed-controlled centrifugal compressor (bottom).[4]

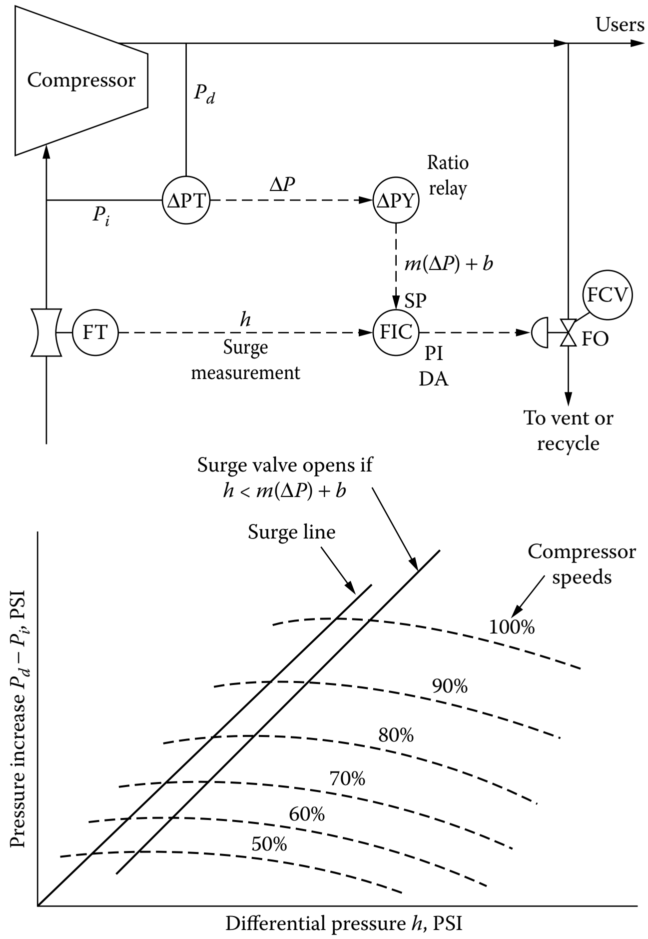


FIG. 8.15l
On a plot of ΔP vs. h, the surge curve becomes a straight line. (Adapted from Reference 4.)

point and the surge line: 1) a decrease in suction pressure; 2) an increase in suction temperature; 3) a decrease in molecular weight; 4) a decrease in specific heat ratio. These conditions will also increase the probability of surge (Figure 8.15k).

On a plot of ΔP vs. h (Figure 8.15l), these effects are most favorable. A decrease in suction pressure moves the surge line in the safe direction, temperature has no effect, and the effect of the other variables is also less pronounced. Therefore, although the ΔP vs. h plot is accurate only at low compression ratios, it does have the advantage of being independent from the effects of composition and temperature changes. However, suction pressure should be included in the model in order to be exact; most ΔP vs. h plots disregard it.

Variations in the Surge Curve Figure 8.15k shows the surge and speed characteristic curves of a centrifugal compressor; Figure 8.15m shows these curves for an axial compressor. The characteristic curves of the axial compressor are steeper, which makes it better for constant-flow services. The centrifugal design is better for constant pressure control.

The effect of guide vane throttling is also shown for both centrifugal and axial compressors (Figure 8.15n). As can be seen, the shape of the surge curve varies with the type of equipment used. The surge curve of speed-controlled centrifugal compressors bends up (Figure 8.15k), whereas for axial (Figure 8.15m) and vane-controlled machines (Figure 8.15n), the surge lines bend over. It is this negative slope of the axial compressor's surge curve that makes it sensitive to speed variations, because an increase in speed at constant flow can quickly bring the unit into surge.

As can be seen from the above information, the shape of the surge curve varies with compression ratio and with equipment design. It should also be noted that in case of multistage compressors, the surge line is discontinuous. If the compressor characteristics are as shown in Figure 8.15o, the addition of a compressor stage causes a break point in the surge curve. With more stages, more break points would also be added, and the resulting net effect is a surge curve that bends over instead of bending up, as does the surge curve for the single-stage compressor in Figure 8.15k.

Figure 8.15o also shows the choke curve. This curve connects the points at which the compressor characteristic lines become vertical. Below this curve, flow will stay constant

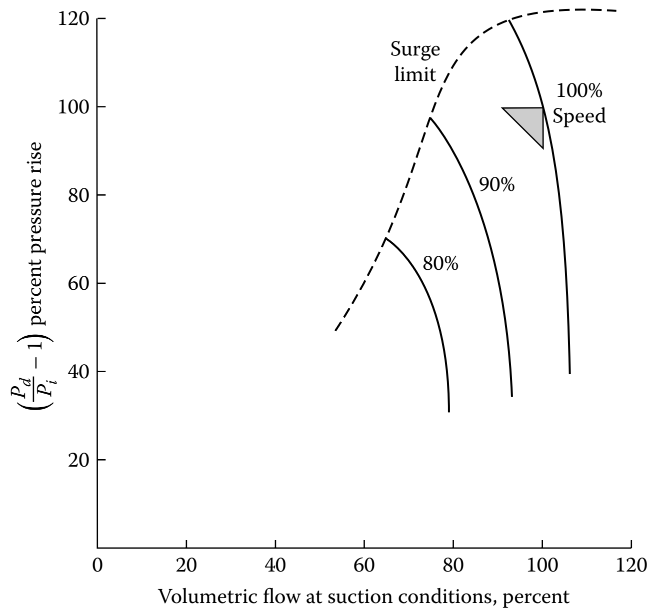


FIG. 8.15m
The surge line for an axial compressor is steep at low flows and flat at high flows.[3]

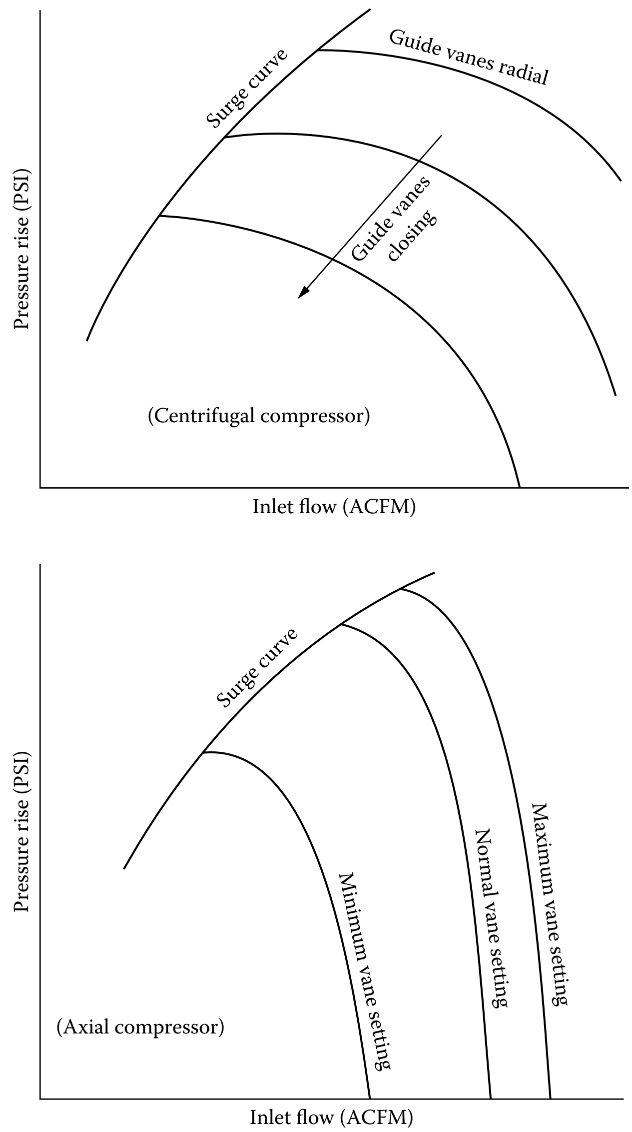


FIG. 8.15n
When guide vanes are used for throttling, both the centrifugal and the axial compressor's surge curve bends over.[1]

even if pressure varies, as long as the compressor is at a constant speed.[1] As speed is reduced, the surge and choke lines intersect. Below this intersection, the traditional methods of surge protection (venting, recycling) are ineffective; only a quick raising of the compressor speed can bring the machine out of surge.

This is similar to the situation when the compressor is being started up. As the operating point is moving on the load curve (Figure 8.15p), it must pass through the unstable region on the left of the surge curve as fast as possible in order to avoid damage from vibration.

Flow Measurement The two critical components of a surge control loop are the flow sensor and the surge valve. Both must be fast, accurate, and reliable.

Flow oscillations under surge conditions occur on a cycle of little more than a second. The flow transmitter should be fast enough to detect these. The time constants of various transmitter designs are as follows:[1]

Pneumatic with damping: Up to 16 sec
Electronic *d/p*: 0.2–1.7 sec
Diffused silicone *d/p*: Down to 0.005 sec

Only the diffused silicone-type sensor design is fast enough to follow the precipitous flow drop that occurs at the beginning of surge or the oscillations during surge. Measurement noise is another serious concern, because it necessitates a greater margin between the surge and the control lines. Noise can be minimized by the use of 20 pipe diameters of upstream and 5 diameters of downstream straight runs around the streamlined flow tube-type sensor. Noise will also be reduced if the low-pressure tap of the *d/p* cell is connected to a piezometric ring in the venturi-type flow tube (Figure 8.15q). The addition of straightening vanes will also contribute to the reduction of noise.

Antisurge control usually requires a flow sensor on the suction side of the compressor. If good, noise-free flow measurement cannot be obtained on that side, a corrected discharge side differential pressure reading (*hd*) can be substituted. Equation 8.15(3) can be used to obtain the suction side differential pressure (*hs*) from readings of (*hd*) and the suction plus discharge pressures and temperatures (*Ps*, *Pd*, *Ts*, and *Td*):

$$hs = hd(Pd/Ps)(Ts/Td) \qquad \textbf{8.15(3)}$$

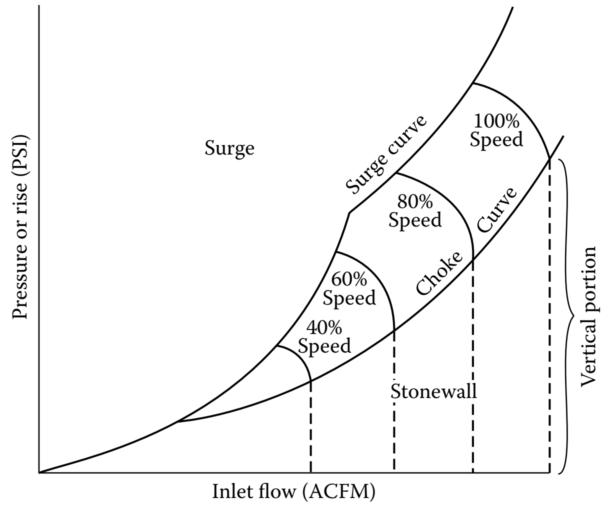


FIG. 8.15o
With multistage compressors, operation must be confined to the area between the surge and choke curves.[1]

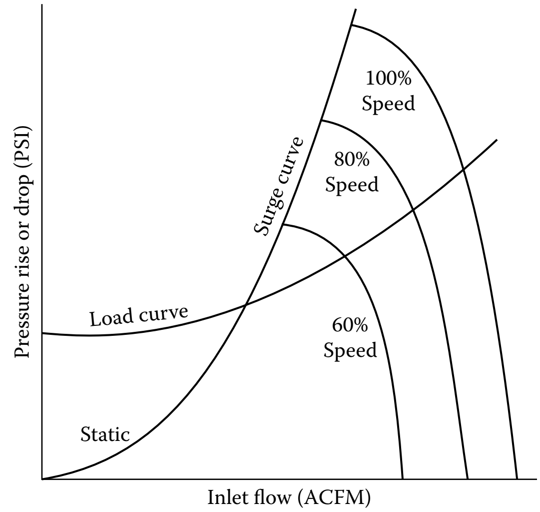


FIG. 8.15p
The speed and surge curves are both characteristics of the compressor, while the load curve is a function of the process load.[1]

Some of the performance characteristics of a number of flow tubes and flow nozzles are given in Table 8.15r.

Surge Control Valves Surge valves are usually fail-open, linear valves that are tuned for fast and precise throttling. Because the valve is the weak link in the total surge protection system, and because testing and maintenance cannot be done on-line if only one valve is used, total redundancy is recommended. Each of the two parallel surge valves should be sized for the full flow of the compressor but for only 70% of the discharge pressure. This pressure reduction is caused by the flow reversals during surge.

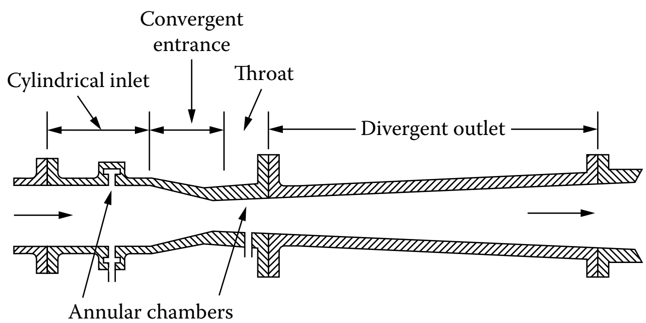


FIG. 8.15q
Herschel venturi with annular pressure chamber.

TABLE 8.15r
Venturi, Flow Tube, and Flow Nozzle Inaccuracies (Errors) in Percent of Actual Flow for Various Ranges of Beta Ratios and Reynolds Numbers

Flow Sensor		*Line size, inches (1 in. = 25.4 mm)*	*Beta Ratio*	*Pipe Reynolds Number Range for Stated Accuracy*	*Inaccuracy, Percent of Actual Flow*
Herschel standard[1]	Cast	4–32	0.30–0.75	2×10^5 to 1×10^6	±0.75%
	Welded	8–48	0.40–0.70	2×10^5 to 2×10^6	±1.5%
Proprietary true venturi[2]	Cast	2–96	0.30–0.75	8×10^4 to 8×10^6	±0.5%
	Welded	1–120	0.25–0.80	8×10^4 to 8×10^6	±1.0%
Proprietary flow tube[3]	Cast	3–48	0.35–0.85	8×10^4 to 1×10^6	±1.0%
ASME flow nozzles[4]		1–48	0.20–0.80	7×10^6 to 4×10^7	±1.0%

[1] No longer manufactured because of long laying length and high cost.
[2] Badger Meter Inc.; BIF Products; Fluidic Techniques Inc.; Primary Flow Signal Inc.; Tri-Flow Inc.
[3] ABB Instrumentation; Badger Meter Inc.; BIF Products; Preso Industries.
[4] BIF Products; Daniel Measurement and Control.

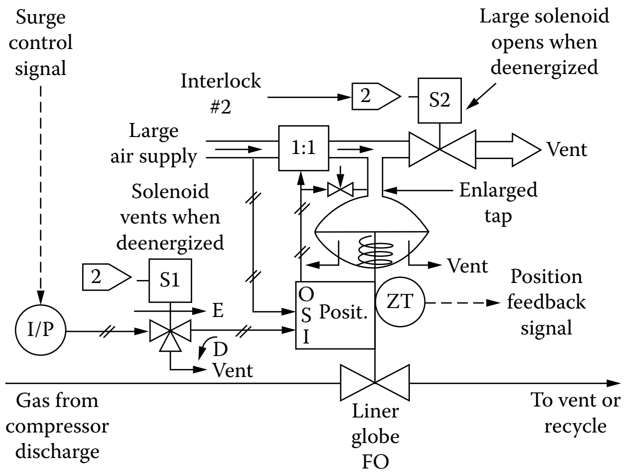


FIG. 8.15s
A well-designed surge control valve will have a positioner, a large air supply with a one-to-one booster repeater with bypass, vent solenoids, and position feedback. (Adapted from Reference 1.)

Globe valves (Section 6.19) are preferred to rotary designs because of hysteresis and breakaway torque considerations. Boosters (Figure 8.15s) can cut the sum of prestroke dead time and full-scale stroking to under 1 sec. Actually, valves as large as 18 in. have been made to throttle to any position in under a half-second.[1]

Digital valves (Section 6.18) are even faster: They can stroke in 0.1 sec without any overshoot. Their limitations are in the plugging of the smaller ports and in the difficulty of inspecting them.

Positioners are frequently required to reduce hysteresis (packing friction), dead band (boosters), disc flutter (butterfly valves), and overshoot; they are also often needed just to provide the higher air pressure required by piston actuators.

Figure 8.15s shows some of the main components of a surge control valve.[1] The electronic control signal is converted to a pneumatic one by the I/P converter, and this pneumatic signal is sent to the positioner through a three-way solenoid ($S1$). The output signal from the positioner is sent simultaneously into a booster relay (1:1) and an adjustable bypass. The output from the booster relay is connected by short- and large-diameter tubes to the valve actuator with an enlarged pressure tap and to the large venting solenoid ($S2$). Some of the operational features of the system are as described below.

When interlock #2 is de-energized, requiring instantaneous full opening of the surge valve (because of discharge line blockage or other reasons), $S1$ vents the positioner inlet and $S2$ vents the actuator top-work. Because $S2$ is large, air removal is fast and the valve opens quickly. On return to normal, both solenoids are energized. This closes $S2$ and opens $S1$ to the control signal.

Depending on the size or restriction in the line from $S1$, air might enter the system more slowly and the valve might close more slowly than it opened. Such fast-opening, slow-closing designs can respond to the first precipitous drop in flow and thus can prevent the second surge cycle from developing. It is important not to slow the speed of valve opening too much, because quick throttling is still required.

The valve must also respond quickly to the control signal and throttle quickly in either direction. In order to speed up the air movement into and out of the actuator, the vent and signal ports on the actuator are both drilled out, an air booster is installed, and a large-capacity path is established between the booster and the actuator. In order to reduce the dead band of the 1:1 booster, it is advisable to use a signal range of 6–30 PSIG instead of the usual 3–15 PSIG.

The bypass needle valve around the booster in Figure 8.15s is required because without it the volume on the outlet of the positioner would be much smaller than on the outlet of the booster. This would allow the positioner to change the input to the booster faster than the booster could change its output, resulting in a limit cycle. This limit cycle is eliminated by the addition of the adjustable bypass.

All surge valves should be throttle-tested before shipment. It is also desirable to monitor the surge valve opening through the use of a position transmitter.

Surge Control Curves As was shown in Figure 8.15l, a parabolic surge curve with a positive slope will appear as a straight line on a ΔP vs. h plot. The purpose of surge control is to establish a surge control line to the right of the actual surge line, so that corrective action can be taken before the machine goes into surge. Such a control system is shown in Figure 8.15t.

The biased surge control line is implemented through a biased ratio relay (ΔPY), which generates the set point for FIC as follows:

$$SP = m(\Delta P) + b \qquad \textbf{8.15(4)}$$

where

SP = desired value of h in inches H_2O
m = slope of the surge line at the operating point
ΔP = compressor pressure rise in psi
b = bias of the surge set point in inches H_2O

The offset between the surge and the control lines should be as small as possible for maximum efficiency, but it must be large enough to give time to correct upsets without violating the surge line.[6] The slower the upsets and the faster the control loop, the less offset is required for safe operation. In a good design, the bias b is about 10%, but in bad ones, as disturbances get faster or instrument slower, it can grow to 20%.

A second line of defense is the backup interlock FSL. It is normally inactive, as the value of h is normally above the FIC set point SP. When the surge controller FIC is not fast enough to correct a disturbance, h will drop below SP. FSL

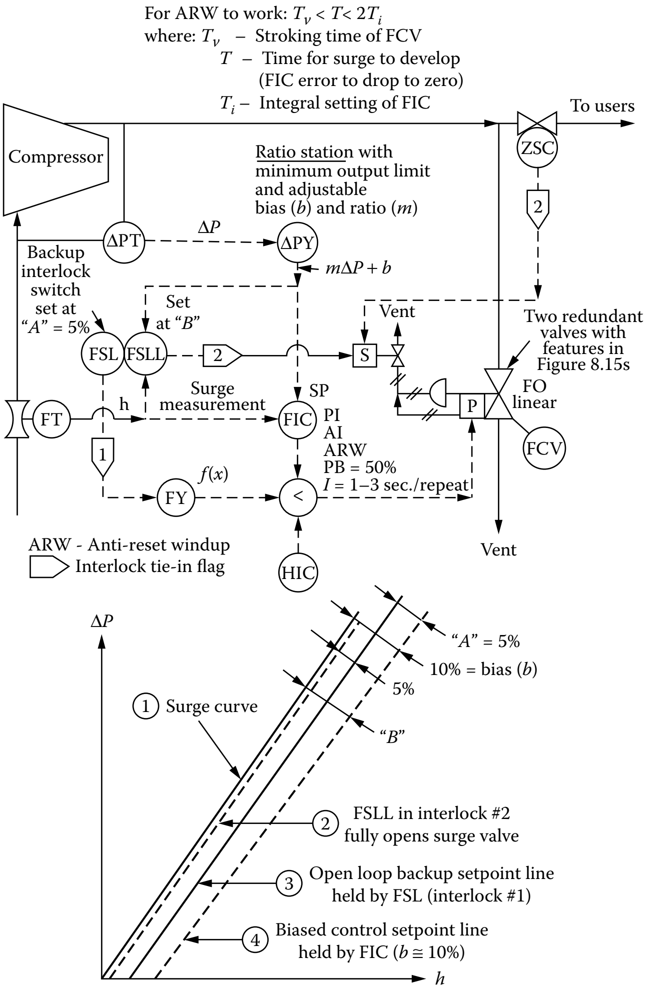


FIG. 8.15t
The surge control system (top) and the surge margin settings for controller set point and interlocks (bottom) for compressors operating at low compression ratios. (Adapted from Reference 1.)

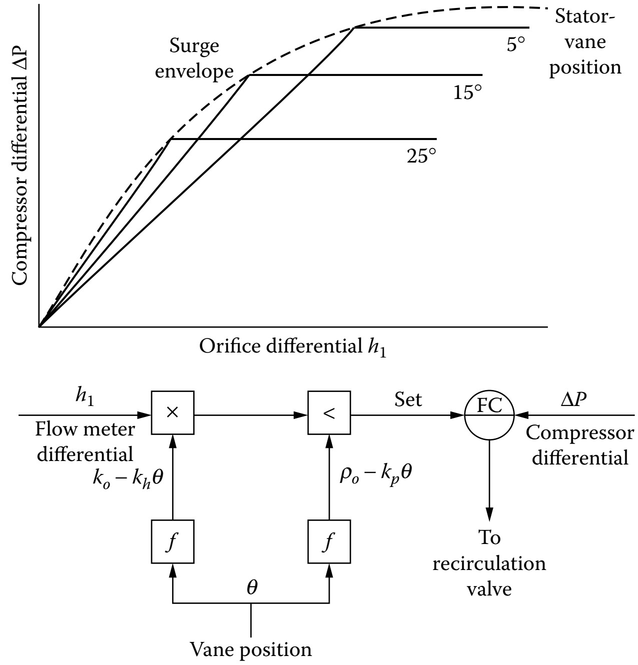


FIG. 8.15u
If the surge line of the compressor bends over, the control loop should measure both the flow meter differential and the vane position. (Adapted from Reference 3.)

is set to actuate when h has fallen 5% below SP, and at that point it fully opens the surge valve through interlock #1.

The shape of the control and backup lines shown in Figure 8.15t is applicable only when the surge curve is parabolic. As was shown in the surge curves in Figures 8.15m, 8.15n, and 8.15o, the surge curves can also be linear or have negative slopes and discontinuities. If the surge line is not parabolic, the following techniques can be considered for generating the surge control lines.

If the surge curve is linear, the signal h should be replaced by a signal representing flow. This can be obtained by adding a square root extractor to the FT in Figure 8.15t. If the surge curve bends down, the use of two square root extractors in series[1] can be used to approximate its shape.

On multistage machines with high compression ratios, it might be necessary to substitute a signal characterizer for ΔPY in Figure 8.15t. On variable vane designs, it has been proposed that the flowmeter differential (h) could be corrected to be a function of actual vane position to better approximate the surge curve[3] (see Figure 8.15u). If a flowmeter differential cannot be obtained from either the suction or the discharge side of the compressor, the surge curve can also be approximated on the basis of speed, power, or vane position measurements.[3]

The Surge Flow Controller The surge controller (FIC) in Figure 8.15t is a direct-acting controller with proportional and integral action and with antireset windup (AWR) features. It is a flow controller with a remote set point and a narrower proportional band than would be used for flow control alone. In a well-tuned surge controller, both the proportional band (PB) and the integral settings must be minimized (PB ~ 50%, 1–3 sec/repeat). With this level of responsiveness, the FIC set point can be less than 10% from the surge curve, as shown in Figure 8.15t.

Under normal conditions, the actual flow (h) is much higher than the surge control limit (SP), and therefore the integral mode of the FIC has a tendency to wind up on its positive error toward a saturated maximum output. If this was allowed to happen, the controller could not respond to a surge condition (a quick drop in h), because it would first need to develop a negative error equal to that which caused the saturation.

The aim of an antireset windup feature in this controller is to hold the FIC output under normal conditions at around

105%, so that the valve is closed but the signal is just lingering above the 100% mark without saturation.

As soon as h starts dropping toward the set point, the valve should start to open, and it should reach full opening before the set point is reached. As soon as h starts dropping, the proportional contribution to the FIC output decreases. If the approach is slow, the increase in the reset contribution will be greater than the decrease caused by the proportional contribution. Therefore, if h does not drop to SP, the valve slowly returns to the closed position. The effect of the internal feedback in the FIC is to activate the integral action only when the valve is not closed, while keeping the proportional action operational all the time.

The main goal of a feedback surge controller is to protect the machine from going into surge. Once the compressor is in surge, the FIC is not likely to be able to bring it out of it, because the surge oscillations are too fast for the controller to keep up with them. This is why backup systems are needed.

The surge controller is usually electronic, and the more recent installations tend to be microprocessor-based. Such digital units can memorize complex, nonlinear surge curves and can also provide adaptive gain, which is a means of increasing controller gain as the operating point approaches the surge curve. In digital controllers, it is desirable to set the sample time at about one quarter of the period of surge oscillation, or at 0.3 sec, whichever is shorter. If a flow-derivative backup control (Figure 8.15v) is used, the sample time must be 0.05 sec or less, because if it is slower, the backup system would miss the precipitous drop in flow as surge is beginning.

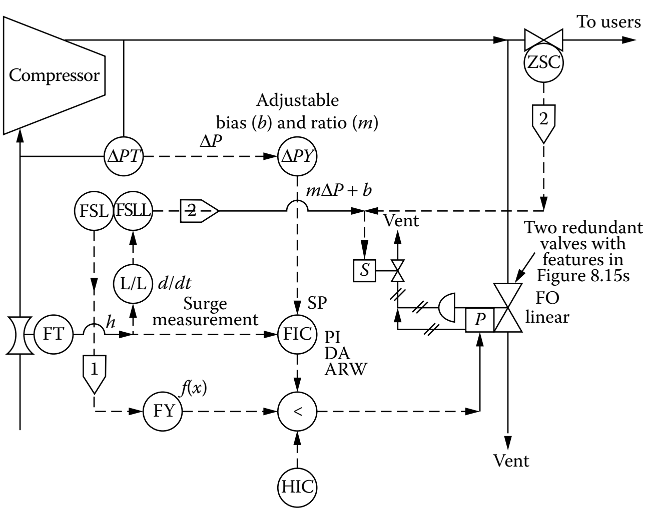


FIG. 8.15v
In this configuration the interlock backup (FSLL) is triggered by measuring the rate of flow change (L/L). If this is done by a digital control system, the sample time must be 0.05 sec or less. (Adapted from Reference 1.)

Tuning the Controller The *closed loop ultimate oscillation* method cannot be used for tuning the surge loop, because there is a danger of this oscillation triggering the start of surge. The *open loop reaction curve* technique of tuning is used with both the surge and the throughput controller in manual. The output of each controller is changed by 10%, and the resulting measurement response is analyzed on a high-speed recorder, yielding the time constant and the dead time of the process (Figure 8.15w).

The tuning settings of the surge controller can be tested by first increasing the bias (b in Figure 8.15t) until the surge valve opens and then analyzing the loop response. The FIC must be so tuned that its overshoot is less than the distance between the FSL and FIC set points (A in Figure 8.15t). The throughput controller should be kept in automatic while the surge controller is tested so that interaction problems will be noted. Once the surge controller is tuned, it should not be switched to manual but should remain always in automatic.

The antireset windup feature is not effective on slow disturbances, because the integral contribution that increases the FIC output outweighs the proportional contribution that lowers it; thus, the valve stays closed until the surge line is crossed. In order for the ARW to be effective, the time for the controller error to drop from its initial value to zero (T) must fall within two limits. It must be slower than the stroking time of the surge valve (Tv), but it must be faster than two integral times of the controller:[1]

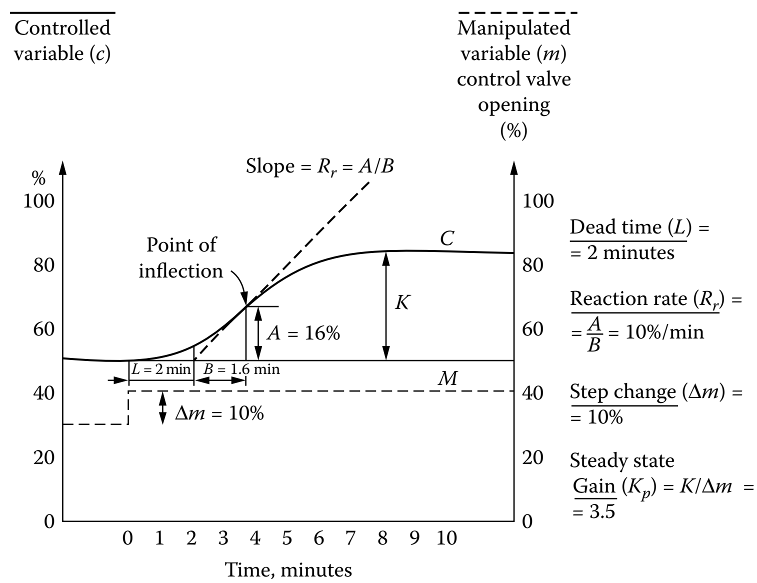
$$Tv < T < 2Ti \qquad \textbf{8.15(5)}$$

If the above requirement cannot be satisfied—because, for example, one of the potential disturbances is a slow closing discharge valve—an “anticipator” control loop should be added (Figure 8.15z).

Surge Protection Backup The purpose of the FIC in Figure 8.15t was to protect the compressor from going into surge. The feedback surge controller is usually not fast enough to bring the machine out of surge once it has developed, because the surge oscillations are too fast for this controller. Yet even with the best feedback surge controller design, the machine will go into surge.

This can be intentionally initiated to verify the surge curve, it can be caused by errors or shifts in the surge curve as the unit ages, and it can occur because of misoperation or failure of either the controller or downstream block and check valves. For these reasons, a second line of defense—a backup system—is required in addition to the surge FIC.

The backup system usually consists of two subsystems, identified as interlocks #1 and #2 in Figure 8.15t. In interlock #2, FSLL detects conditions that will cause the complete stopping of all forward flow from the compressor; when such a condition is detected, FSLL instantaneously and fully opens



Recommended Ziegler-Nichols settings (noninteracting)

Type of controller \ Settings	Proportional	Integral (min/repeats)	Derivative
Proportional	$100\ R_rL/\Delta m = 200\%$	–	–
PI controller	$111\ R_rL/\Delta m = 222\%$	$3.33\ L = 6.66$	–
PID controller	$83\ R_rL/\Delta m = 166\%$	$2\ L = 4.0$	$0.5\ L = 1.0$

FIG. 8.15w
An example of open loop tuning where the settings are determined on the basis of the dead time (L) and reaction rate (R_r) of the controlled process.

the surge valve. One such condition in Figure 8.15t is signaled by a closed position limit switch (ZSC) on the block valve.

Interlock #1 in Figure 8.15t provides open-loop backup as follows: The FSL detects an approach to surge that is closer than the FIC set point; when it detects such a condition, it takes corrective action. Under normal conditions, FSL senses an *h* value that is higher than SP, meaning that the operating point is to the right of curve ④. As surge approaches, the operating point crosses the control line (curve 4) as *h* drops below SP. The FSL is set to actuate interlock #1 when *h* has crossed curve ③. This occurs when the flow has dropped below the FIC set point by the amount "*A*" (usually 5%).

When interlock #1 is actuated, it triggers the signal generator FY to drop its output signal to zero and then, when the operating point has returned to the right of curve ③, gradually increase it back to 100%. When the FY output drops to zero, the low signal selector (in the output of the FIC) will select it and send it to the valve. This causes the surge valve to open fully in less than a second, followed by a slow closure according to the program in the signal generator FY. As the FY output rises, it will reach the output of the FIC, at which point control is returned from the backup system to the feedback controller.

This type of backup is called the *operating point method.* Its advantage is that it takes corrective action *before* the surge curve (curve ①) is reached. Its disadvantages are that it is not usable on multistage machines or on systems with recycle and that the protection it provides is lost if Δ*PY* in Figure 8.15t fails by dropping its output to zero. For this reason, it is desirable to provide a minimum limit on the output of Δ*PY.*

The manual loader input (HIC) to the low-signal selector is used during start-up. Because its output passes through the low selector, it can increase but cannot decrease the valve opening.

Flow-Derivative Backup The difference between the operating point technique of backup (Figure 8.15t) and the flow-derivative method (Figure 8.15v) is that the first method acts before the surge curve is reached, whereas the second is activated by the beginning of surge. Because flow drops off very quickly at the beginning of surge, the instruments that measure the rate of this drop must be very fast.

If implemented digitally, the sample time of the flow derivative loop must be 0.05 sec or less in order not to miss the initial drop in flow. The lead-lag station (*L*/*L*) that detects the rate of flow reduction is adjusted so that the lag setting will filter out noise and the lead setting will give a large output when the flow drops.

The function of interlock #1 is the same as was described in connection with Figure 8.15t. When FSL detects the drop in flow, it causes the signal generator (FY) output to drop to zero, fully opening the valve within a second. If this action succeeds in arresting the surge, the signal generator output slowly rises and control is returned to the feedback FIC.

If interlock #1 does not succeed in bringing the machine out of surge, then after a preset number of oscillations FSLL is actuated. This triggers interlock #2, which keeps the surge valve fully open until surge oscillations stop.

The flow derivation method of backup has the advantage of providing protection even if the compressor pressure-rise instruments (ΔPT, ΔPY) fail, but it also has the disadvantage of not being usable when the flow signal is noisy.

Optimized Adaptations of Surge Curve The surge curve shifts with wear and with operating conditions. Figure 8.15x describes a surge control loop that recognizes such shifts and automatically adapts to the new curve. The adaptation subroutine consists of two segments, the set point adaptation section (blocks ① to ⑥) and the output backup section (blocks ⑦ to ⑪).

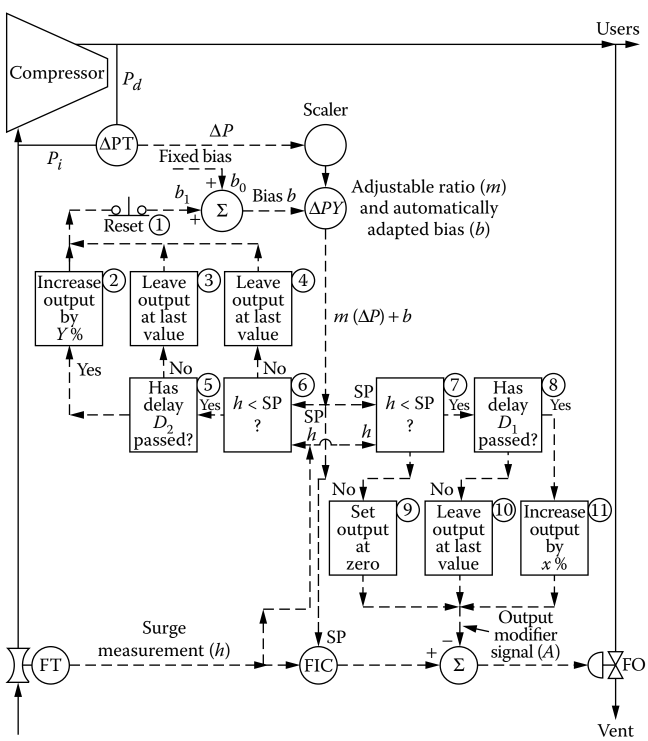


FIG. 8.15x
This control system recognizes changes in the surge curve and adapts the surge controller to the new curve. (Adapted from Reference 1, describing U.S. Patent 856,302, owned by Naum Staroselsky.)

The purpose of the output backup section is to recognize the approach of a surge condition that the feedback controller (FIC) was unable to arrest and to correct such a condition when it occurs. As long as the surge measurement (h) is not below the set point (SP), no corrective action is needed, and therefore blocks ⑦ and ⑨ will set the FIC output modifier signal (A) to zero. Once h is below SP, the operating point is to the left of curve ④ in Figure 8.15t, and the output of block ⑦ in Figure 8.15x is switched to "Yes."

Next, block ⑧ checks if the adjustable time delay D1—having typical values of 0.3 to 0.8 sec[1]—has passed. If it has not, signal A remains at its last value. If D1 has passed, A is increased by an increment X. Typical values of the X increment range from 15 to 30%.[1] As A is subtracted from the FIC output signal, this backup loop will open up the surge valve at a speed of 15–30% per 0.3–0.8 sec. When h is restored to above set point, the signal A is slowly returned to zero, allowing the surge valve to reclose.

The purpose of the set point adaptation loop (blocks ① to ⑥) is to recognize shifts in the surge curve and, as a response, to move the control line ④ in Figure 8.15t to the right by increasing to total bias b of the ratio relay ΔPY. The logic of blocks ② to ⑥ is similar to that described for blocks ⑦ to ⑪, except that the resulting variable bias signal (b_1) is added to the fixed bias (b_0) to arrive at the adapted new bias (b). The speed of set point adaptation does not need to be as fast as the opening of the surge valve. Therefore, the time delay D2 tends to be longer than D1, and the increment Y is smaller than X. The purpose of the reset button in block ① is to provide a means for the operator to reset the b_1 signal back to zero.

Override Controls

Figures 8.15t and 8.15v show backup and overrides implemented by a low-signal selector on the outlet of the feedback FIC. Figure 8.15y shows some additional overrides that might

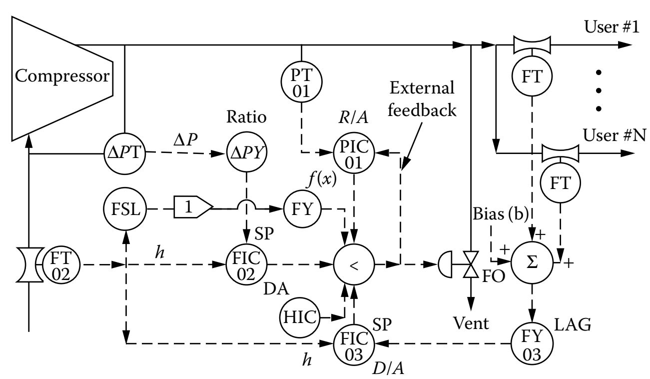


FIG. 8.15y
The low-signal selector provides this control system with a high-pressure (PIC-01) and a user shutdown (FIC-03) override. (Adapted from Reference 1.)

be desirable in some installations. Under normal conditions all override signals are above 100%, and they leave the feedback FIC in control. Sending the override signals through the low-signal selector guarantees that they can only increase the opening of the surge valve, not decrease it.

The high-pressure override controller (PIC-01 in Figure 8.15y) protects downstream equipment from overpressure damage. Normally its measurement is far below its set point, and therefore its output signal is high. In order to prevent reset windup, external feedback is used.

The other override loop (FIC-03) guarantees that the shutdown of a user or a sudden change in user demand will not upset the mass balance (balance between inlet and outlet flow) around the compressor. This is an important feature, because without it the resulting domino effects could shut down other users or the whole plant.

In Figure 8.15y, when a user is shut down, FT-03 detects the drop in total flow within milliseconds, if a fast flow transmitter is used. This drop in flow reduces the measurement signal (h) to FIC-03, while its set point is still unaltered because of the lag (FY-03). Therefore, the surge valve is opened to maintain the mass balance. As FY-03 slowly lowers the set point to the new user flow rate (plus bias), FIC-03 recloses the valve. The bias b is used to keep the set point of FIC-03 above its measurement, thus keeping the surge valve closed when conditions are normal.

Figure 8.15z illustrates another method of maintaining mass balance as users are suddenly turned off. The velocity limiter (VY-03) limits the rate at which the surge signal (h) can be lowered. If the surge signal drops off suddenly, the VY-03 output remains high and, through the high signal selector, keeps the surge controller set point up. This opens the surge valve to maintain the mass balance until it is reclosed by the limiter. This method is called an *anticipator algorithm,* because it opens the surge valve sooner than the time when the operating point (h) would otherwise drop to the set point, computed by the ratio station (ΔPY).

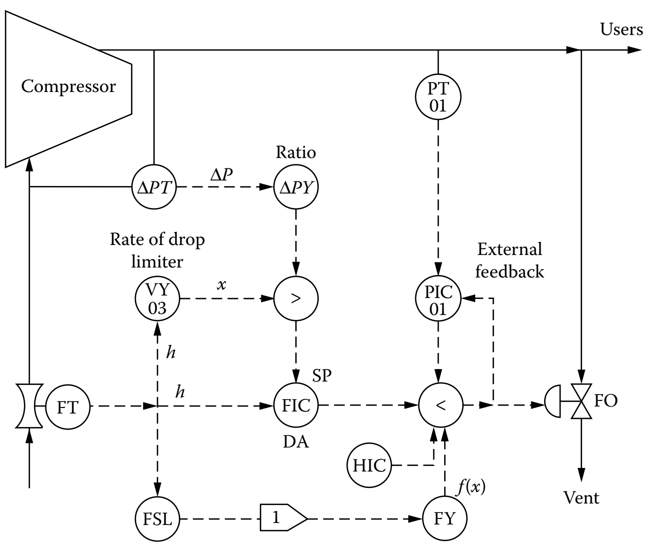


FIG. 8.15z
The velocity limiter serves to protect against sudden load changes by temporarily opening the surge vent valve when the load is quickly dropping.[7] This is achieved by providing VY-03, which, in case of a sudden drop in h, does not immediately lower x and therefore it (x) is selected by the high selector as the set point.

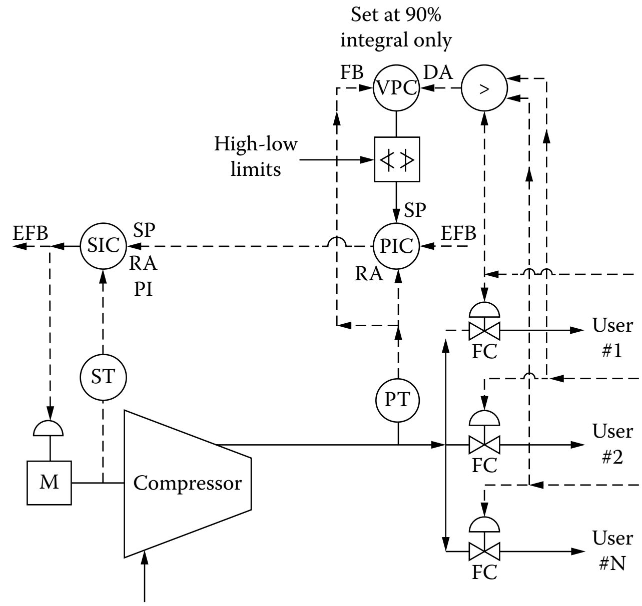


FIG. 8.15aa
By keeping the compressor's discharge pressure just high enough so that even the most open user valve does not open beyond 90%, the needs of all users are satisfied with a minimum investment of energy.

Optimized Load-Following

When a compressor is supplying gas to several parallel users, the goal of optimization is to satisfy all users with the minimum investment of energy. The minimum required header pressure is found by the valve position controller (VPC) in Figure 8.15aa. It compares the highest user valve opening (that identifies the needs of the most demanding user) with a set point of 90%, and if even the most open valve is not yet 90% open, it lowers the header pressure set point.

This supply-demand matching strategy not only minimizes the use of compressor power but also protects the users from being undersupplied, because it protects any and all supply valves from the need to open fully. On the other hand, as the VPC causes the user valves to open farther, these valves not only reduce their pressure drops but also become more stable.

The VPC is an integral-only controller, with its integral time set for about 10 times that of the PIC. This guarantees

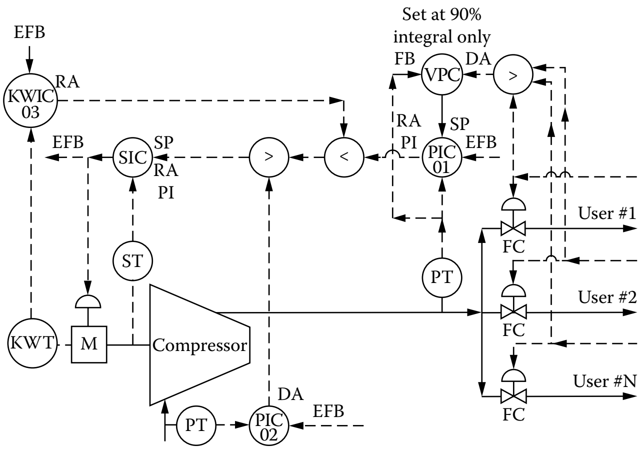


FIG. 8.15bb
Protective overrides can be added to optimized load-following controls, so that the system is protected against excessively low pressures on the suction side of the compressor or from overloading the compressor's motor drive.

that the VPC will act more slowly than all the user controllers, thus giving stable control even if the user valves are unstable. The external feedback (FB in Figure 8.15aa) protects the VPC from reset windup when its output is limited or when the PIC has been switched to manual.

In addition to following the load, it is also necessary to protect the equipment. In Figure 8.15bb, one protective override prevents the development of excessively low suction pressures (PIC-02), which could result in drawing oil into the compressor. The other override (KWIC-03) protects from overloading the drive motor and thereby tripping the circuit breaker.

In order to prevent reset windup when the controller output is blocked from affecting the SIC set point, external feedback (EFB) is provided for all three controllers. This arrangement is typical for all selective or selective-cascade control systems.

Interaction and Decoupling

Both the load and the surge control loops are shown in Figure 8.15cc. The manipulated variable of both of these loops is the compressor throughput. Under normal conditions, there is no problem of interaction; because the surge loop is inactive, its valve is closed.

Under abnormal conditions, when point "*A*" in Figure 8.15dd is reached, FIC-02 quickly opens the surge valve, which causes the discharge pressure to drop off as the flow increases. PIC-01 responds by increasing the vane setting or speed of the machine.

The faster the PIC-01 loop instrumentation is in Figure 8.15cc and the higher its gain (the narrower the proportional band), the larger and faster will be the increase in the speed of the compressor. If the load curve is steeper than the surge curve

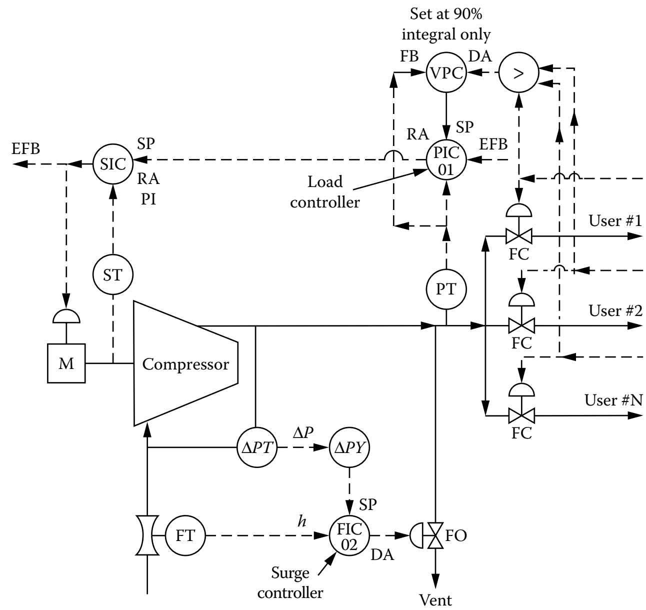


FIG. 8.15cc
The load controller (PIC-01) and surge controller (FIC-02) both affect the compressor throughput.

(as in Figure 8.15dd), the action of PIC-01 will bring the operating point closer to the surge line. In this case, a conflict exists between the two loops, because as FIC-02 acts to correct an approaching surge situation, PIC-01 responds by worsening it. The better tuned (narrower proportional band) and faster (electronic or hydraulic speed governors) the PIC-01

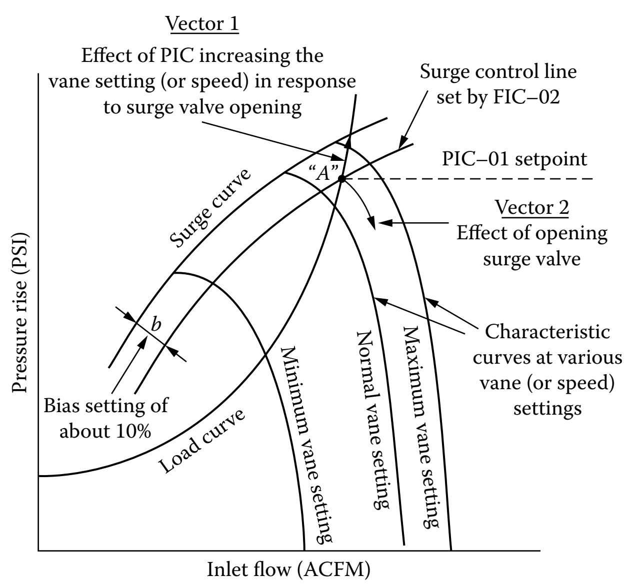


FIG. 8.15dd
If the load curve is steeper than the surge curve and if the controlled variable is pressure, the load and surge control loops will fight each other. (Adapted from Reference 1.)

TABLE 8.15ee
Type of Interactions between Load and Surge Controller

Controlled Variable	*Load Curve Steeper than Surge Curve*	*Nature of Interaction*
Pressure	Yes	Conflict
	No	Assist
Flow	Yes	Assist
	No	Conflict

TABLE 8.15ff
The Nature of the Interaction as a Function of the RG Value

RG Value	*Effect of Other Loop*
0 to 1.0	Assists the primary loop
Above 1.0	Conflicts with the primary loop
Below 0	Conflict that also reverses the action of primary loops

loop is, the more dangerous is its effect of worsening the approach of surge.

The throughput controller (PIC-01 in Figure 8.15cc) moves the operating point on the path of the load curve. The surge controller (FIC-02 in Figure 8.15cc) moves the operating point on the path of the characteristic curve of the compressor at constant speed. If the characteristic curve is steep, the action of the surge controller will cause a substantial upset in the discharge pressure controlled by PIC-01 (Figure 8.15cc). If the load curve is flat, the effect of the pressure controller on the surge controller will be the greatest.

If the throughput is under flow control, the opposite effects will be observed. The effect of the surge controller on the flow controller will be the greatest when the characteristic curves of the compressor are flat, and the effect of the flow controller on the surge controller will be greatest when the load curve is steep.

As shown in Table 8.15ee, whether the loops will assist or conflict with each other, both the relative slopes of the load and surge curves and the variable selected for process control need to be considered.

As both loops try to position the operating point on the compressor map, the resulting interaction can cause the type of inverse response that was described in Figure 8.15dd, or it can cause oscillation and noise. The oscillating interaction is worst if the proportional bands and time constants or periods of oscillation are similar for the two loops. Therefore, if tight control is of no serious importance, one method of reducing interactions is to reduce the response (widen the proportional band) of the load controller. Similarly, the use of slower actuators (such as pneumatic ones) to control compressor throughput will also reduce interaction, but at the cost of less responsive overall load control.

Relative Gain In evaluating the degree of conflict and interaction between the load and surge loops, it is desirable to calculate the relative gain between the two loops. The relative gain is the ratio between the open-loop gain when the other loop is in manual, divided by the open-loop gain when the other loop is in automatic. The open-loop gain of PIC-01 (Figure 8.15cc) is the ratio of the change in its output to a change in its input that caused it. Therefore, if a 1% increase in pressure results in a 0.5% decrease in compressor speed, the open-loop gain is said to be –0.5. Assuming that the open-loop gain of PIC-01 is –0.5 when FIC-02 is in manual and the switching of FIC-02 into automatic causes the PIC-01 loop gain to drop to –0.25, the relative gain is 0.5/0.25 = 2. Table 8.15ff lists the correct interpretations of the relative gain (RG) values.

If the calculated relative gain values are put in a 2-by-2 matrix, the best pairing of controlled and manipulated variables is selected by choosing those that will give the least amount of conflict. These are the RG values between 0.75 and 1.5, preferably close to 1.0. The regions of inhibition, reinforcement, and reversal are shown in Figure 8.15gg.

Decoupling Decoupling is the means of reducing the interaction between the surge and the load control loops. If the goal of decoupling is to maintain good pressure control even during a surge episode, the system shown in Figure 8.15hh can be used. In this system, as the antisurge controller opens the vent valve, a feedforward signal (*X*) simultaneously increases the speed of the compressor in proportion. The negative sign at the summing device is necessary because the surge valve fails to open. If the two vectors shown in Figure 8.15dd are correctly weighed in the summer, the end

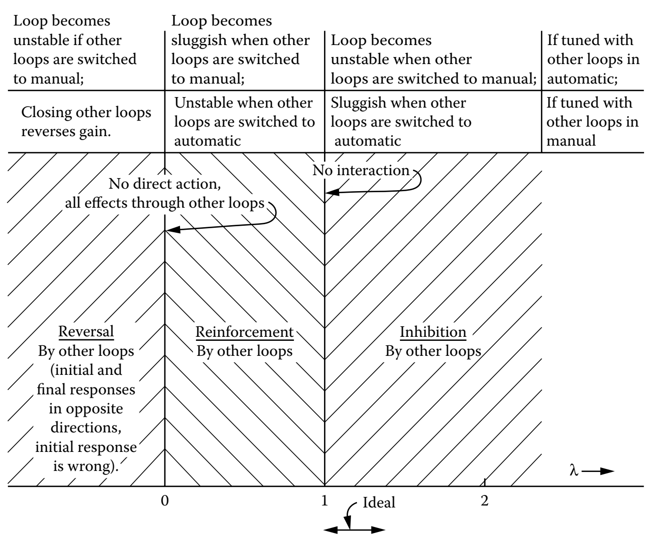


FIG. 8.15gg
The relative gain spectrum.

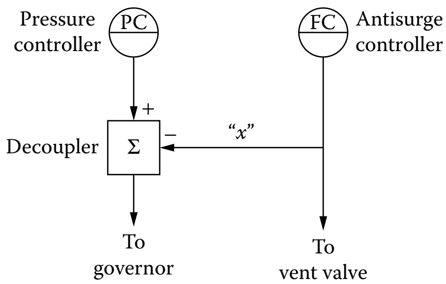


FIG. 8.15hh
The compressor discharge pressure can be kept unaffected by a surge episode, if the opening of the surge vent valve is converted into a proportional increase in speed setting of the governor.[3]

result of their summation will be a horizontal vector to the right, and PIC-01 will stay on set point.

On the other hand, if the main goal of decoupling is to temporarily reduce the size of vector #1 in Figure 8.15dd so that it will not contribute to the worsening of the surge condition, then the half-decoupling configuration shown in Figure 8.15ii can be considered. Here the decoupling summer also receives a feedforward signal (X) corresponding to the opening of the surge valve, but it acts to slow down (not speed up) the machine.

As the output of FIC-02 drops, the compressor speed is temporarily lowered, because the added value of X is reduced. This brings the operating point farther away from the surge line, as shown in Figure 8.15dd. The lagged signal Y later on eliminates this bias, because when its time constant has been reached, the values of X and Y will be equal and will cancel each other. Therefore, this decoupler will serve to temporarily desensitize a fast and tightly turned PIC-01 loop, which otherwise might worsen the situation by overreacting to a drop in pressure when the vent valve opens.

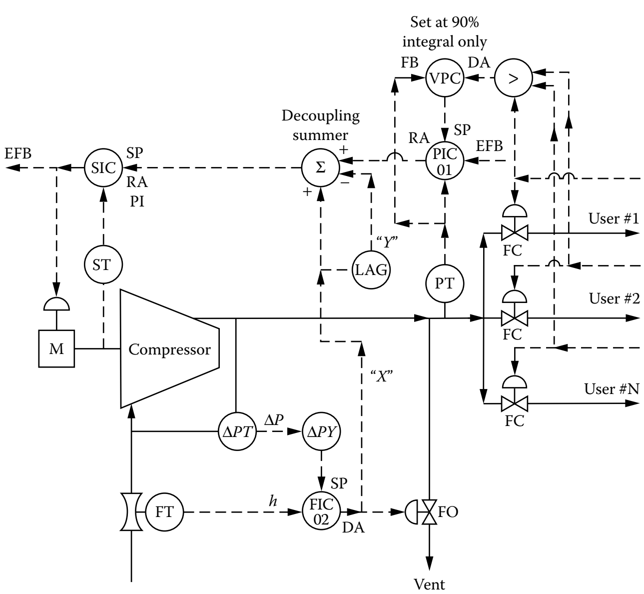


FIG. 8.15ii
The interaction between surge and load control loops can also be decoupled by temporarily reducing the speed of the compressor simultaneously with the opening of the surge valve. (Adapted from Reference 1.)

Multiple Compressor Systems

Compressors can be connected in series to increase their discharge pressure (compression ratio) or they can be connected in parallel to increase their flow capacity. Series compressors on the same shaft can usually be protected by a single antisurge control system.

When driven by different shafts, they require separate antisurge systems, although an overall surge bypass valve can be common to all surge controllers through the use of a low-signal selector.[3] This eliminates the interaction that otherwise would occur between surge valves, as the opening of a bypass around one stage not only would increase the flow through the higher stages but would also decrease it through the lower ones. In some installations, this interaction has been found to be of less serious consequence than the time delay caused by the use of a single overall bypass surge valve, which cannot quickly increase flow in the upper stages.[1] In such installations it is best to duplicate the complete surge controls around each compressor in series.

If streams are extracted or injected between the compressors, which are in series and their flows are not equal, a control loop needs to be added to keep both compressors away from their surge lines by automatically distributing among them the total required compression ratio. Such a control system is shown in Figure 8.15jj.

In this system, PIC-01 controls the total discharge pressure by adjusting the speed of compressor #1. The speed of

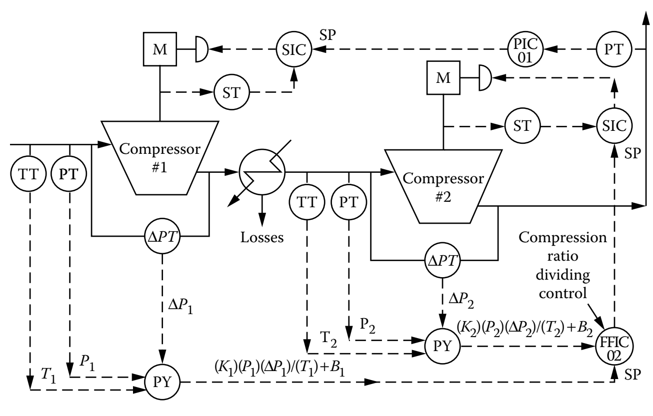


FIG. 8.15jj
When two compressors operate in series, one can be dedicated to maintain the total discharge pressure (PIC-01) from the pair, while the speed of the other can be manipulated to keep them at equal distance from their surge curves. This is achieved by FFIC-02 controlling the distribution of the total compression ratio between them. (Adapted from Reference 1.)

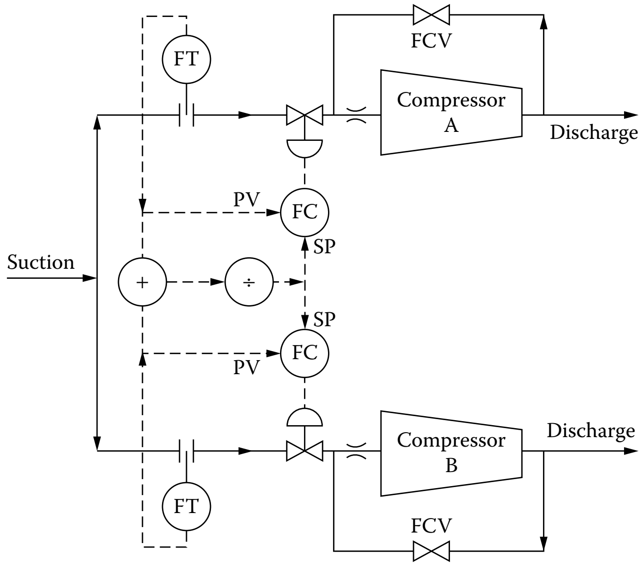


FIG. 8.15kk
When operating multiple compressors in parallel, the total flow (load) can be so distributed among the machines that one is fully loaded and the other handles the variations in demand.

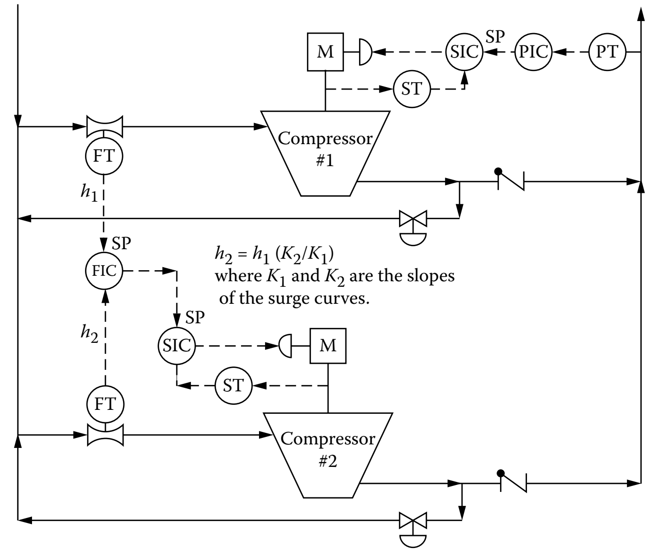


FIG. 8.15ll
Two parallel compressors can be so loaded as to keep both of them at equal distance from their surge curves. (Adapted from Reference 1.)

compressor #2 is set by FFIC-02; both compressors are thus kept at equal distances from their respective surge lines. This is accomplished by maintaining the following equality:

$$(K_1(P_1)(\Delta P_1)/(T_1)) + B_1 = (K_2(P_2)(\Delta P_2)/(T_2)) + B_2 \qquad \mathbf{8.15(6)}$$

In this equation, K is the slope and B is the bias of the surge set points of the respective compressors.

Parallel Control of Compressors Controlling two or more compressors operating in parallel and having identical characteristics would be relatively simple. It is very difficult, if not impossible, to find two compressors having identical performance characteristics. Slight variations in flow can cause one compressor to be fully loaded. The parallel machine then has useless recycle. The control scheme shown in Figure 8.15kk alleviates that problem. Typically, care is exercised to ensure that the suction valve that receives the lower flow is kept 100% open. This prevents both suction valves from going fully closed to balance the flow. For an example of flow balancing controls, see Figure 8.17e.

Figure 8.15ll illustrates how two compressors can be proportionally loaded and unloaded, while keeping their operating points at equal distance from the surge curve. The lead compressor (#1 in Figure 8.15ll) is selected either as the larger unit or as the one that is closer to the surge curve when the load rises or is further from it when the load drops.

In Figure 8.15ll it is assumed that the compressors were so chosen that their ratio of bias to slope (b/K) of the surge set point is equal. In that case,

$$h_2 = h_1(K_2/K_1) \qquad \mathbf{8.15(7)}$$

where h is the flow meter differential and K is the surge set point slope of the respective compressors.

Because of age, wear, or design differences, no two compressors are identical. A change in load will not affect them equally and each should therefore be provided with its own antisurge system.

Another reason for individual surge protection is that check valves are used to prevent backflow into idle compressors. Therefore, the only way to start up an idle unit is to let it build up its discharge head while its surge valve is partially open. If this is not done and the unit is started against the head of the operating compressors, it will surge immediately. The reason why the surge valve is usually not opened fully during start-up is to protect the motor from overloading.

Improper distribution of the load is prevented by measuring the total load (summer #9 in Figure 8.15mm) and assigning an adjustable percentage of it to each compressor by adjusting the set points of FFIC-01 and FFIC-02.

Optimization of Efficiency The load distribution can be computer-optimized by calculating compressor efficiencies (in units of flow per unit power) and loading the units in the order of their efficiencies. The same goal can be achieved if the operator manually adjusts the ratio settings of FFIC-01 and -02.

In the control system of Figure 8.15mm, the pressure controller (PIC-01) directly sets the set points of SIC-01 and -02 while the balancing controllers (FIC-01 and -02) slowly bias those settings. This is a more stable and responsive configuration than a pressure-flow cascade, because the time constants of the two loops are similar.

The output of PIC-01 must be corrected as compressors are started or stopped. One method of handling this is illustrated

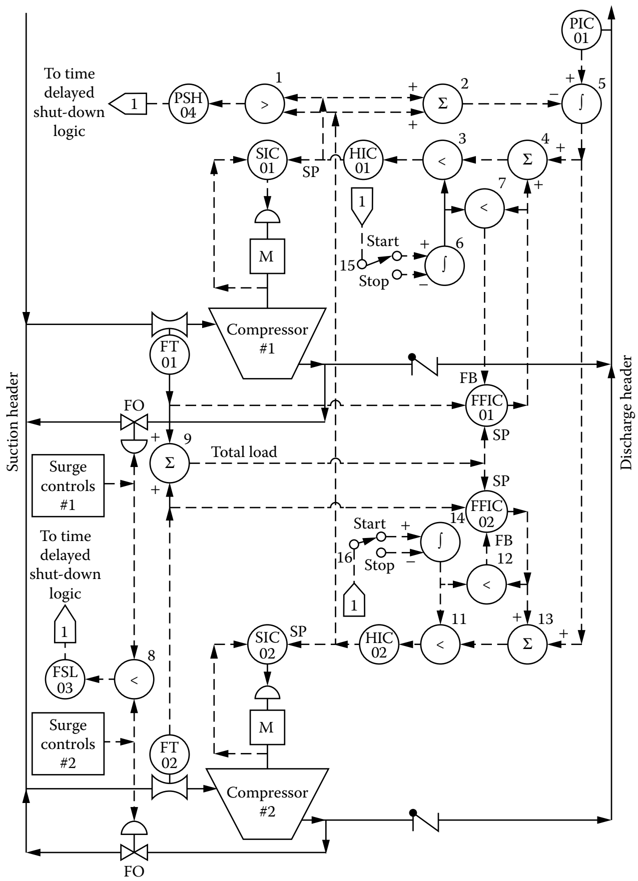


FIG. 8.15mm
A flow-balancing bias can be superimposed on direct pressure control. This control system can distribute the load between the two machines in the order of their efficiencies. (Adapted from Reference 3.)

in Figure 8.15mm. Here a high-speed integrator (item #5) is used on the summed speed signals to ensure a correspondence between the PIC output signal and the number of compressors (and their loading) used. The integrator responds in a fraction of a second and therefore does not degrade the speed of response of the PIC loop.

Figure 8.15mm also illustrates the automatic starting and stopping of individual compressors as the load varies. When the total flow can be handled by a single compressor or when any of the surge valves open, FSL-03 triggers the shutdown logic interlock circuit #1 after a time delay.

Operation with an open surge valve would be highly inefficient, because the recirculated gas is redistributed among all the operating units. When a compressor is to be stopped, item #15 (or #16) is switched to the stop position, causing the integrator #6 (or #14) to drive down until it overrides the control signal in low selector #3 (or #11) and reduces the speed until the unit is stopped.

Automatic starting of an additional compressor is also initiated by interlock #1 when PSH-04 signals that one of the compressors has reached full speed. When a compressor is to be started, interlock #1 switches item #15 (or #16) to the start position, causing the integrator #6 (or #14) to drive up (by applying supply voltage to the integrator) until PIC-01 takes over control through the low selector #3 (or #11). The integrator output will continue to rise and then will stay at maximum, so as not to interfere with the operation of the control loop.

The ratio flow controllers (FFIC-01 and -02) are protected from reset windup by receiving an external feedback signal through the low selector #7 (or #12), which selects the lower of the FFIC output and the ramp signal.

Interlock #1 is also provided with "rotating sequencer" logic, which serves to equalize run times between machines and protects the same machine from being started and stopped frequently. A simple approximation of these goals is achieved if the machine that operated the longest is stopped and the one that was idle the longest is started.

If only one of the compressors is variable-speed, the PIC-01 output signal can be used in a split-range manner. For example, if there were five compressors of equal capacity, switches would be set to start an additional constant-speed unit as the output signal rises above 20, 40, 60, and 80% of its full range. The speed setting of the one variable-speed compressor is obtained by subtracting from the PIC output the sum of the flows developed by the constant-speed machines and multiplying the remainder by 5. The gain of 5 is the result of the capacity ratio between that of the individual compressor and the total.

Multi-Inlet Compressor Control

These compressors most often are refrigeration compressors. The main problem here is to keep the process temperatures, which are controlled by the evaporators, within tolerable limits. This is achieved by controlling the vapor temperature (by quenching) in a feedforward manner, based upon the amount of recycle flow or recycle valve position. A typical control scheme to control a three-inlet compressor is shown in Figure 8.15nn.

Installation

A check valve in the discharge line as close to the compressor as possible will protect it from surges. On motor-driven compressors, it is helpful to close the suction valve during starting to prevent overload of the motor. After the unit is operating, it should be brought to stable operating range as soon as possible to prevent overheating.

The recycle valve control should be fast opening and slow closing to come out of surge quickly and then stabilize the flow. The transmitter ranges should be such that the recycle valve is able to open in a reasonable amount of time with reasonable proportional and integral control constants of the antisurge controller. The importance of proper controller tuning cannot be underestimated. Typical settings are PB = 50%, I = 1–3 sec/repeat.

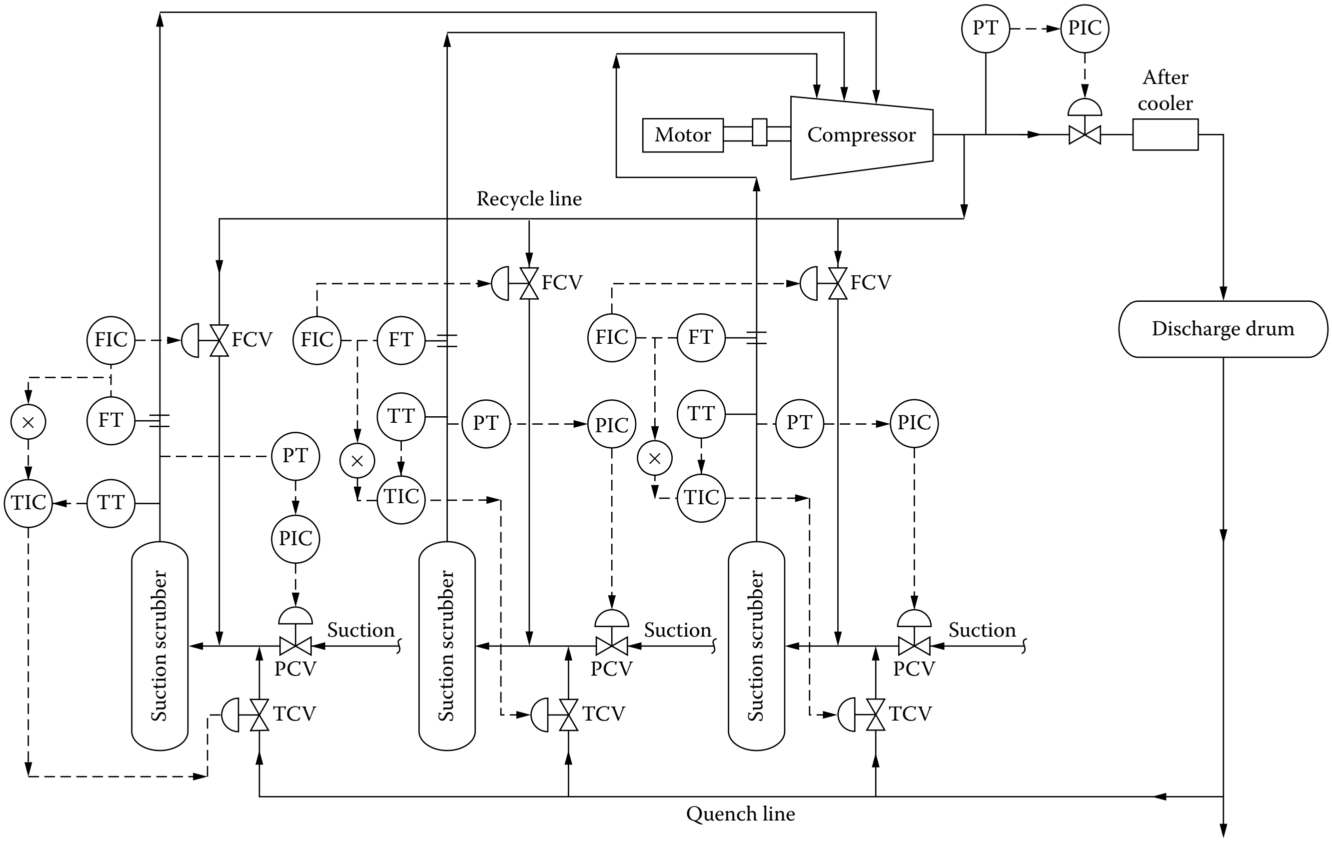


FIG. 8.15nn
Control strategy for a multiple inlet compressor.

Lube and Seal Systems A typical lube oil system is shown in Figure 8.15oo. Dual pumps are provided to ensure the uninterrupted flow of oil to the compressor bearings and seals.

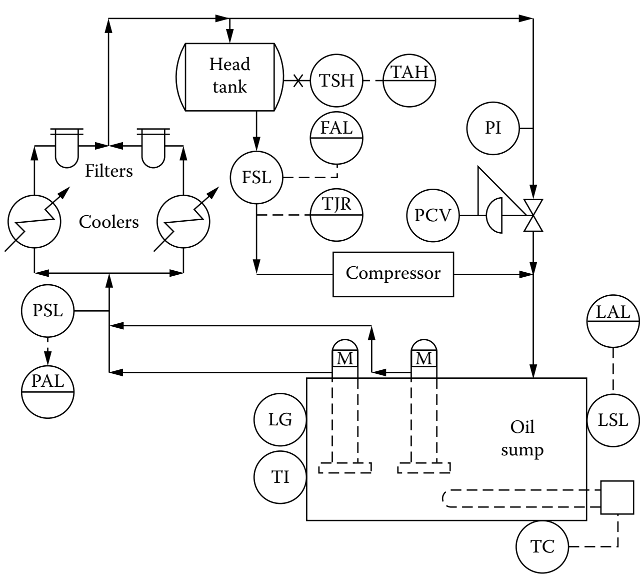


FIG. 8.15oo
Lube oil controls for the compressor's bearings and seals.

Panel alarms on low oil level, low oil pressure (or flow), and high oil temperature are provided. A head tank provides oil for coasting down in case of a power failure.

The design of these systems is critical, because failure of the oil supply could mean a shutdown of the entire process. In cases in which the process gas cannot be allowed to contact the oil, an inert gas seal system may be used. This is shown in Figure 8.15pp for a centrifugal compressor with balanced seals.

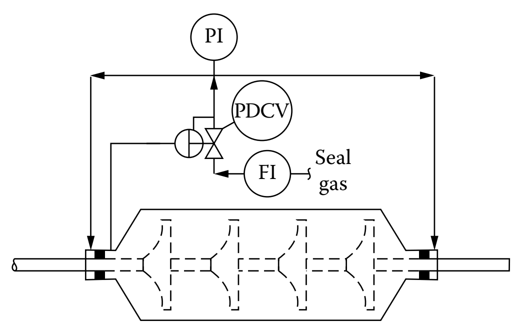


FIG. 8.15pp
Balanced seal controls on a centrifugal compressor.

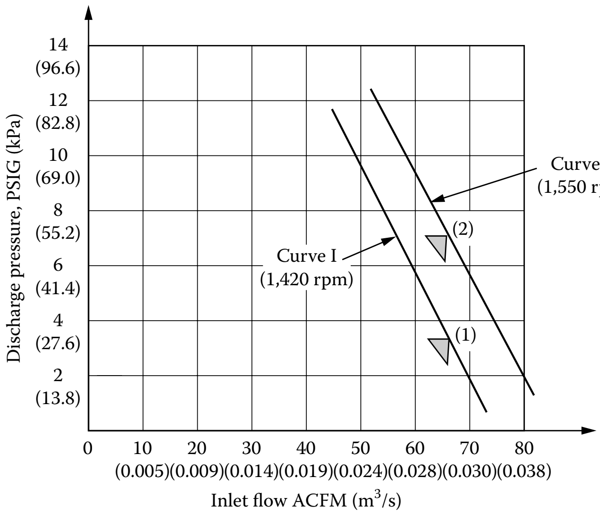


FIG. 8.15qq
The rotary compressor is a positive displacement machine. At constant inlet flow, the discharge pressure rises if the speed is increased, while at constant speed, the flow drops as the discharge pressure rises.

ROTARY COMPRESSORS, BLOWERS

The rotary compressor is essentially a constant-displacement, variable-discharge pressure machine. Common designs include the helical-screw, the lobe, the sliding vane, and the liquid ring types. The characteristic curves for a lobe-type unit are shown in Figure 8.15qq. As shown by curves I and II, the inlet flow varies linearly with the speed of this positive-displacement machine.

The small decrease in capacity at constant speed with an increase in pressure is a result of gas slippage through impeller clearances. It is necessary to compensate for this by small speed adjustments as the discharge pressure varies. For example, when the compressor is operating at point 1, it delivers the design volume of 66 ACFM (1.9 m^3/m) and 3.5 PSIG (24 kPa). In order for the same flow to be maintained when the discharge pressure is 7 PSIG (48 kPa), the speed must be increased from 1420 rpm to 1500 rpm at point 2 by the flow controller in the discharge line.

In addition to speed variation, the discharge flow can also be adjusted by throttling the suction, the bypass, or the vent line from the rotary blower. Vent throttling of the excess gas is shown in Figure 8.15rr in a process where the discharge is throttled by a temperature controller. This can be a dryer application, where the temperature of the outlet gas is controlled to prevent product degradation and to provide the proper product dryness. In other systems, instead of venting, the gas is returned to the suction of the blower under pressure control.

An important application of the liquid ring rotary compressor is in vacuum service. The suction pressure is often the independent variable and is controlled by bleeding gas

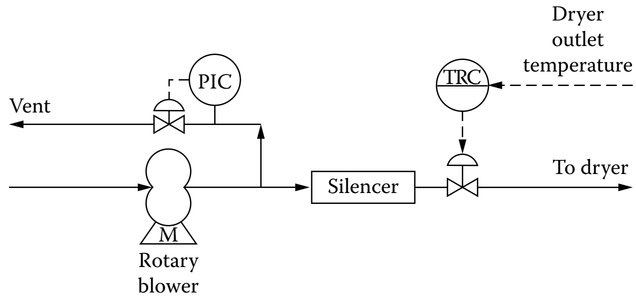


FIG. 8.15rr
The blower discharge pressure can be controlled by venting the excess gas that is not required by the process.

into the suction on pressure control. This is shown in Figure 8.15ss, where suction pressure control is used on a rotary filter, maintaining the proper drainage of liquor from the cake on the drum.

The optimization of rotary compressors will be discussed later, together with that of reciprocating compressors.

RECIPROCATING COMPRESSORS

The reciprocating compressor is a constant-volume, variable-discharge pressure machine. A typical compressor curve is shown in Figure 8.15tt, for constant-speed operation. The curve shows no variation in volumetric efficiency in the design pressure range, which may vary by 8 PSIG (53.6 kPa) from unloaded to fully loaded.

The volumetric inefficiency is a result of the clearance between piston end and cylinder end on the discharge stroke. The gas that is not discharged reexpands on the suction stroke, thus reducing the intake volume.

The relationship of speed to capacity is a direct ratio, because the compressor is a displacement-type machine. The

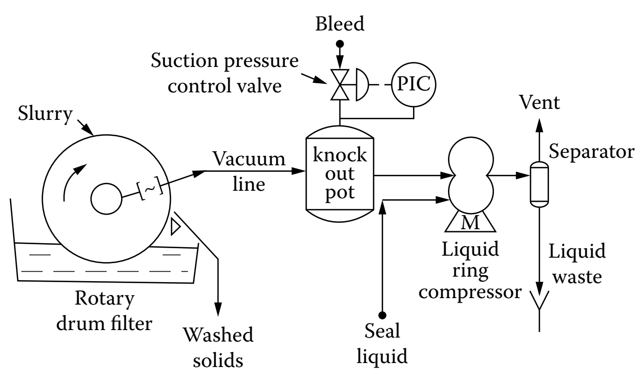


FIG. 8.15ss
The suction pressure control of a liquid ring-type rotary compressor is illustrated here.

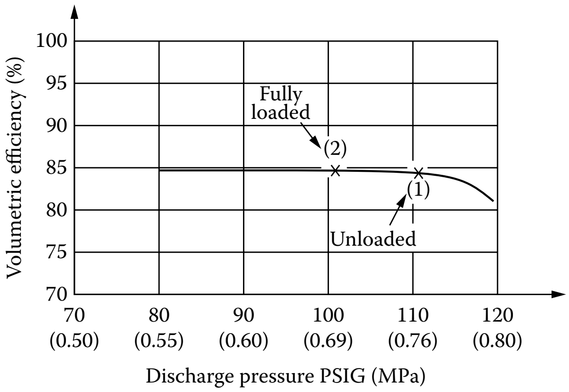


FIG. 8.15tt
Reciprocating compressor curve.

typical normal turndown with gasoline or diesel engine drivers is 50% of maximum speed, in order to maintain the torque within acceptable limits.

On/Off Control

For intermittent demand, where the compressor would waste power if run continuously, the capacity can be controlled by starting and stopping the motor. This can be done manually or by the use of pressure switches. Typical switch settings are on at 140 PSIG (1 MPa), off at 175 PSIG (1.2 MPa). This type of control would suffice for processes in which the continuous usage is less than 50% of capacity, as shown in Figure 8.15uu, where an air mix blender uses a rapid series of high-pressure air blasts when the mixer becomes full. The high-pressure air for this purpose is stored in the receiver.

Constant-Speed Unloading

In this type of control, the driver operates continuously, at constant speed, and one varies the capacity in discrete steps by holding suction valves open on the discharge stroke or opening clearance pockets in the cylinder. The most common schemes are three- and five-step unloading techniques. The larger number of steps saves horsepower because it more closely matches the compressor output to the demand.

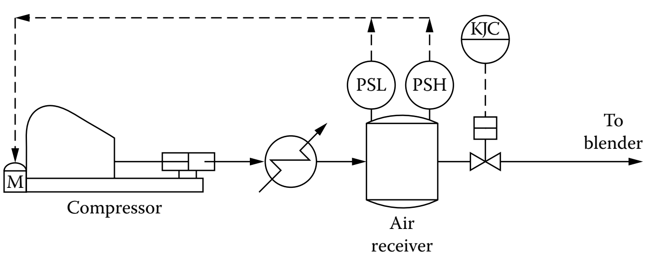


FIG. 8.15uu
On/off control of a reciprocating compressor.

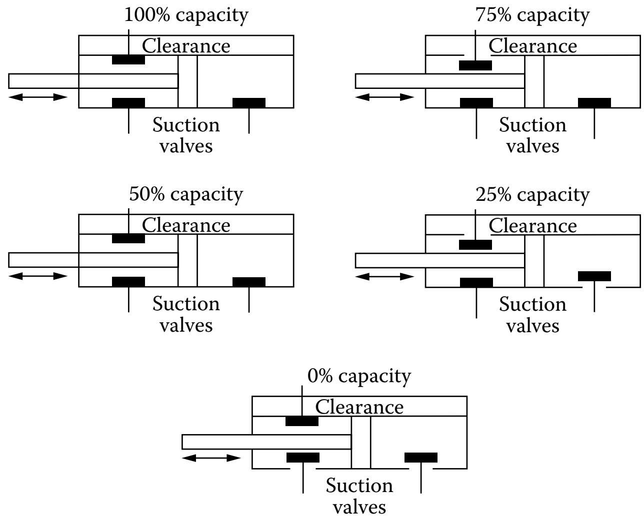


FIG. 8.15vv
The five steps in a constant-speed, positive displacement compressor with five-step unloading.

In three-step unloading, capacity increments are 100, 50, or 0% of maximum flow. This method of unloading is accomplished by the use of valve unloading in the double-acting piston. At 100%, both suction valves are closed during the discharge stroke. At 50%, one suction valve is open on the discharge stroke, wasting half the capacity of the machine. At 0%, both suction valves are held open on the discharge stroke, wasting total machine capacity.

For five-step unloading, a clearance pocket is used in addition to suction valve control. The capacity can be 100, 75, 50, 25, or 0% of maximum flow. This is shown in Figure 8.15vv. At 100%, both suction valves and the clearance pocket are closed. At 75%, only the clearance pocket is open. At 50%, only one suction valve is open on the discharge stroke. At 25%, one suction valve and the clearance pocket are open. At 0%, both suction valves are opened during the discharge stroke.

The use of step unloading is most common when the driver is inherently a constant-speed machine (Figure 8.15ww),

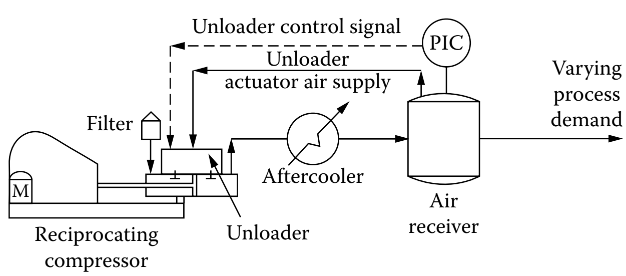


FIG. 8.15ww
Constant-speed capacity control of a reciprocating compressor, provided with pneumatic unloading.

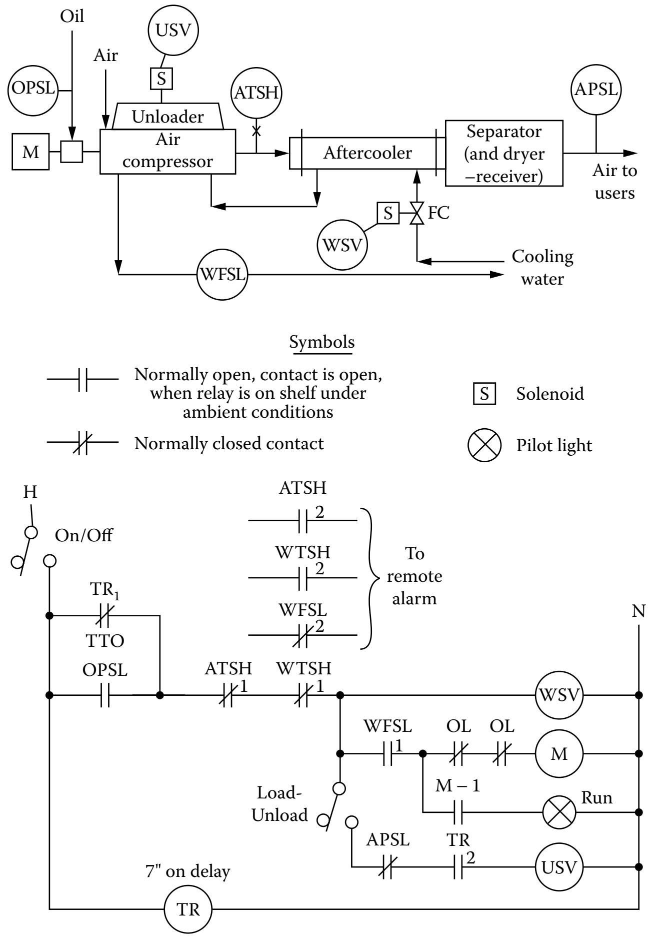


FIG. 8.15xx
The layout and control logic of a typical stand-alone air compressor.[2]

such as an electric motor. The pressure controller signal from the air receiver operates a solenoid valve in the unloader mechanism. The action of the solenoid valve directs the power air to lift the suction valve or to open the clearance port, or both.

For three-step unloading, two pressure switches can be used. The first switch loads the compressor to 50% if the pressure falls slightly below its design level, and the second switch loads the compressor to 100% if the pressure falls below the setting of the first switch.

For five-step unloading, a pressure controller is usually substituted for the four pressure switches otherwise required, and the range between unloading steps is reduced to not more than 2 PSI (13.8 kPa) deviation from the design level, keeping the minimum pressure within 8 PSI (55 kPa) of design. In cases in which exact pressure conditions must be met, a throttling valve is installed, which bypasses the gas from the discharge to the suction of the compressor. This device smoothes out pressure fluctuations and, in some cases, eliminates the need for a gas receiver in this service. This can prove economical in high-pressure services above 500 PSIG (3.45 MPa), in which vessel costs become significant.

The Stand-Alone Air Compressor

Abbreviations and Terminology

APSL Low air pressure switch with contact that closes as pressure drops below setting.

ATA Air temperature alarm, actuated on high temperature.

ATSH High air temperature switch with two contacts.

ATSH-1 First contact of ATSH is normally closed; it opens on high temperature.

ATSH-2 Second contact of ATSH is normally open; it closes on high temperature.

H Fused, 120 V, hot power supply of the stand-alone compressor control circuit.

HO Fused, 120 V, hot power supply of the integrated remote controls.

M Motor.

N Neutral wire of the stand-alone compressor control circuit.

NO Neutral wire of the integrated remote control system.

OFF DELAY This type of time delay introduces a delaying action when the relay is de-energized. A 2 min off-delay means that for the first 2 min period after de-energization, the relay contacts will remain as if the relay was energized. When the delay expires, the contacts will switch to their de-energized state.

ON DELAY This type of time delay introduces a delaying action when the relay is energized. A 3 sec on-delay means that for the first 3 sec after it is energized, the relay contacts will remain as if the relay was still de-energized. When the delay expires, the contacts will switch to their energized state.

TR Time delay relay. The delay time setting is marked next to it.

TTC Time to close action. This contact is open, then the TR relay is de-energized. This contact stays open when the TR relay is de-energized until the delay time setting expires. Then it closes.

TTO Time to open action. This contact is closed when the TR relay is de-energized. This contact stays closed when the TR relay is energized until the delay time setting expires. Then it opens.

USV Unloading solenoid. The compressor is loaded when this solenoid is energized.

WFA Water flow alarm. This alarm is energized if the flow drops below the minimum allowable.

WFSL Low water flow switch, with two contacts.

WFSL-1 This contact opens on low flow.

WFSL-2 This contact closes on low flow.

WSV Cooling water solenoid valve, opens when energized.

WTA Water temperature alarm, energized on high temperature.
WTSH High water temperature switch, with two contacts.
WTSH-1 This contact opens on high temperature.
WTSH-2 This contact closes on high temperature.

Operation A typical air compressor, together with the controls that are normally provided by its manufacturer, is shown in Figure 8.15xx.

Such a stand-alone compressor usually operates as follows: When the operator turns the control switch to "On," this will energize the time delay (TR) and will open the water solenoid valve (WSV) if neither the air temperature (ATSH) nor the cooling water temperature (WTSH) is high.

After 7 sec, TR-1 opens. Assuming that this time delay was sufficient for the oil pressure to build up, the opening of TR-1 will have no effect, because OPSL will have closed in the meantime.

If the opening of WSV resulted in a cooling water flow greater than the minimum setting of the low water flow switch, then WFSL closes and the compressor motor (M) is started. Whenever the motor is on, the associated "Run" pilot light is energized. This signals to the operator that oil (OPSL), water (WTSH, WFSL), and air (ATSH) conditions are all acceptable and, therefore, the unit can be loaded.

When the operator turns the other control switch to "Load," the TR-2 contact is already closed because the 7 sec have passed. Therefore, the APSL contact will determine the status of the machine. If the demand for air at the users is high, the pressure at APSL will drop, which in turn will cause the APSL contact to close and the compressor to load. As a result of loading the compressor, the pressure will rise until it exceeds the control gap of APSL, causing its contact to reopen and the machine to unload.

The compressor continues to load and unload automatically as a function of plant demand. As the demand rises, the loaded portion of the cycle will also rise. Once the load reaches the full capacity of the compressor, the unit stops cycling and stays in the loaded state continuously. If the demand rises beyond the capacity of this compressor, it is necessary to start another one. The following paragraphs describe how this is done automatically, requiring no operator participation.

If at any time during operation the cooling water flow (WFSL) drops too low, the motor will stop. If either water or air temperature rises to a high valve, this condition not only will stop the motor but will also close the water solenoid valve (WSV).

The above three conditions (WFSL, WTSH, ATSH) will also initiate remote alarms, as shown in Figure 8.15xx, to advise the operator of the possible need for maintenance. If the oil pressure drops below the setting of OPSL, it will also cause the stoppage of the motor and the closure of WSV, but after such an occurrence, the compressor will not be allowed to restart automatically when the oil pressure returns to normal. In order to restart the machine, the operator will have to go out to the unit and reset the system by turning the control switch to "Off" and then to "On" again to repeat the complete start-up procedure.

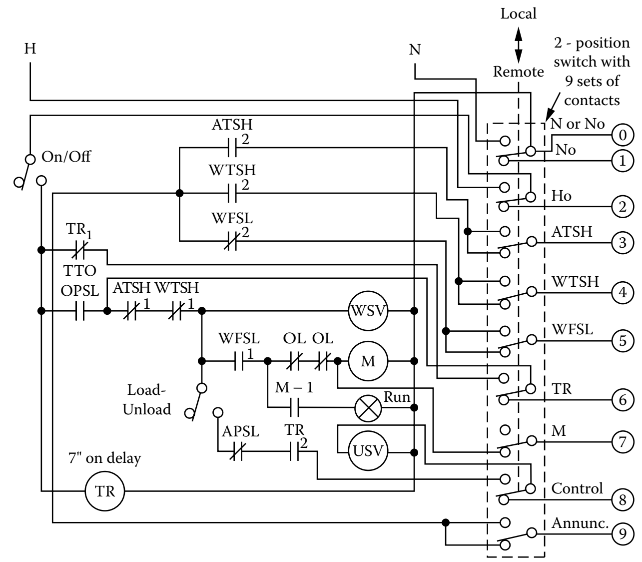


FIG. 8.15yy
The local/remote switch is wired to allow stand-alone or integrated compressor operation.[2]

Local/Remote Switch The first step in integrating several compressors into a single system is the addition of a local/remote switch at each machine. As shown in Figure 8.15yy, when this switch is turned to "Local," the compressor operates in the stand-alone mode, as was described in connection with Figure 8.15xx. When this switch is turned to "Remote," the compressor becomes a part of the integrated plantwide system, consisting of several compressors.

As shown in Figure 8.15yy, this two-position switch has nine sets of contacts and is mounted near the compressor. It can be installed in a few hours, and once installed, the compressor can again be operated in the "Local" mode. Only ten wires need to be run from each compressor to the remote controls. These ten wires serve the following functions:

#0 The working neutral, N in local, No in the remote mode of operation
#1 The common neutral (No) of the integrated controls
#2 The common hot (Ho) of the integrated controls
#3 The high air temperature (ATSH) alarm
#4 The high water temperature (WTSH) alarm
#5 The low water flow (WFSL) alarm
#6 The 7 sec time delay (TR) of the integrated controls
#7 The motor (M) status indication

#8 The load/unload control signal from the remote system
#9 The common hot for the remote annunciator (Annunc.)

When integrating several compressors into a single system, it is advisable to number these ten wires in a consistent manner, such as:

Compressor #1	Wire #10 to #19
Compressor #2	Wire #20 to #29
Compressor #3	Wire #30 to #39, and so on

In this system, the first digit of the wire number indicates the compressor, and the second digit describes the function of the wire. Immediately knowing, for example, that wire number 45 comes from compressor #4 and serves to signal a low water flow condition on that machine simplifies check-out and start-up.

Annunciator Of the ten wires from each of the compressors, six are used for remote alarming. Figure 8.15zz shows a remote annunciator for two compressors. This alarm system can be expanded to serve any number of compressors.

The position of the local/remote switch in Figure 8.15yy does not affect the operation of the annunciator. It provides the following remote indications for each compressor:

"Run" light
High air temperature light
High water temperature light
Low water flow light
Audible alarm bell with silencer
Alarm reset buttons

The only time these circuits are deactivated is when the associated compressor is off.

Lead-Lag Selector As plants grow, their compressed air requirements also tend to increase. As a result of such evolutionary growth, many existing plants are served by several uncoordinated compressor stations. When, because of space limitations, the new compressors are installed in different locations, the manual operation of such systems becomes not only inefficient but also unsafe.

The steps involved in integrating such stand-alone compressor stations into an automatically operating, load-following single system are described here. In such integrated systems, the identity of the "lead" and "lag" compressors, or the ones requiring maintenance ("Off"), can all be quickly and conveniently altered, while the system continues to efficiently meet the total demand for air. Thus, air supply shortage or interruption is eliminated, together with the need for continuous operator's attention.

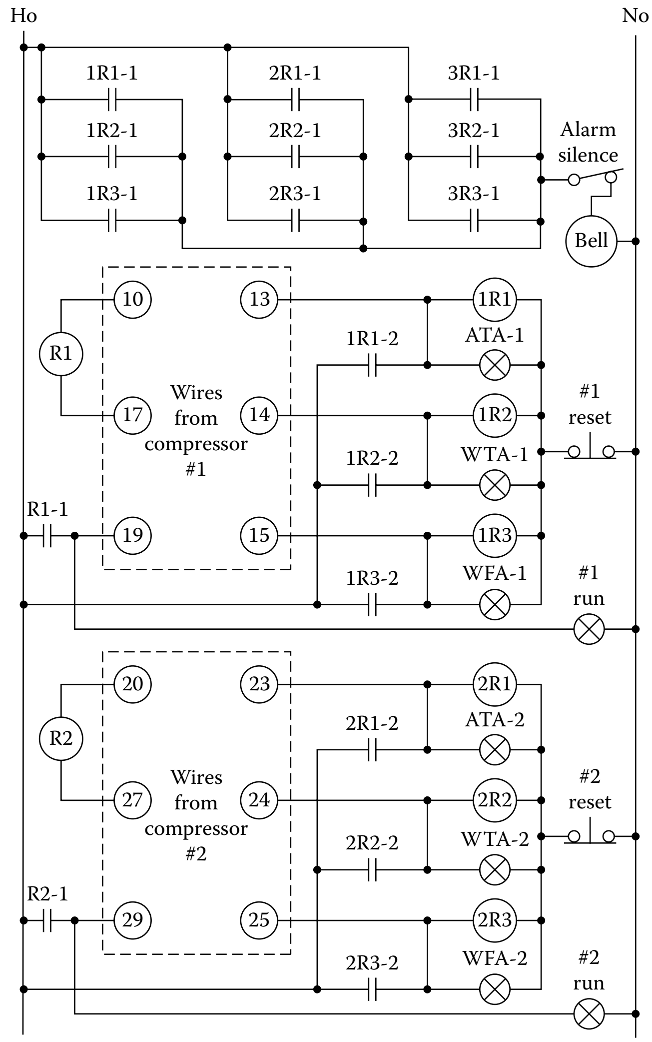


FIG. 8.15zz
The wiring requirements for an annunciator serving two compressors.[2]

When it is desirable to combine two compressors into an integrated lead-lag station, all that needs to be added are two pressure switches, as shown in Figure 8.15aaa.

APSL-A is the lead and APSL-B is the lag control pressure switch. To maintain a 75 psig air supply at the individual users, these normally closed pressure switches can be set as shown in Table 8.15bbb.

With the above settings, the system performance will be as shown in Table 8.15ccc.

When two stand-alone compressors are integrated into such a lead-lag system, only five wires need to be brought from each compressor to the lead-lag switch, shown in Figure 8.15ddd. Turning this single switch automatically reverses the lead-lag relationship between the two compressors. In addition to the two-position lead-lag switch, running lights are also provided; they show whether either or both compressors are unloaded (standby) or loaded.

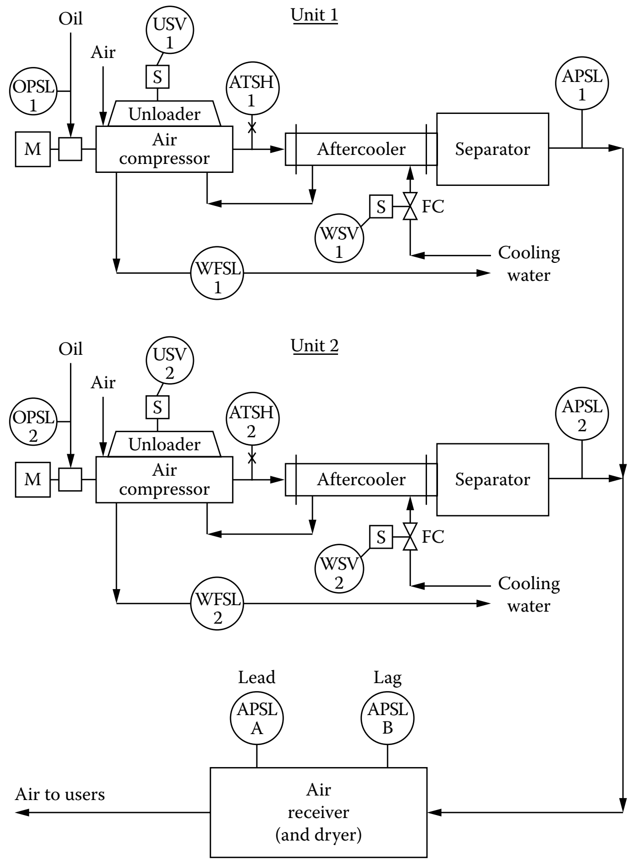


FIG. 8.15aaa
By adding two pressure switches (APSL-A and APSL-B), two compressors can be integrated into a single system.[2]

TABLE 8.15bbb
Set Pressures and Differential Gaps for Integrating the Controls of Two Compressors

Pressure Control Switch	*Pressure Below which Switch Will Close (psig)*	*Differential Gap (psi)*
ASPL-A	90	10
ASPL-B	85	10

As the demand for air increases, the lead machine will load and unload to meet that demand. If the lead machine is continuously loaded and the demand is still rising, the lag compressor will be started automatically. The interlocks provided are as follows:

The on-time delays 2TR and 3TR in Figure 8.15ddd are provided for stabilizing purposes only. They guarantee that a system configuration (or reconfiguration) will be recognized only if it is maintained for at least 3 sec. Responses to quick changes are thus eliminated.

The on-time delays ITR and 4TR are provided to give time for the oil pressure to build up in the system. If after 7 sec the oil pressure is not yet established, and therefore OPSL in Figure 8.15xx is still open, the contacts ITR-1 and 4TR-1 in Figure 8.15ddd will open and the corresponding compressor will be stopped.

The off-time delay 5TR guarantees that the lag machine will not be cycled on and off too frequently. Once started, the lag compressor will not be turned off (but will be kept on standby) until the off delay of 2 min has passed.

Off Selector Three stand-alone compressors can be combined into an integrated load-following single system, by the

TABLE 8.15ccc
System Performance when ASPL-A Is Set to Close at 90 psig, APSL-B Is Set to Close at 85 psig, and the Differential Gap Is Set at 10 psid

Pressure in Air Receiver (psig)	*APSL-A Contact*	*APSL-B Contact*	*Lead Compressor*	*Lag Compressor*
Over 100	Open	Open	Off	Off
100	Open	Open	Off	Off
95	Open	Open	Off	Off
90	Closed	Open	On	Off
85	Closed	Closed	On	On
80	Closed	Closed	On	On
85	Closed	Closed	On	On
90	Closed	Closed	On	On
95	Closed	Open	On	Off
100	Open	Open	Off	Off

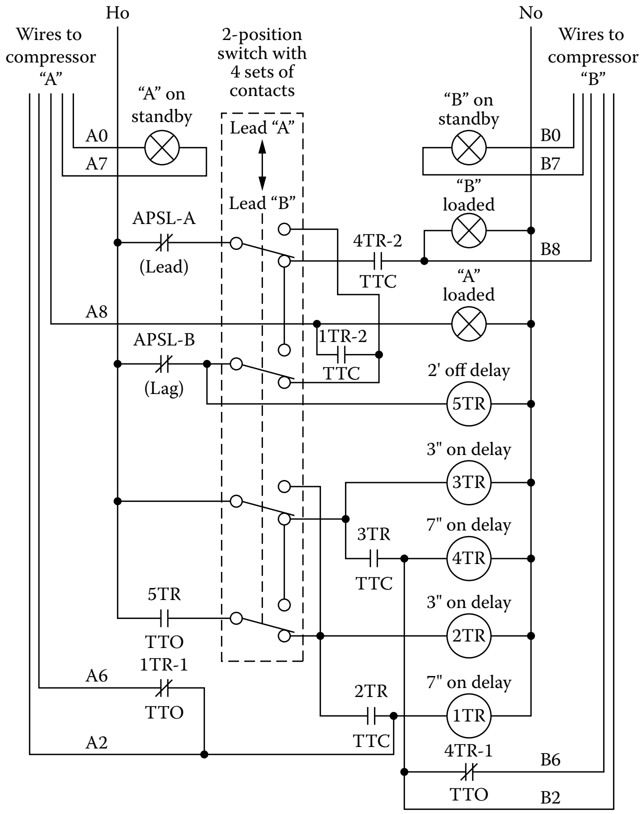


FIG. 8.15ddd
The wiring of a lead-lag selector switch that can automatically reverse the lead-lag relationship between two compressors.[2]

addition of a three-position "off" selector switch, illustrated in Figure 8.15eee. With the addition of the off-selector, any of the compressors can be selected for load, lag, or off duties, just by turning these conveniently located switches.

Depending on the position of the off selector, the identities of Compressors A and B in Figure 8.15ddd will be as follows:

Selected Off	*Compressor A*	*Compressor B*
1	3	2
2	1	3
3	1	2

For an integrated three-compressor system, only 30 wires need to be run if the remote annunciator is included. Without remote alarms, only 15 wires are needed (5 per compressor). The front face of a remote control cabinet of a three-compressor lead-lag system is illustrated in Figure 8.15fff.

Large Systems From the building blocks discussed in the previous paragraphs, an integrated remote controller can be configured for every compressor combination. Figure 8.15ggg

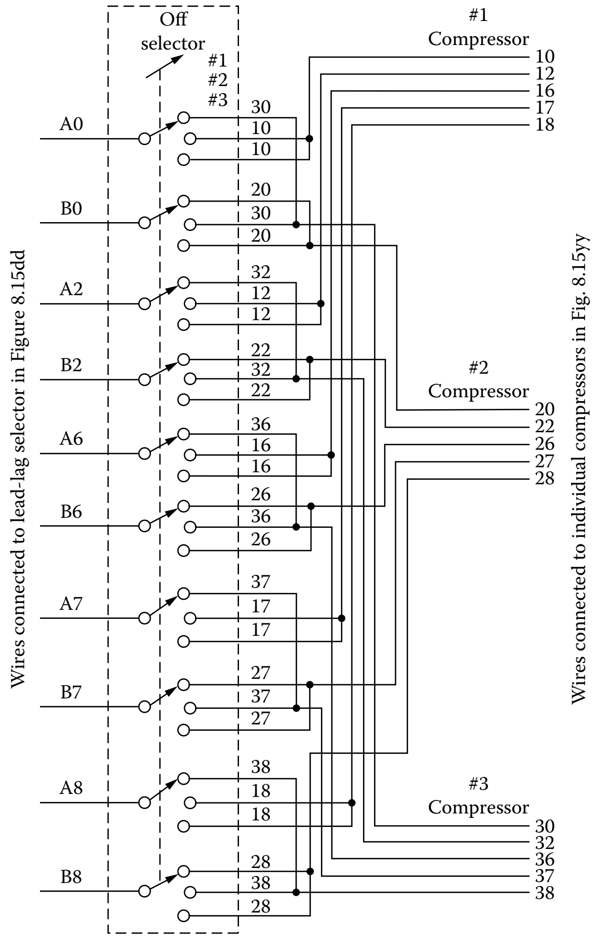


FIG. 8.15eee
The wiring of an "off" selector switch that can be used to identify the common spare compressor, or the compressor that is undergoing maintenance from among a set of three compressors.[2]

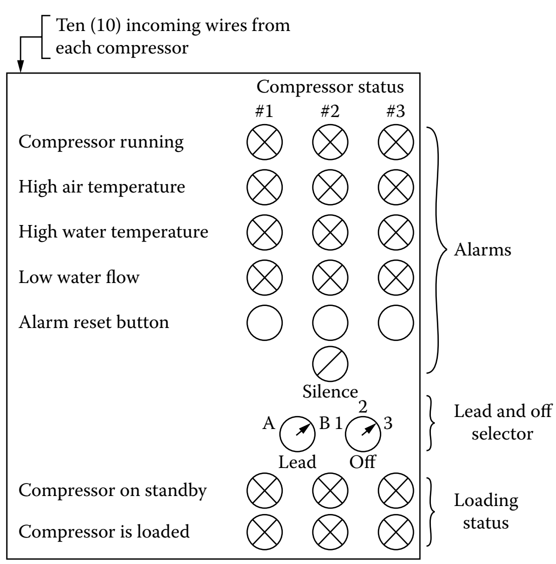


FIG. 8.15fff
The faceplate of the control panel that can be used to operate an integrated three-compressor system.[2]

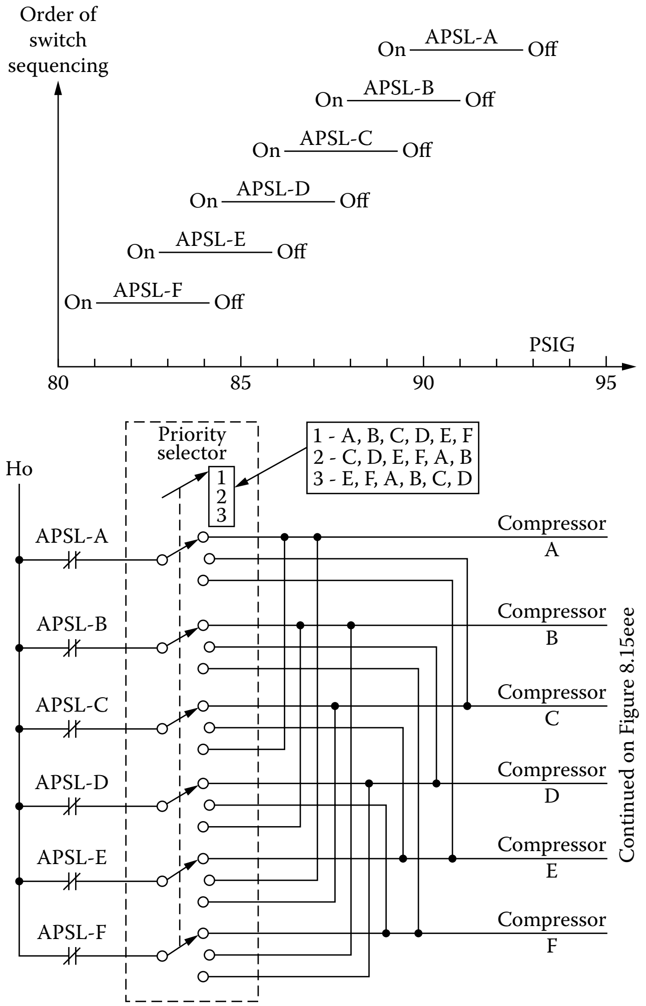


FIG. 8.15ggg
In large, multiple-compressor systems, a priority selector can be used to change the order of priorities of the compressors within the system.[2]

illustrates how six stand-alone compressors might be integrated into a single load-following system.

As the system pressure drops below 90 psig, APSL-A closes, starting the first compressor. As the load rises, causing the system pressure to drop further, more switches will close, until at 80 PSIG all six switches will be closed and all six compressors will be running.

The "priority selector" shown in Figure 8.15ggg is provided for the convenient reconfiguration of compressor sequencing. When the selector is in position #1, as the load increases, the compressors will be started in the A, B, C, D, E, F order. When switched to position #2, the sequence becomes C, D, E, F, A, B. In position #3, the sequence is E, F, A, B, C, D.

If a system consists of nine compressors, and full flexibility in integrated remote control is desired, the faceplate of a control cabinet might look like Figure 8.15hhh. With this system, the operator can set the lead, lag, or off status for all

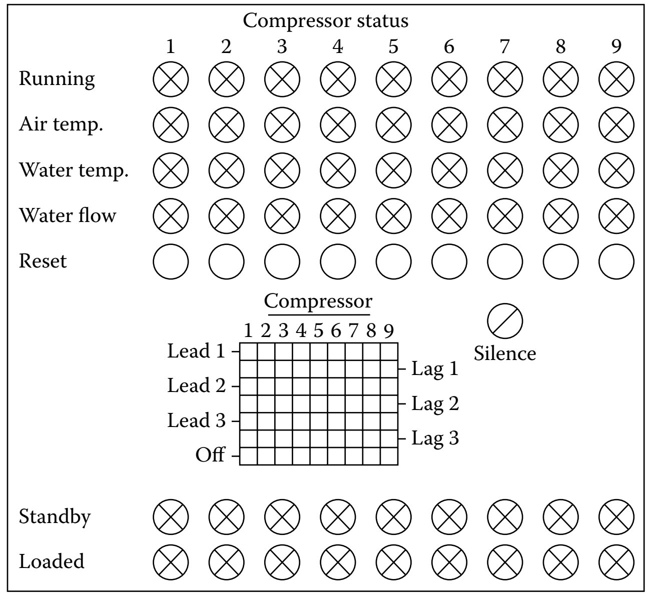


FIG. 8.15hhh
A control cabinet can provide full flexibility in remote control for an integrated nine-compressor system. Such a mechanically interlocked pushbutton station or its microprocessor controlled digital equivalent can be so designed that only one button can be pressed in each column or row, except in the bottom one. In the bottom row any or all buttons can be simultaneously pressed.[2]

nine compressors and can also select the order in which they are to come on.

Wired or Digital Bus From the previous discussion it can be seen that the building-block approach to compressor control system design is very flexible. Any number of stand-alone compressors can be integrated into automatic load-following control systems, with complete flexibility for priority, lead-lag, or off selection. The logic blocks described can be implemented either in hardware or in software. All wires and terminals can be prenumbered to minimize installation errors.

The potential benefits of compressor control system integration are listed below:

1. Unattended, automatic operation relieves operators for other tasks.
2. Automatic load-following eliminates the possibility of accidents caused by the loss of the air supply.
3. Because the supply and the demand are continuously and automatically matched, the energy cost of operation is minimized.

If priority or lead-lag switching is under computer control, this would also make the digital implementation of the system more desirable. On the other hand, if a dedicated, hard-wired system is preferred, because it is more familiar

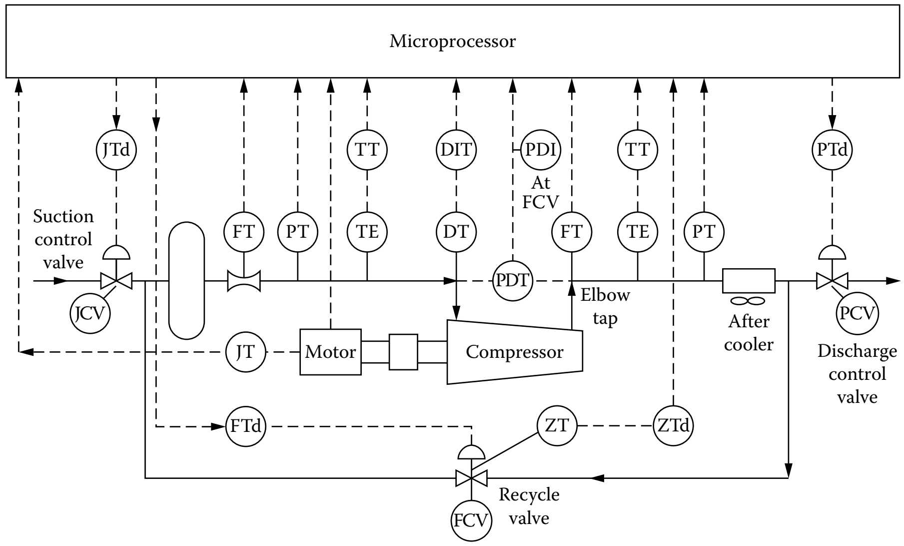


FIG. 8.15iii
A digital unit controller that is connected to the compressor I/O by a digital bus is well suited for both compressor optimization and for integrating the compressor controls into the total control system of the plant.

to the average operator, hardware implementation can be the proper choice.

CONCLUSIONS

As was shown in Figures 8.15g and 8.15j, the energy cost of operating a single centrifugal compressor at 60% average loading can be cut in half if optimized variable-speed control is applied. In the case of multiple compressor systems, similar savings can be obtained by the use of optimized load-following and supply-demand matching control strategies.

The full automation of compressor stations — including automatic start-up and shutdown — not only will reduce operating cost but will also increase operating safety as human errors are eliminated. Figure 8.15iii illustrates the use of a microprocessor-based digital unit controller. Such units can be provided with all the features that were discussed in this section in the form of software algorithms. Such digital systems provide flexibility by allowing changes in validity checks, alteration of limit stops, or reconfiguration of control loops.

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