SECTION 3

Description of Surge

3-1 SURGE VERSUS STALL

Surge is a dynamic instability that occurs in dynamic compressors. Surge can also occur in axial and centrifugal pumps and blowers, but the occurrence is less frequent and the damage less severe. Pump surge seldom occurs except infrequently during cavitation and two-phase flow (Ref. 15). Blower surge does occur but does not damage the blower internals unless the pressure rise exceeds 2 psi and the size exceeds 150 BHP (brake horsepower) (Ref. 22). Surge does not occur in positive displacement compressors and pumps.

Stall is another instability that occurs in dynamic compressors and is sometimes confused with surge. The consequences of stall are usually less severe. Pump stall has caused extensive vibration in the piping of boiler feedwater systems (Ref. 15). Stall will be described in enough detail to distinguish it from surge and to understand how it can develop into surge.

As the flow through a compressor is reduced, a point is reached where the flow pattern becomes unstable. If the flow oscillates in localized regions around the rotor, the instability is called *stall*. For axial compressors these regions of unstable forward flow can extend over just a few blades or up to 180 degrees around the annulus in the compressor. In rotating stall, the region of unstable forward flow rotates around the annulus. The average flow across the annulus is still positive. The frequency of the localized flow oscillations ranges from 50 to

100 hertz. The stall frequency is some fraction of the compressor speed. The stall frequency does not depend on the design of the piping system in which the compressor is installed. Stall can develop into a more global type of instability called *surge*. In surge the average flow across the annulus goes through large amplitude oscillations. The frequency of these oscillations ranges from 0.5 to 10 hertz. The surge frequency depends on the designs of both the compressor and piping system. The surge frequency for most industrial compressor installations is slightly less than 1 hertz.

Figure 3-1 shows a *compressor map* for a variable-speed centrifugal compressor. A compressor map is the single most important piece of information for describing surge. A compressor map shows the compressor characteristic curve for different operating conditions. Each curve traces the rise in discharge pressure developed by the compressor as the suction flow is varied for a given operating condition (such as speed). The operating condition is not limited to that shown by the few curves plotted but is continuously adjustable to intermediate values. The *X* axis is almost always volumetric flow in acfm at the stated suction temperature, pressure, and molecular weight. The *Y* axis is either the discharge pressure in psia, the ratio of discharge pressure to suction pressure (dimensionless), or the differential pressure rise from suction to discharge in psi. The compressor characteristic curves supplied by the compressor manufacturer cover the negative sloped region and end where the slope approaches zero. (A negative slope means the pressure will change in the opposite direction for a change in flow, and a zero slope means the pressure will not change at all for a change in flow.) If a curve is drawn through the point of zero slope for each characteristic curve, the region to the left of this line is where the instabilities of surge and stall occur. In the literature, this line is called either the stall line or the surge curve. In this text it will be always called the surge curve. The characteristic curve to the left of this curve is difficult to obtain, but its general shape resembles that shown in Figure 3-1. The negative flow portion of the curve can be found by supplying pressurized gas to the compressor discharge to force a steady negative flow. The positive sloped portion of the curve is obtained by smoothly connecting a third-order polynomial curve between the negative sloped negative flow and the positive flow portions of the curve.

Surge can be better understood by visualizing a block or throttle valve closing downstream of the compressor. As this valve closes, the suction flow decreases, and the operating point moves to the left along a characteristic curve. If the compressor speed or the guide vane position does not change, the operating point eventually passes by the point of zero slope on the characteristic curve. Just to

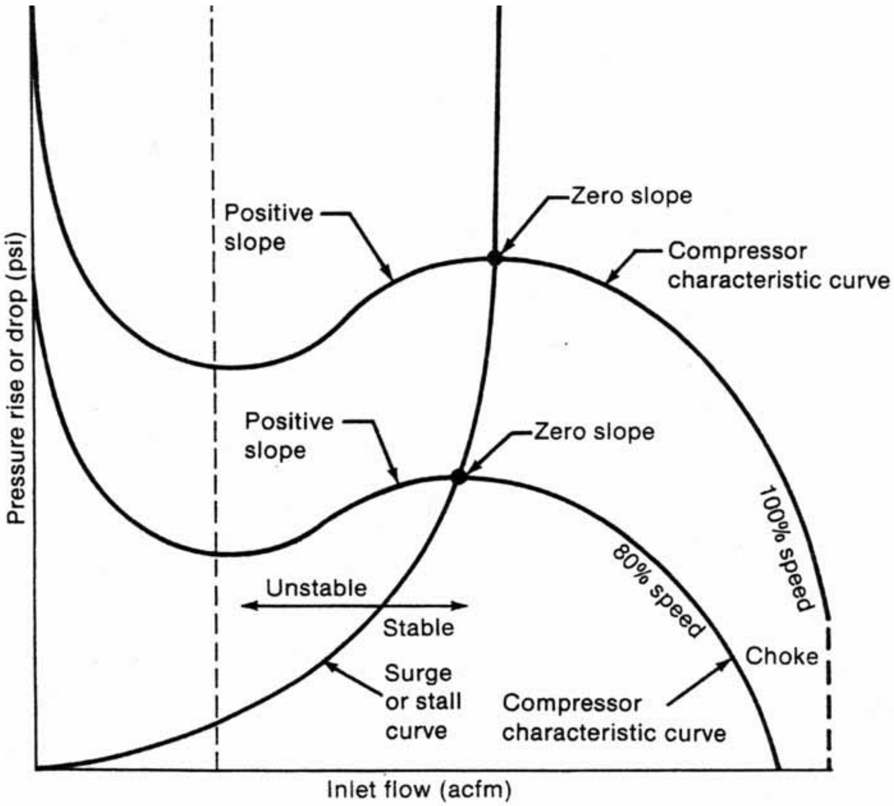

Figure 3-1 Compressor Map

the left of this point, which is the intersection of the surge curve with the characteristic curve, the pressure developed by the compressor is less than the pressure in the piping between the compressor discharge and the closing valve. The forward flow through the compressor stops and reverses direction. The gas flows from the discharge piping back through the compressor to the suction piping. The pressure in the discharge piping starts to drop. When this pressure is below the pressure developed by the compressor, forward flow starts again. If the gas in the discharge piping is still trapped by the closing block or throttle valve, the gas pressure builds up and the surge cycle is repeated. Note that the alternating pressure buildup and decay cycle does not require the flow to actually reverse direction but only to alternately decrease below or increase above the flow rate through the block or throttle valve. Only in severe surges does the flow through the compressor actually reverse direction.

Whether a compressor system is stable or unstable depends on how the operating point reacts to a disturbance. If the operating point returns to its initial value after a disturbance, the system is stable. If the deviation of the operating point from its initial value grows or continually oscillates after a disturbance, the system is unstable. A uniform growing deviation is classified as a static instability and an oscillating deviation of growing or constant amplitude is classified as a dynamic instability. Surge is a dynamic instability.

Key Concepts

- Surge and stall occur in dynamic compressors.
- Stall consists of localized flow oscillations around the rotor.
- Surge consists of total flow oscillations around the whole rotor.
- The surge curve is the zero slope points of the characteristic curves.

3-2 STATIC INSTABILITY

Static instability is a uniform (non-oscillatory) growing deviation. Static instability will occur when the slope of the compressor characteristic curve is greater than the slope of the demand load curve.

As previously mentioned, the demand load curve is a plot of how the pressure drop in the system (due to piping, equipment, and valve resistance) increases with flow. The general criterion is:
For static instability,

$$S_c > S_l \quad (3\text{-}1)$$

where:

S_c = slope of the compressor characteristic curve (psi/acfm)

S_l = slope of the demand load curve (psi/acfm)

Figure 3-2 illustrates how the criterion for static instability can be checked by drawing the demand load curve on a compressor map. The compressor map shows stability at operating point 1 but static instability at operating point 2 where the compressor curve slope is steeper than the load curve slope. Since the compressor curve and the load curve are both steady-state curves, this instability criterion can

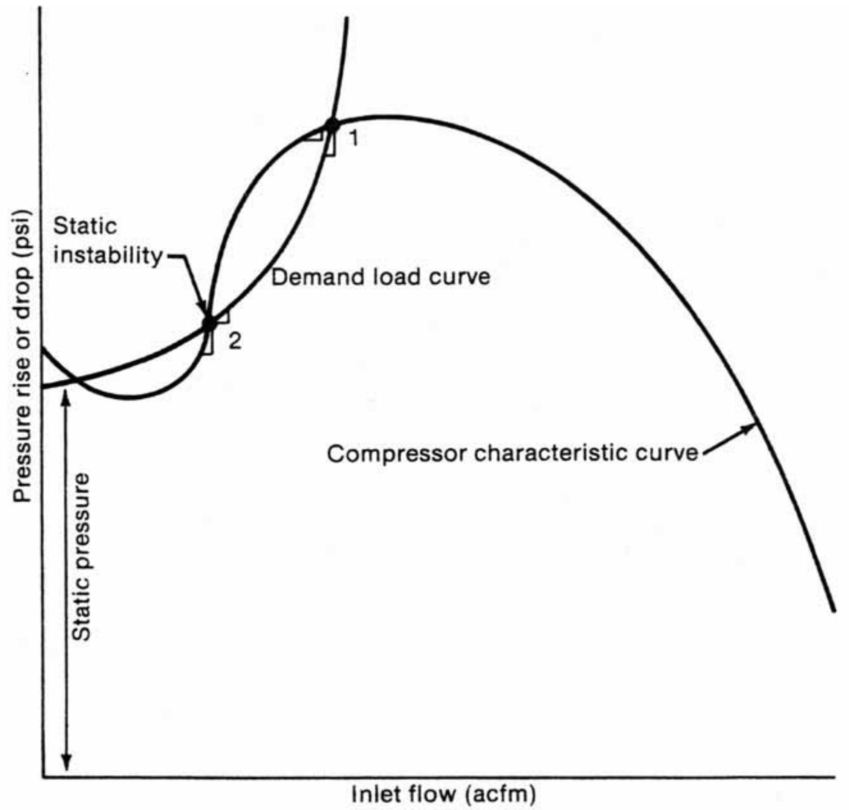

Figure 3-2 Static Instability Criterion

be checked at one steady state or a succession of steady states without knowledge of the system dynamics; hence the name, static instability. An electrical circuit analogy consists of a voltage or current source and a resistance. In this analogy, voltage represents pressure, current represents flow, and electrical resistance represents piping, equipment, and valve resistance. If the compressor characteristic curve is nearly horizontal, which is typical for centrifugal compressors, the circuit has a nonideal voltage source. If the compressor characteristic curve is nearly vertical, which is typical for axial compressors, the circuit has a nonideal current source. This circuit analogy has no dynamic components such as inductors or capacitors so the current flow does not vary with time if the system is stable. The circuit develops static instability if the slope of the source curve is greater than the slope of the resistance curve for voltage versus current. If the slope of the resistance is always positive (voltage drop always increases as current increases) and the slope of the source curve is always negative (voltage rise always decreases as current increases), the circuit is stable. If the slope of the source curve is positive and greater than the slope of the resistance curve, the following sequence of events occurs for any disturbance that causes an increase in voltage output:

(1) Increase in source voltage
(2) Increase in voltage (driving force) across the resistance
(3) Increase in current
(4) Increase in source voltage (sequence repeats)

The corresponding sequence for a compressor system is:

(1) Increase in compressor pressure
(2) Increase in pressure (driving force) across the load
(3) Increase in flow
(4) Increase in compressor pressure (sequence repeats)

A similar sequence and exponential growth of any positive deviation occurs in the open-loop response of temperature for runaway exothermic reactors and in the open-loop response of cell concentration for runaway biological reactors (Ref. 21). In a compressor system, static instability occurs when the load curve is nearly flat and intersects the negative sloped portion of the compressor curve. The load curve is nearly flat if a piece of equipment with pressure control such as a reactor, condensor, or absorber is connected to the discharge of a compressor by a short duct or pipe without a throttle valve. Figure 3-2 illustrates this case. The pressure at the starting point of the load curve at zero flow is the static pressure or set point of the pressure control loop for the downstream equipment. The slope of the load curve is proportional to the flow multiplied by twice the resistance of the duct or pipe. Since this duct or pipe is short, its resistance is low, and the slope of the load curve is low at low flow.

$$S_l = 2 \cdot K_l \cdot Q \quad (3\text{-}2)$$

where:

K_l = resistance of load (psi/(acfm · acfm))

Q = volumetric flow at suction conditions (acfm)

S_l = slope of the demand load curve (psi/acfm)

If the duct or pipe is long, the resistance coefficient K_l can still be small if the pipe or duct diameter is large with respect to the flow capacity of an individual compressor. This situation occurs when multiple parallel compressors are supplying a distribution header. The header diameter is selected based on the total flow capacity of all the compressors. The change in pressure drop in the header with a change in flow from an individual compressor is small, and thus the slope of the load curve on the compressor map is small.

Static instability occurs only if the compressor system can reach an operating point on the positive sloped portion of the compressor characteristic curve. Normally the flow drops precipitously when the operating point reaches the zero sloped portion of the compressor curve and surge oscillations develop, which is a dynamic instability.

Key Concepts

- Static instability is more likely to occur for parallel compressors.
- Dynamic instability usually occurs before static instability.

3-3 DYNAMIC INSTABILITY

Dynamic instability is characterized by growing oscillations. Dynamic instability can occur if either the discharge plenum volume or the compressor impeller speed is large enough to cause a system response parameter to exceed a minimum value and the slope of the compressor characteristic curve is positive. The general criterion is:

For dynamic instability,

$$B > B_m \quad \text{(3-3)}$$

$$S_c > 0 \quad \text{(3-4)}$$

$$B = \{N \cdot \sqrt{V_p/(A_c \cdot L_c)}\} \,/\, (2 \cdot a) \quad \text{(3-5)}$$

where:

a = speed of sound in the gas (ft/sec)

A_c = cross-sectional area of flow path in compressor (sq ft)

B = system dynamic response parameter (dimensionless)

B_m = minimum B for dynamic instability (typically 0.1 to 1.0)

L_c = length of flow path in compressor (ft)

N = compressor impeller speed (rev/sec)

V_p = volume of the plenum (see Figure 3-3) (cu ft)

Figure 3-3 shows a simple compressor system with the components identified for the dynamic instability criterion. The plenum is any enclosed volume of gas. In a compressor installation, the plenum may be a vessel or distribution header between the compressor and the block or throttle valve. If the valve is installed directly at the end of a section of pipe or duct connected to the compressor discharge, the volume of this section of pipe or duct can be used as an equivalent plenum volume without seriously degrading the accuracy of Equation 3-3. The compressor flow path length and area are small enough and the plenum volume is large enough in most industrial installations to satisfy the criterion for dynamic instability when the operating point crosses over from the negative to the positive sloped portion of the compressor characteristic curve. However, the oscillation amplitude stops growing due to the nonidealities and nonlinearities of compression in industrial applications. The resulting constant amplitude oscillation

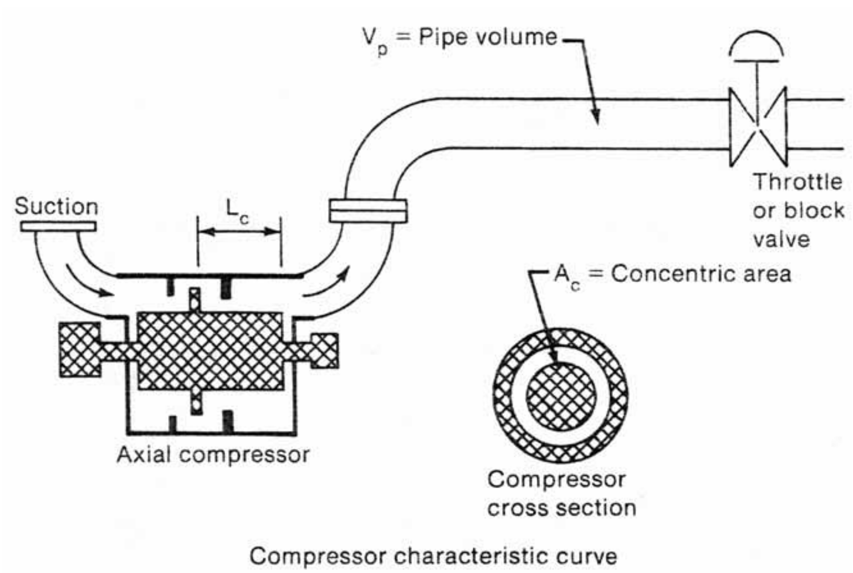

Figure 3-3 Simple Compressor System

is called a limit cycle. Surge is a dynamic instability that develops into a limit cycle. Stall is not a dynamic instability because the amplitude of the oscillations decreases (decays) with time. Thus the minimum B value (typically 0.1 to 1.0) is the boundary line between surge and stall. Surge occurs when B is above the minimum B, and stall occurs when B is below the minimum B. Also, the severity of surge is proportional to the magnitude of B. Note that this minimum B parameter depends on the compressor speed and the system dimensions and is independent of compressor manufacturer and model number. Thus, different compressors can be compared on a consistent basis.

Key Concepts

- Surge is a dynamic instability (sustained oscillations).
- The severity of surge increases with speed for a given installation.
- The severity of surge increases with volume for a given installation.

3-4 CHARACTERISTICS OF SURGE

Deep surge starts with a precipitous drop in flow. The flow will typically drop from its set point to its minimum (possibly negative) in less than 0.05 second. No other physical phenomenon can cause such a drop in flow.

Figure 3-4 shows the precipitous drop in flow measured by both a slow pneumatic transmitter and a fast electronic d/p flow transmitter. Since this precipitous drop in flow is unique to surge, it can be used as a trigger to actuate an interlock that will open the surge valves or start a surge counter. This drop can be detected by taking the derivative of the d/p signal if the flow measurement signal is not noisy and the transmitter is fast enough (see Section 6-4 for more detail on surge detection methods and Section 7-4 for the effect of transmitter time constant on surge detection).

The period of the surge oscillation (time between successive peaks in flow) is shorter than the control loop period unless fast instruments are used and the controller is tuned tight. The surge period is typically less than 2 seconds, while the loop period is usually greater than 4 seconds. Since the surge period is shorter than the loop period, by the time the control loop reacts to a particular portion of the surge cycle, that portion of the surge cycle is long gone. The corrective action by the control loop may become in phase with subsequent surge

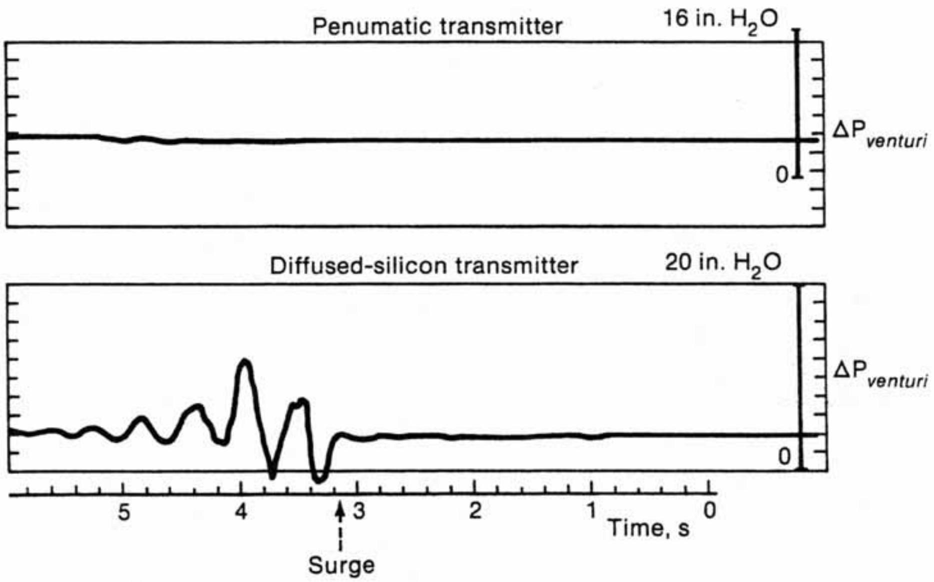

Figure 3-4 Precipitous Drop in Flow Measured by Two Transmitters *(Courtesy Compressor Controls Corporation)*

cycles and can accentuate the severity of the surge. If the surge period is less than one-fourth the loop period, the surge cycle can be considered as equivalent to uncontrollable noise (Ref. 21). The minimum loop period is approximately equal to 4 times the summation of the instrument time constants and dead times. (See Section 12-1 for more details on how to estimate the loop period for tuning the surge controller.) The minimum surge period that occurs when *B* is about equal to the minimum *B* for dynamic stability can be estimated by the following equation:

$$T_s = \{2\pi \cdot \sqrt{(L_c \cdot V_p)/A_v}\}/a \quad (3\text{-}6)$$

where:

a = speed of sound (ft/sec)

A_c = flow path cross-sectional area in compressor (sq ft)

L_c = flow path length in compressor (ft)

T_s = surge oscillation period (sec)

V_p = volume of the plenum (cu ft)

Equation 3-6 shows that the surge oscillation period increases as the plenum volume increases. Most industrial compressor installations have a check valve installed on the compressor discharge to prevent backflow during the opening of the surge valves or during the flow reversal caused by surge. This check valve is particularly important for fluidized bed reactor applications to prevent the backflow of catalyst or flammable mixtures. It is also important for parallel compressor applications to prevent pumping gas from one compressor back through another. The check valve will slam shut during the start of surge and will effectively reduce the plenum volume to that of the piping between the compressor and the check valve. Thus, surge oscillation periods are not large even when there is a long header or vessel between the compressor and the block or throttle valve.

The plenum pressure does not precipitously drop as does suction flow at the start of surge. Also the oscillation amplitude in pressure is usually less than that in flow during surge. Therefore surge counters or backup control systems that try to detect surge by pressure measurement are liable to experience false or missed surge counts or trips. This is particularly true if pneumatic pressure transmitters are used or if the pressure is noisy or swings with load.

The flow oscillation peaks correspond to the filling of the plenum, and the valleys correspond to the emptying of the plenum. The time duration of the peaks and the valleys is typically much longer than the time duration of the transition between the peaks and valleys. The time duration of the peaks and valleys depends on the piping system friction (resistance) and volume (capacitance), while the time duration of the transition depends on the fluid inertia (inductance). The system dynamic response parameter B is proportional to the ratio of the pressure driving force to the inertial impedance. B is representative of the capability to accelerate the gas. As parameter B increases, the amplitude of the oscillations becomes larger and the shape becomes more non-sinusoidal.

Figure 3-5a shows the path traced out by the operating point on a compressor map for severe surge where B is about 6 times the minimum B for dynamic stability as a throttle valve is closed downstream of the compressor. Figure 3-5b shows the oscillations in suction flow and discharge pressure that correspond to the path traced out by the operating point in Figure 3-5a. Notice that the oscillation amplitude is extremely large and the shape is non-sinusoidal. In Figure 3-5a, the operating point starts at point A and moves to the left along the compressor characteristic curve as the throttle valve closes. When the operating point reaches point B, which is where the compressor characteristic curve slope is zero, the operating point jumps to point C. This jump corresponds to the precipitous drop in flow that signals the start of the surge cycle. The operating point

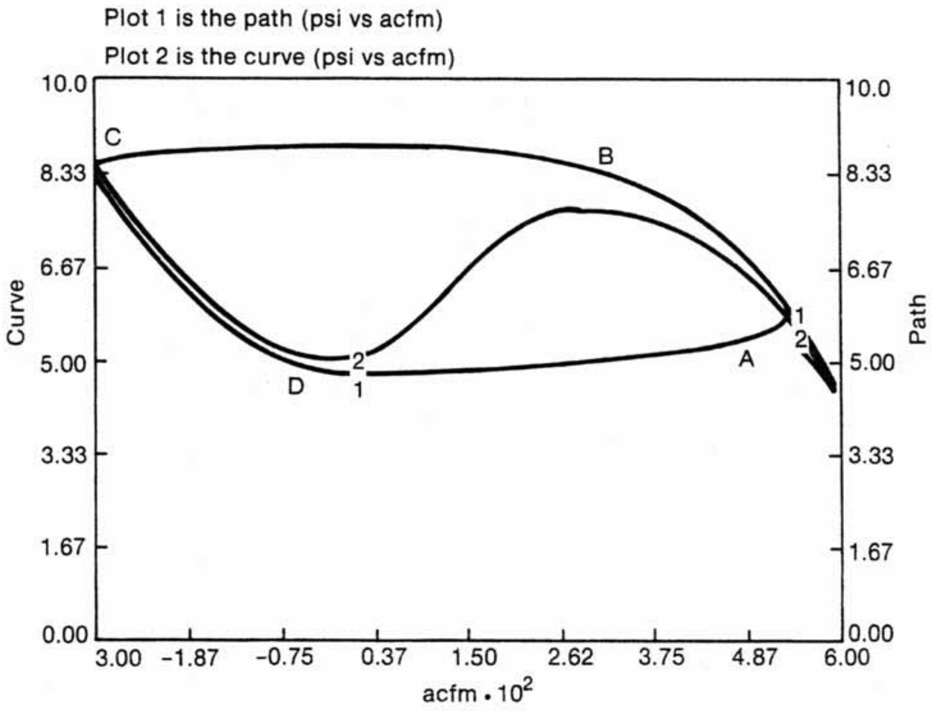

Figure 3-5a Operating Point Path and Compressor Curve (Severe Surge)

cannot follow the positive sloped portion of the compressor curve because the reduced flow out of the plenum, caused by the closing throttle valve, requires that the plenum pressure increase instead of decrease per the curve. (The plenum pressure must increase if the flow into the plenum exceeds the flow out of the plenum.) The flow pattern around the impeller breaks up, and the flow reverses direction to backward flow from the discharge volume to the compressor suction. After this jump to point C, the operating point follows the compressor curve from point C to point D as the plenum volume is emptied due to reverse flow. When the operating point reaches point D, which is where the compressor characteristic slope is zero again, the operating point jumps to point A. The operating point cannot follow the positive sloped portion of the compressor curve because the reduced forward flow into the plenum volume requires that the plenum pressure decrease instead of increase per the curve. (The plenum pressure must decrease if the flow out of the plenum exceeds the flow into the plenum.) The flow pattern around the impeller is established and the flow forward rapidly increases. After the jump to

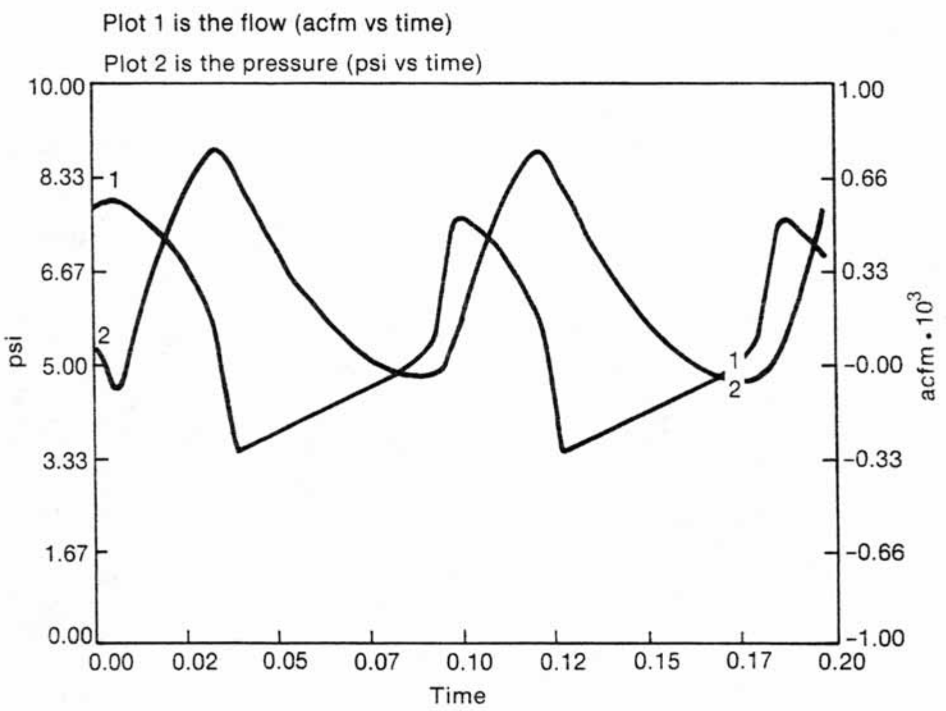

Figure 3-5b Suction Flow and Discharge Pressure Oscillations (Severe Surge)

point A, the operating point follows the compressor curve to point B as the plenum volume is filled. The surge cycle repeats itself unless the throttle valve or a surge control valve is opened. The operating point follows the compressor curve only during the peaks and valleys of the surge flow oscillations. The jumps on the compressor map correspond to the rapid transition between the peaks and valleys. The oscillation period is about 1.5 times the period predicted by Equation 3-6.

Figure 3-6a shows the path traced out by the operating point on a compressor map for the transition between surge and stall. *B* has been decreased until it is about equal to the minimum *B* for dynamic stability by decreasing the speed as a throttle valve is closed downstream of the compressor. Figure 3-6b shows the oscillations in suction flow and discharge pressure that correspond to the path traced out by the operating point in Figure 3-6a. Notice that the oscillation amplitude is an order of magnitude less than that in Figure 3-5b, and the shape is nearly sinusoidal. The *B* value is actually slightly less than the minimum *B* because the oscillation amplitude is gradually decreasing, and the path traced out by the

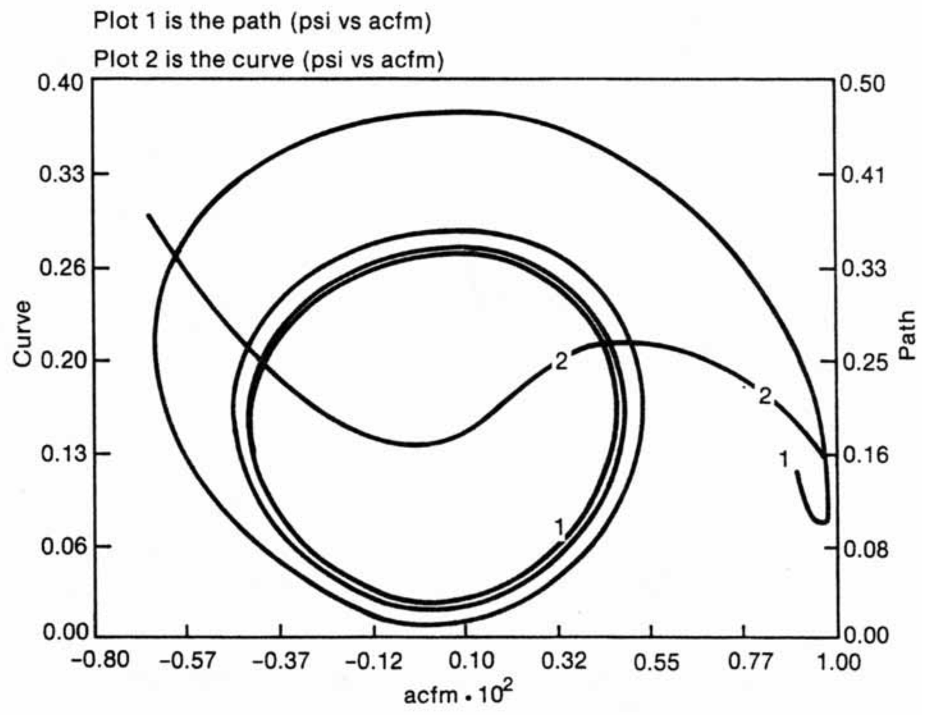

Figure 3-6a Operating Point Path and Compressor Curve (Surge to Stall Transition)

operating point in Figure 3-6a is gradually spiralling inward. The surge oscillation period is about equal to that predicted by Equation 3-6.

Figure 3-7a shows the path traced out by the operating point on a compressor map for stall where *B* has been decreased to about 0.01 times the minimum *B* for dynamic stability by decreasing the speed as a throttle valve is closed downstream of the compressor. Figure 3-7b shows the oscillations in suction flow and discharge pressure that correspond to the path traced out by the operating point in Figure 3-7a. Notice that the oscillation amplitude is 2 orders of magnitude less than in Figure 3-6b and decays to zero. The path traced out by the operating point in Figure 3-7a rapidly spirals inward and converges to the point of zero flow. The operating point approaches zero flow because the throttle valve downstream closed completely, which stopped flow out of the system except for a small leakage flow. If the throttle valve had remained slightly open, the spiral would have converged to a point of positive flow on the positive sloped portion of the compressor curve. The stall oscillation period is about 1/2 the period predicted by

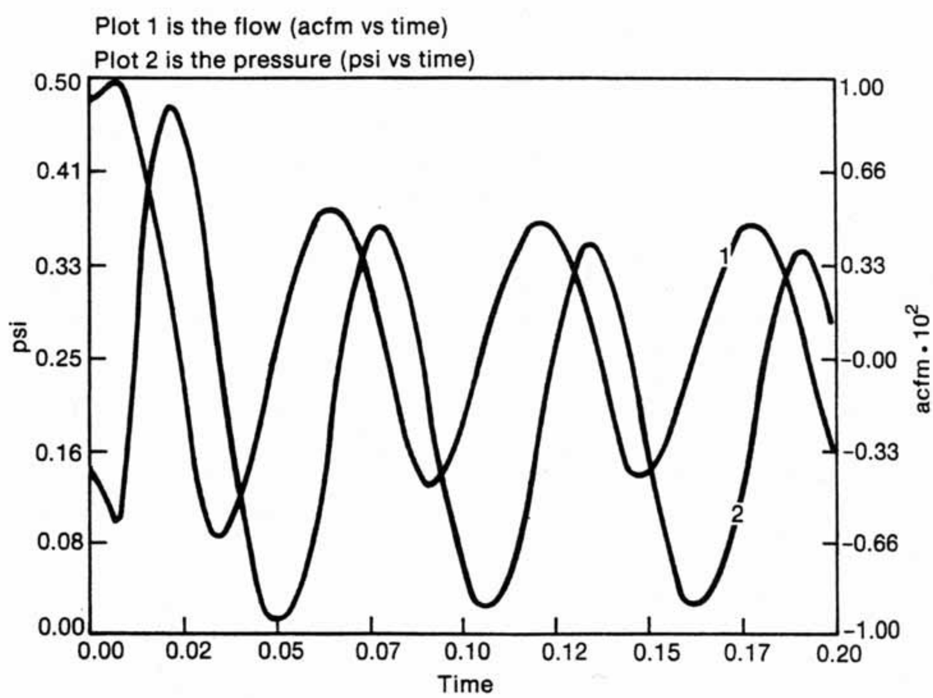

Figure 3-6b Suction Flow and Discharge Pressure Oscillations (Surge to Stall Transition)

Equation 3-6. This characteristic of compressor response, where the oscillation period decreases as the amplitude decreases, is the opposite of the characteristic of general control system response. The ultimate period of a control system occurs at the transition from stable to unstable operation (similar to the transition from stall to surge). However, for unstable operation (growing oscillations) the control loop period decreases, and for stable operation (decaying oscillations) the loop period increases.

The plots in Figures 3-5 and 3-6 were generated by an advanced continuous simulation language (ACSL) program documented in Appendix B. The program is based on the Greitzer model of surge that was described and experimentally tested by K.E. Hansen et al. (Ref. 16). The ACSL program integrates the differential equations for the momentum and mass balances and computes the stability parameter *B*, the surge period, and the system flows and pressures. The user must enter data on compressor curve shape, compressor speed, system geometry, surge valve, and throttle valve.

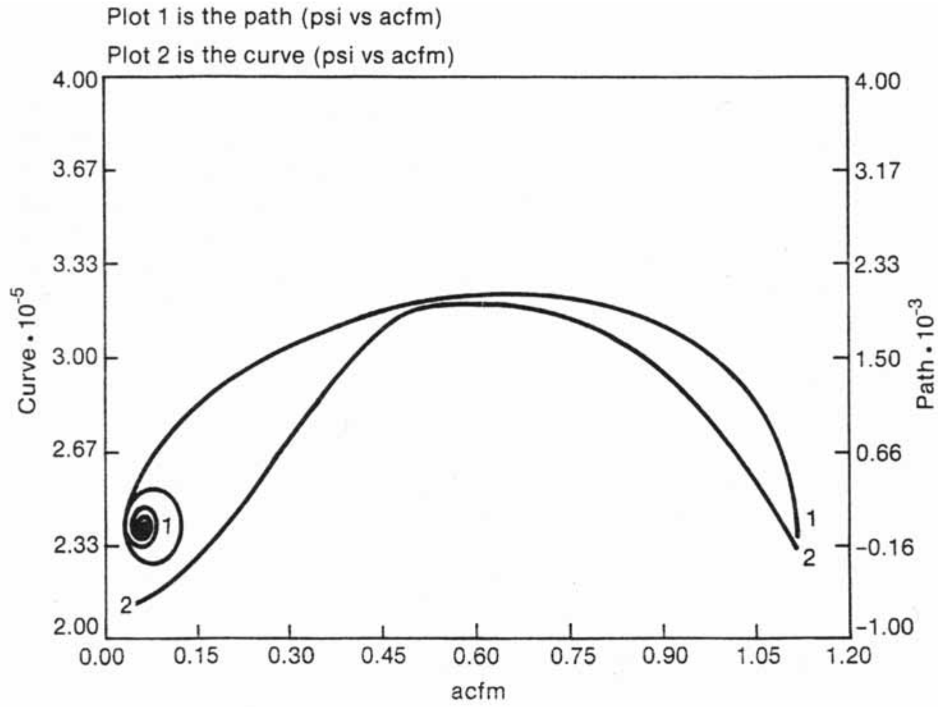

Figure 3-7a Operating Point Path and Compressor Curve (Stall)

Key Concepts

- Deep surge starts with a precipitous drop in total flow.
- The surge period is much shorter than the control loop period.
- The surge period increases with volume for a given installation.
- The surge period increases slightly with surge severity.
- The surge shape becomes more non-sinusoidal with surge severity.

3-5 CONSEQUENCES OF SURGE

The rapid flow reversals cause extensive radial vibration and axial thrust displacement. The reheating of the same mass of gas during each surge cycle causes a large temperature increase. The gas temperature rise is particularly dramatic for axial compressors, with an observed increase in temperature of 3000°F after 10 surge

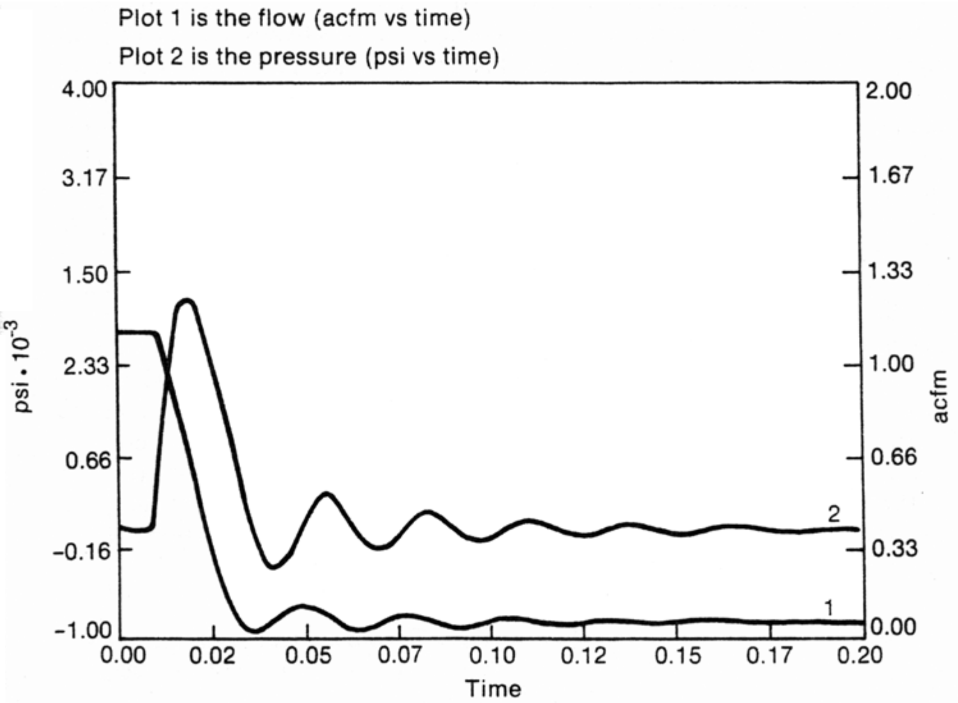

Figure 3-7b Suction Flow and Discharge Pressure Oscillations (Stall)

cycles for one installation (Ref. 26). Vibration and thrust monitors have built-in electronic delays to prevent false alarms or trips due to noise. Temperature sensors have thermal lags due to the resistance to heat transfer of the sensor and thermowell construction (see Section 7). As a result these instruments will usually actuate an alarm or shutdown after several surge cycles have occurred. In the interim, the increase in vibration, thrust, and temperature can cause extensive damage to the compressor. The repair costs can range from thousands to millions of dollars. If the compressor is the sole supplier of gas for a plant unit, the business interruption loss can be much greater than the repair costs, particularly if a replacement rotor must be manufactured, the plant product was sold out at the time of the interruption, or customers switch permanently to a competitor. Even if the damage is not noticeable, the surge can change the internal clearances enough to decrease the efficiency of the compressor. Up to a 0.5 percent loss in efficiency may result from a few cycles of surge (Ref. 26). Repeated surges will cause a significant accumulated loss in efficiency.

The rapid unloading of the impeller during flow breakdown can cause overspeed of the compressor. Field measurements (with an oscillographic recorder) of the speed of one compressor during the start of surge indicated that the compressor acceleration increased to 2000 rpm per sec until power was removed. This increase in the derivative of the speed (acceleration) is characteristic of a runaway or positive feedback open-loop response (Ref. 21). The runaway response was so rapid that the speed controller for the tested compressor could not react and prevent overspeed damage at the start of surge. Consequently, the only alternative was to shut down the compressor when the output of a derivative module indicated the start of a runaway speed response (700 rpm per second). The same module started the oscillographic recorder at 300 rpm per second acceleration to capture a record of the operating conditions just prior to the shutdown. The power was removed from the compressor by closing the steam turbine and expander supply valves in less than one half second.

The flow reversals during surge can produce a booming noise loud enough to make surge a memorable experience for personnel in the vicinity of the compressor. The booming noise can originate from collapsing gas voids within the compressor, the flexing of the suction filter walls, or the slamming shut of the check valve.

KEY CONCEPTS

- High vibration, thrust, temperature, and speed can occur during surge.
- Many instruments are too slow to detect surge quickly enough.
- Internal damage and loss in efficiency can result from surge.