TUTORIAL SESSION
on
PRACTICAL APPROACH TO SURGE AND SURGE CONTROL SYSTEMS

Tutorial Leaders

Meherwan P. Boyce (Session Chairman)
Boyce Engineering International, Incorporated
Houston, Texas

William R. Bohannan
Bechtel, Incorporated
San Francisco, California

John R. Gaston
Dresser Industries—Clark Division
Olean, New York

Robert H. Meier
Cooper Energy Services
Mt. Vernon, Ohio

Royce N. Brown
Dow Chemical, U.S.A.
Houston, Texas

Cyrus Meher-Homji
Boyce Engineering International, Incorporated
Houston, Texas

Norbert E. Pobanz
Bechtel, Incorporated
San Francisco, California

PRACTICAL ASPECTS OF CENTRIFUGAL COMPRESSOR SURGE AND SURGE CONTROL

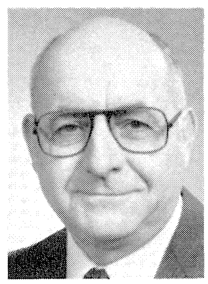
Meherwan P. Boyce is President of Boyce Engineering International, Inc., a major Houston-based consulting and engineering firm. His past experience incorporates many academic and professional positions, including Director of the Gas Turbine Laboratories and Professor of Mechanical Engineering at Texas A&M University, where he organized and was chairman of the Turbomachinery Symposium. Dr. Boyce has authored more than fifty significant papers and technical reports on fluid mechanics and turbomachinery, and he is a member of SAE, ASME, and several other professional and honorary societies. He is also a registered professional engineer in the State of Texas.

Dr. Boyce received B.S. and M.S. degrees in Mechanical Engineering from the South Dakota School of Mines and Technology and the State University of New York, respectively, and his Ph.D. in Mechanical Engineering from the University of Oklahoma.

William R. Bohannan is Chief Mechanical Engineer and Technology Coordinator for the San Francisco Division of Bechtel Petroleum, Inc. He heads a group of engineering specialists engaged in the selection studies, specification, evaluation, application engineering, testing, and commissioning of mechanical equipment associated with projects handled by this division. He also guides the Division's involvement in technical societies and committees. Much of his thirty years' experience has been involved in compressor and turbine applications. His previous assignments include seven years with a compressor manufacturer and five years with petroleum refining companies.

Mr. Bohannan holds a B.S. degree in Mechanical Engineering from South Dakota School of Mines and Technology. He is a registered professional engineer in the State of California and a member of ASME and the Vibration Institute, and he participates in the mechanical equipment activities of API.

Royce N. Brown is a Consulting Engineer with Dow Chemical U.S.A., Engineering and Construction Services, Houston, Texas. His responsibilities include specifications, consulting and field assistance in the area of large rotating equipment for Dow world-wide. Together with his responsibilities as Consulting Engineer, he heads the Rotating Equipment and Critical Equipment Instrumentation Group. He holds a B.S. degree in Mechanical Engineering from the University of Texas and a M.S. degree in Mechanical Engineering from the University of Wisconsin. He is a fellow member of ISA; a member of ASME, SME, and API; and an associate member of SAE. He is active in the Mechanical Equipment subcommittee of the CRE, where he is the chairman of the API 617 Task Force. Mr. Brown is a registered professional engineer in Texas, Louisiana, Wisconsin, and California.

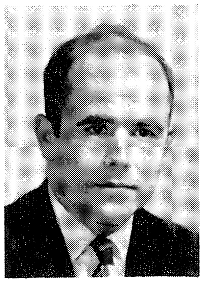


John R. Gaston is a Senior Controls Design Engineer for Dresser Industries, Inc., Clark Division, Olean, NY. He joined Dresser Clark in 1956 and has held various positions in Field Service. Since 1967, he has been responsible for application and design engineering of control systems for turbo (centrifugal and axial flow) compressors. He has presented papers on compressor control at various ISA and other conferences. He received the Chemical-Petroleum Division of ISA's Dick Pond award in recognition of the best technical paper at the annual Symposium held at San Francisco in 1974. Mr. Gaston is a member of the National Management Association, a Senior Member of the Instrument Society of America and a registered professional engineer (control systems) in the State of California.

Cyrus Meher-Homji is Engineering Consultant Manager with Boyce Engineering International, where he is involved with various consulting work in the turbomachinery and engineering management areas.

His counsulting activities include design, vibration and performance analysis, economic analysis and reliability engineering. In the past, he was a development engineer with Boyce Engineering and was involved in the design and development of a prototype externally fired steam injected gas turbine for the U.S. Department of Energy. His areas of interest are rotor dynamics, turbomachinery prognosis and diagnosis, engineering management, and reliability and maintainability.

Mr. Meher-Homji has a B.S. degree in Mechanical Engineering and an M.E. degree in Industrial Engineering from Texas A&M University, and a M.B.A. degree from the University of Houston.



Robert H. Meier joined Cooper Energy Services Company in 1966 and has had a long association with centrifugal compressor design and development. Presently, he is Manager of Design and Development for Rotating Products. His ability to read and speak German and French is especially valuable in presentations and specifications review. He holds a degree from the Swiss Federal Institute of Technology.

Norbert E. Pobanz graduated from UCLA with a B.S. degree in General Engineering in January, 1958. In June, 1981, he received a MBA, Management, from Golden Gate University in San Francisco. During the past twenty-four years, he has been closely associated with the use of computer simulation to evaluate engineering designs in many technical fields. During the past eight years, he has been in charge of a simulation and advanced control specialty group for Bechtel Petroleum, Inc. Recently, his responsibilities have expanded to include management of a dedicated simulation facility to perform studies and to analyze distributed control systems. Prior to his years with Bechtel, he was employed by Electronic Associates, Inc. for fourteen years. His responsibilities included dynamic analysis, technical support for marketing, and management of computer simulation facilities.

Mr. Pobanz is a registered professional engineer (control systems) in the State of California. He has Senior Member status in the Instrument Society of America and the Society for Computer Simulation (SCS). Presently, he is Senior Vice President of SCS. He has published numerous articles in the field of simulation.

ABSTRACT

This paper addresses the area of compressor stability, surge and surge control and relates to the practical aspects involved. An emphasis is placed on the physical understanding of surge phenomena and on the practical limitations of surge control systems. Topics discussed are physical interpretation of instability, causative factors, types of stall, machine and process design factors, surge effects and characteristics, control system types and practical application aspects. Some case studies also are presented. The discussion primarily relates to centrifugal compressors, but several aspects pertain to axial flow compressors as well. The paper is split into three sections: Section A consists of an introduction to surge and a discussion of centrifugal compressor design and process factors that affect operating stability. Section B discusses the various types of control schemes and surge protection devices. Several examples and common pitfalls are addressed. Section C addresses the important design tool, surge system simulation. Several references are provided to enable the reader to pursue this topic in greater detail.

A. INTRODUCTION TO CENTRIFUGAL COMPRESSOR SURGE

Centrifugal compressor systems at numerous installations have suffered serious mechanical damage to compressor internals or to other parts of the piping systems as a result of operation in the surge condition. In other cases, compressors, usually smaller machines, have operated for long periods with intermittent or even continuous light surge without mechanical harm, although with significant impairment of aerodynamic performance. These conditions generally are the product of one or more of the following deficiencies:

1. Poor matching of the compressor to the system's requirements.
2. Inappropriate compressor design.
3. Inadequate anti-surge control system.
4. Unfavorable arrangement of piping and process components of the system. (This often can magnify surge.)

Because of the shape of their performance curves, the application of centrifugal compressors frequently is considerably more complex than that of reciprocating machines. A centrifugal compressor driven by an electric motor offers an imposing challenge to the controls engineer when this constant-speed machine must accommodate a wide range of variables in its operating conditions. Centrifugal compressor applications continue to become more complicated with increases in the numbers of stages per casing, casings in tandem with a single driver, and sidestream nozzles; higher pressures and speeds; and more operating conditions for a given machine involving wider ranges of flows, molecular weights, and pressures. This ever-increasing complexity requires a better understanding of the causes of surge and its detrimental effects in order that adequate control systems may be applied. It also creates the need for surge control system simulation studies, which must be conducted at the design phase. This topic is addressed in Section C.

A typical performance map for a centrifugal compressor is shown in Figure 1. The operating line is the surge line modified by a safety margin to ensure trouble-free operation. Note that the total pressure ratio changes with flow, speed, molecular weight, suction pressure and temperature. Also, note that operating at higher efficiency implies operation closer to surge. It should be noted here that total pressure increases occur only in the impeller. To make the curve general, the concept of aerodynamic speeds and corrected mass flow rates has been used.

The surge line slope on multistage compressors can range from a simple, single-parabolical relationship to a complex curve containing several break-points or even "notches." The complexity of the surge line shape depends on whether or not the flow limiting stage changes with operating speed from one compression stage to another; in particular, very closely matched stage combinations frequently exhibit complex surge lines. In the case of compressors with variable inlet guide vanes, the surge line tends to bend more at higher flows than with units which are speed controlled.

Usually, surge is linked with excessive vibration and an audible sound; yet, there have been cases in which surge problems which were not audible caused failures. Usually, operation in surge and, often, near surge is accompanied by several indications, including general and pulsating noise level increases, axial shaft position changes, discharge temperature excursions, compressor differential pressure fluctuations, and lateral vibration amplitude increases. Frequently, with high pressure compressors, operation in the incipient surge range is accompanied by the emergence of a low frequency, asynchronous vibration signal which can reach predominant amplitudes, as well as excitation of various harmonics of blade passing frequencies. Besides the well-known effects of extended operation in surge (thrust and journal bearing failure, impeller rub), impeller hub and/or shroud failures resulting

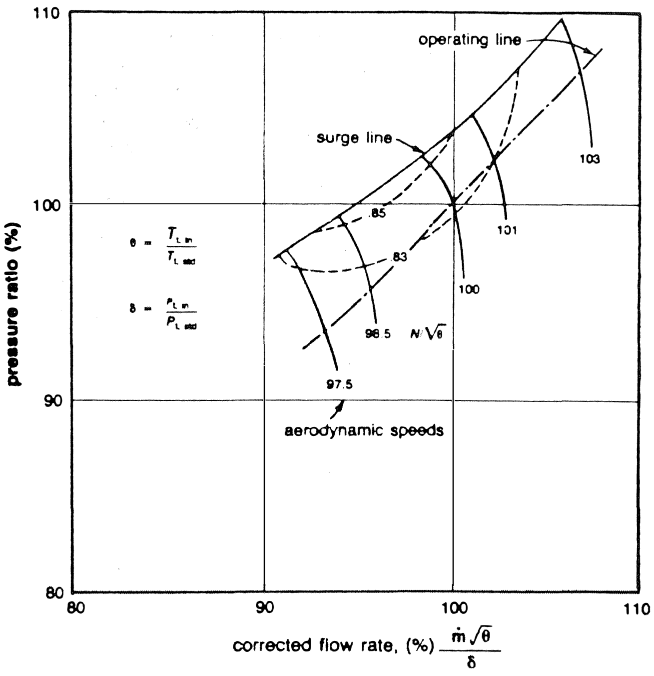


Figure 1. Compressor Performance Map.

from severe stimulation at one of their natural response frequencies occasionally can be found.

Extensive investigations have been conducted on surge, starting as early as 1946 by Bullock [1] and his associates, and later by Emmons [2]. Most of the work involved experiments on particular compressors and, hence, lacked generality. Again, poor quantitative universality of aerodynamic loading capacities of different diffusers and impellers, and an inexact knowledge of boundary layer behavior make the exact prediction of flow in turbomachines at the design stage difficult. It is, however, quite evident that the underlying cause of surge is aerodynamic stall. The stall may occur in either the impeller or the diffuser. Recent significant work in this area is reviewed in Reference 3.

Definition of Surge

The phenomenon of surge, as it pertains to a centrifugal compressor and its connected system, is an unstable condition resulting in flow reversals and pressure fluctuations in the system. This condition occurs when there is sufficient aerodynamic instability within the compressor that the compressor is unable to produce adequate pressure to deliver continuous flow to the downstream system. The system and compressor then interact, causing the surge conditions with large and sometimes violent flow oscillations in the system. Surge, then, is an overall system phenomenon and is not confined to the compressor only.

Surge is the result of an excessive increase in the resistance of the system while the compressor is operating at a certain speed. The added resistance reduces the flow to an unstable level. Alternatively, if the resistance is unchanged, but the speed is reduced appreciably, most systems will surge. Thus, surge occurrence depends on the type of system and the shape of the resistance curve.

The aerodynamic instability is brought about by flow reduction, which causes stalling of one or more of the elements of a stage or stages of the compressor. The stalling can occur at the inducer of the impeller, in the radial portion of the impeller, in the diffuser, or in the volute. A stall in one of these elements may not have sufficient effect to cause the stage to be unstable. In fact, several elements of a stage can stall without the entire stage stalling [3]. However, if the stalling is of sufficient strength, the stage will become unstable, and this can lead to surge of the compressor.

The stall of an element of a compressor stage may be compared with that of an airfoil of an airplane. The lift of an airfoil is related to the velocity of the air flow and the angle of attack (incidence). If the angle becomes excessive, the lift collapses, and a stalled condition results. The stall occurs because the airstream separates from the surface of the airfoil. Airfoil stall is described by Figure 2. Japikse [3] gives a detailed explanation of stalling of compressor elements and covers dynamic instability of rotating stall.

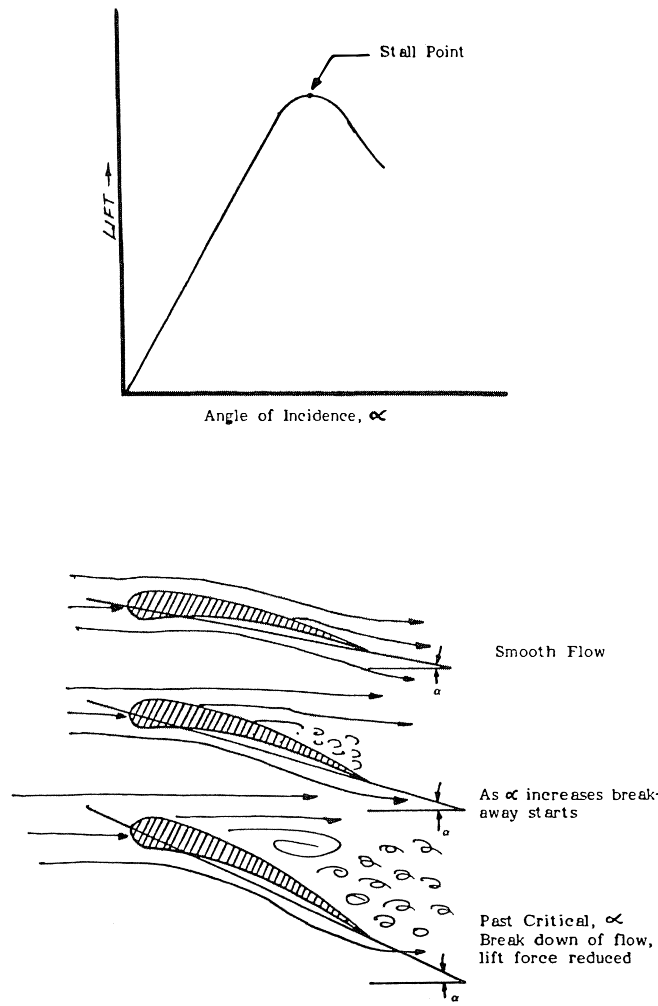


Figure 2. Lift vs. Angle of Incidence and Airfoil Flow Visualization as α Increases.

A typical centrifugal multi-stage compressor performance map used in the process industry is shown in Figure 3. The family of curves depicts the performance at various speeds, where N represents rpm. The ordinate may be polytropic head "H," pressure ratio, discharge pressure, or sometimes differential pressure. The abscissa, usually called "Q" or "Q_i," is almost always shown as actual inlet volume per unit of time, such as acfm or icfm, where "a" is for "actual," or "i" for "inlet." It is important to understand that the inlet flow volume or capacity is based on a gas with a particular molecular weight,

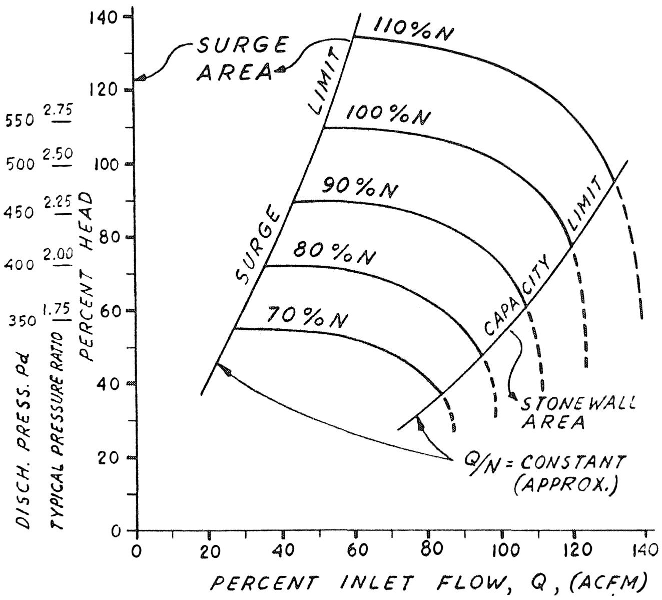


Figure 3. Typical Centrifugal Compressor Performance Map.

specific heat ratio, and compressibility factor at a pressure and temperature corresponding to the gas condition in the suction line to the compressor. If any of these parameters is changed, the performance map is no longer exactly valid. If the deviations are relatively small, the map still may be used with a fair degree of accuracy by making small adjustments. It is often impractical to make a separate set of curves for every small variation in these parameters. Since the impeller recognizes only actual volume, weight flow such as pounds per minute, or standard volume such as scfm can only be used on the abscissa when there are absolutely no significant variations in gas conditions at suction.

The line on the left represents the surge limit, or "pumping limit," as it is often called. Operation to the left of this line is unstable, resulting in unsatisfactory performance, and is often harmful mechanically. Notice that the surge flow increases as the speed increases. The surge line in many cases has an approximately constant value of Q/N. In many multistage units, and especially in units which are controlled by variable inlet guide vanes, the surge line curves bend dramatically at higher flows and speeds.

On the other side of the map, the capacity limit or overload line is shown. Operation to the right of this line causes the head-producing capability of the machine to fall off very rapidly, and the performance is difficult to predict. The area to the right of this line is commonly known as "stonewall," or "choke." Operating the machine in this region is usually harmless mechanically, although a few impeller failures have been ascribed to prolonged operation in stonewall. A compressor in stonewall is like an orifice at critical flow. It means that a gas velocity somewhere in the compressor, usually at the impeller inlet (inducer), has reached sonic velocity. The capacity limit line also is a line of roughly constant Q/N values. Plant operators sometimes get the terms "surge" and "stonewall" confused, because, presumably, machine performance is seriously impaired in either case. However, it must be remembered that the two phenomena are completely different and create completely different operational problems.

Figure 4 is the same as Figure 3, except that some points have been labelled A, B, C, and D, and three typical system operating curves have been plotted. Terms frequently used to define performance are "stable range" and "percent stability." The rated stable range is generally taken as $Q_A - Q_B$, where Q_A is the design or rated point, and Q_B is the surge point along the 100% speed line. The percent stability is

$$\frac{Q_A - Q_B}{Q_A},$$

expressed as a percentage.

The use of these terms is somewhat misleading, since operation between the surge line and the capacity limit line is stable and predictable. Notice that the span between the capacity limit and the surge limit decreases as speed is reduced. It is also important to observe that, at lower speeds, the compressor does not surge until it reaches lower inlet flows.

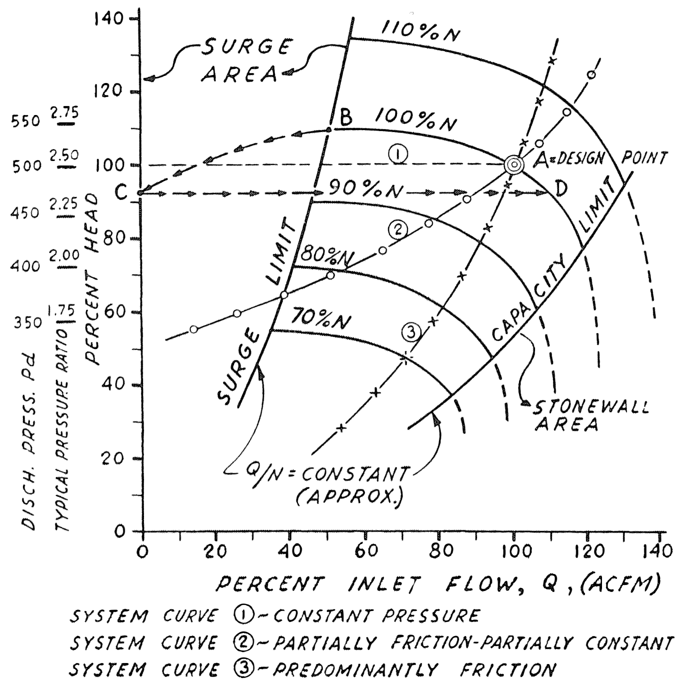


Figure 4. Performance Map With System Curves and Surge Cycle.

Rise-to-surge (RTS) is another term often used to assess the capability of a compressor to recover from a disturbance in the system. A high RTS means that the machine can accommodate a modest increase in discharge pressure with little change in inlet flow. With a low RTS, the curve is relatively flat, and there could be a large reduction in inlet flow corresponding to a very small increase in discharge pressure. A discussion of the control aspects due to the behavior of the discharge pressure with flow is detailed in Section B. Pressure RTS is defined as

$$\frac{P_B - P_A}{P_A},$$

expressed as a percentage. RTS also may be expressed in terms of head. When using the RTS, it is essential to state whether it is based on pressure or head. Pressure RTS is not necessarily numerically equal to head RTS.

The three representative system curves (Figure 4) need little explanation. The shape of these curves is governed by the amount of friction, fixed pressure drop, or pressure control in the particular system external to the compressor through which the flow is being pumped. However, it should be noted that, to follow any one of these system curves, the speed must be changed. This, in turn, changes the flow. With Systems 2 and 3, the head is also changed. Of course, these statements are valid only if no changes within the system itself (such as a control valve setting) occur. Changing the setting of a control valve, adding another piping loop, or changing the catalyst level in a reactor, etc., will modify the system curve. These obvious relationships are mentioned only to emphasize the operational difference between constant speed and variable speed machines. At constant speed, something in the system itself must be changed to relocate an operating point if the gas composition and compressor hardware stay the same.

A typical surge cycle is represented in Figure 4 by the circuit between Points B, C, D, and back to B. Suppose that events gradually or suddenly take place to establish operation at Point B. Here, the pressure in the system is equal to the output pressure of the compressor. If any transient variable caused operation to shift slightly to the left, reverse flow would begin because the compressor discharge pressure would be less than the pressure already in the system. In order for reverse flow to occur, the flow delivered by the compressor must be reduced to zero at Point C, which corresponds to a certain value of head or pressure, called "shut-off head." When the system pressure had blown itself down to the compressor's shut-off head capability at Point C, the machine would begin to pump anew, since the flow volume requirement of the compressor would have been satisfied by the backflow gas. Now that the compressor had some gas to pump, operation would immediately shift to the right in an approximately horizontal path to Point D on the speed line at a discharge pressure about equal to the shut-off head. With the compressor now delivering flow in the forward direction, pressure would build up in the system, and operation would follow the characteristic speed curve back to Points B and C. The cycle would repeat itself again and again unless the original cause of the surge was corrected, or other favorable action, such as increasing the speed, was taken.

To a large extent, the frequency of the surge cycle varies inversely with the volume of the system. For example, if the piping contains a check valve located near the compressor discharge nozzle, the frequency will be correspondingly much higher than that of a system with a large volume in the discharge upstream of a check valve. The frequency can be as low as a few cycles per minute, up to 20 or more cycles per second. Generally speaking, if the frequency is higher, the intensity of surge is lower. The intensity or violence of surge tends to increase with increased gas density, which is directly related to higher molecular weights and pressures and lower temperatures. Higher differential pressure generally increases the intensity. The location of Points C and D (Figure 4) were randomly selected for illustration purposes, so the values of head and flow may not be realistic.

In low head stages, surge usually is initiated in the diffuser inlet section of the stage. In high head stages, surge can be initiated in the inducer section off the impeller. Studies by Dean et al. [4] have shown that, oftentimes, flow rates oscillate continuously at large amplitudes, even when a compressor is operating in its "stable" region. Moreover, there is evidence that instantaneous operation can be stable far to the left of the nominal surge line. Also, the time average operating point must lie at a significantly larger mass flow than the instantaneous stability limit.

Rotating Stall

Rotating stall (propagating stall) consists of large stall zones covering several blade passages, and propagates in the direction of the rotor and at some fraction of rotor speed. The number of stall zones and the propagating rates vary considerably. Rotating stall can and does occur in centrifugal compressors.

The propagation mechanism can be described by considering the blade row to be a cascade of blades (say, an inducer), as shown in Figure 5. A flow perturbation causes Blade 2 to reach a stalled condition before the other blades. This stalled blade does not produce a sufficient pressure rise to maintain the flow around it, and an effective flow blockage or a zone of reduced flow develops. This retarded flow diverts the flow around it so that the angle of attack increases on Blade 3 and decreases on Blade 1. The stall propagates downward relative to the blade row at a rate about half the block speed; the diverted flow stalls the blades below the retarded-flow zone and unstalls the blades above it. The retarded flow or stall zone moves from the pressure side to the suction side of each blade in the opposite direction of rotor rotation, and it may cover several blade passages. The relative speed of propagation has been observed from compressor tests to be less than the rotor speed (40-75% of rotor speed). Observed from an absolute frame of reference, the stall zones appear to be moving in the direction of rotor rotation. This phenomenon can lead to inefficient performance and excitation of the resonant frequency of the inducer, thus leading to failure of that section. Rotating stall is accompanied sometimes by a pulsating sound and pressure pulsations that can be noted at both the inlet and exit sections of the impellers.

Effects of Internal Losses on Characteristic Curve

Several internal factors influence the shape of the characteristic performance curve and the location of the surge point. From the foregoing description of surge, it can be seen that the shape of the head-capacity characteristic curve is fundamentally responsible for the location of the surge point at a certain speed. On the right side of the performance map, the slope of

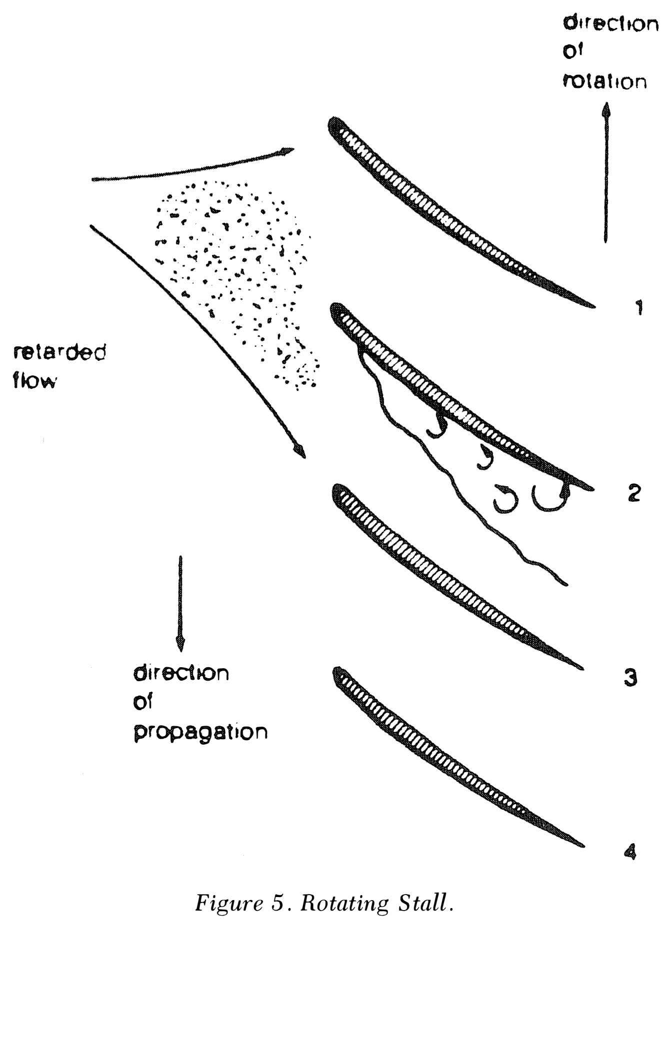


Figure 5. Rotating Stall.

the curve is negative. As inlet flow is reduced, the slope becomes less negative until it reaches essentially zero at the surge point. As flow is reduced further to the left of the surge point, the slope becomes increasingly positive. It should be pointed out that neither zero slope nor a positive slope are absolute criteria for surge; however, in most cases it is recommended that surge controls be set at a point where the slope becomes zero.

As might be expected, there are very definite interdependent effects resulting from the individual geometries of the principal components of the compressor. Such pertinent variables as the impeller configuration and blade angle, inlet guide vane angle, diffuser size and shape, etc. can be adjusted by the machine designer for optimum performance under a specified set of operating conditions. The vectorial summation of all the gas velocity vectors associated with the several components of the machine govern the theoretical head-capacity relationship. Ideal or theoretical head is a straight line when plotted against inlet capacity, and the overall geometry dictates the slope of the line as seen in Figure 6, depending on the exit blade angle. As usual with any machine, inevitable losses, which cannot be recovered, intervene to produce a significant difference between the theoretical and actual output. Mechanical losses such as those incurred in a journal or thrust bearing affect the power input to the machine, but do not influence the shape of the head-capacity curve. Aerodynamic losses that do influence the shape of the curve consist mainly of wall friction, fluid shear, seal losses, recirculation in flow passages, and diffusion blading losses (separation losses). Diffusion blading losses are the result of expansion, contraction, and change of direction and recirculation associated with flow separation, eddies, and turbulence.

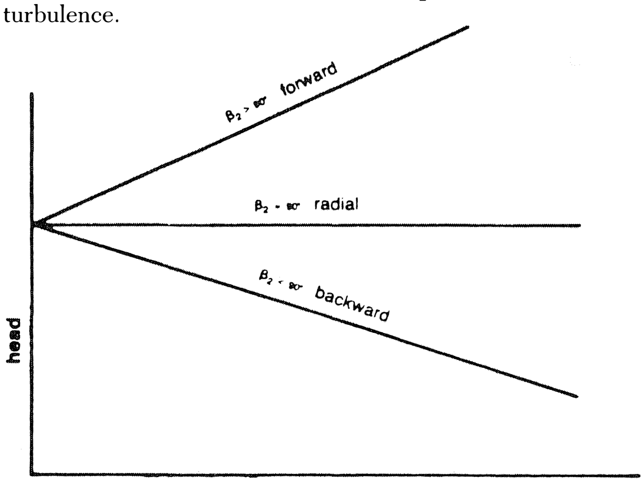


Figure 6. Head Flow Rate Characteristics for Various Outlet Blade Angles.

Figure 7 indicates graphically how the combined losses reduce the linear theoretical head to the actual curve. The geometry and dimensions of the elements within the machine can be originally designed to move the rated operating point to a more desirable location on the curve from the viewpoint of a certain variable, but this relocation will usually create one or more disadvantages from another viewpoint. For instance, if the design point is located as shown, a move slightly to the left would gain some efficiency and head output and might result in eliminating one impeller from a multistage application; however, the cost could be an objectionable loss in the stable operating range. A move to the right would produce opposite results, which could be desirable in some situations. Nevertheless, the reduced capacity overload limit or proximity to stonewall could have significant disadvantages. To summarize, the combined losses force the actual curve to go from a negative to a positive slope as flow decreases; that is, the actual curve always has a "hump" in it. Changes in machine geometry can relocate the crest of the hump, change the breadth of the hump, and alter the severity of the curve to some extent. However, the hump cannot be eliminated. The surge point is located at or very near the point of zero slope on the compressor's characteristic curve.

Friction losses can be reduced somewhat by improving surface finishes and enlarging passages. Blade diffusion losses may sometimes be mitigated by further streamlining of flow passages and assuring that there are no abrupt changes in the flow area. These techniques will improve efficiency and tend to reduce the surge point; however, they also cost money, and there is a point of diminishing marginal returns.

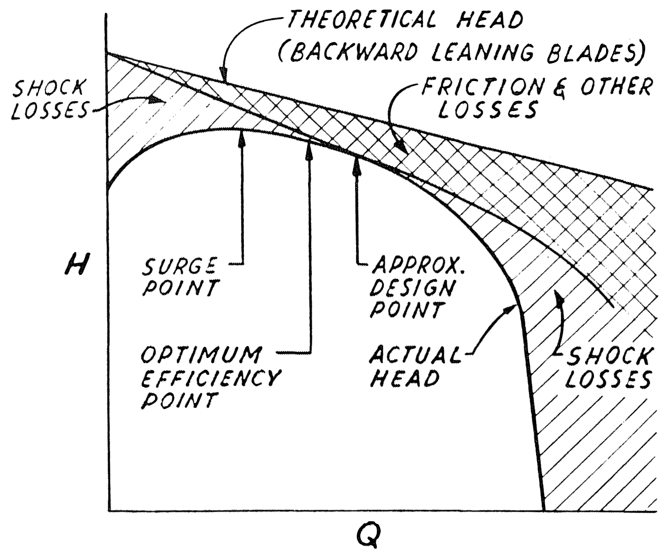


Figure 7. Typical Compressor Head Losses.

The phenomenon of slip, which is due to the build-up of boundary layer on the vanes of the impeller, occurs in all centrifugal machinery. Boundary layer controls have been used to reduce slip and increase the flow stability of the impeller, as seen in Figure 8.

Design Factors Affecting Surge

Some elaboration on the principal machine design factors that affect surge will now be covered. A greater number of impellers in a given casing will tend to reduce the stable range. This effect is presented in Figure 9.

Manufacturers have found it physically and economically impractical to design each stage of a multi-stage machine to be exactly the optimum size. As the pressure builds up across the stages, the volumetric flow is reduced. Ideally, then, each succeeding wheel should be proportionately smaller with respect to flow passage area. For example, the width of the flow exit at the tip of the impeller should actually get smaller and smaller. It would be virtually impossible to achieve perfect proportionality. This lack of optimum proportionality aggravates the stability problem. Some larger machines are custom-built, and, therefore, better proportionality can be attained. Mechanical limitations, such as the axial length of the shaft between bearings and its relation to critical speed, may dictate the axial width of the impellers through the hub and the

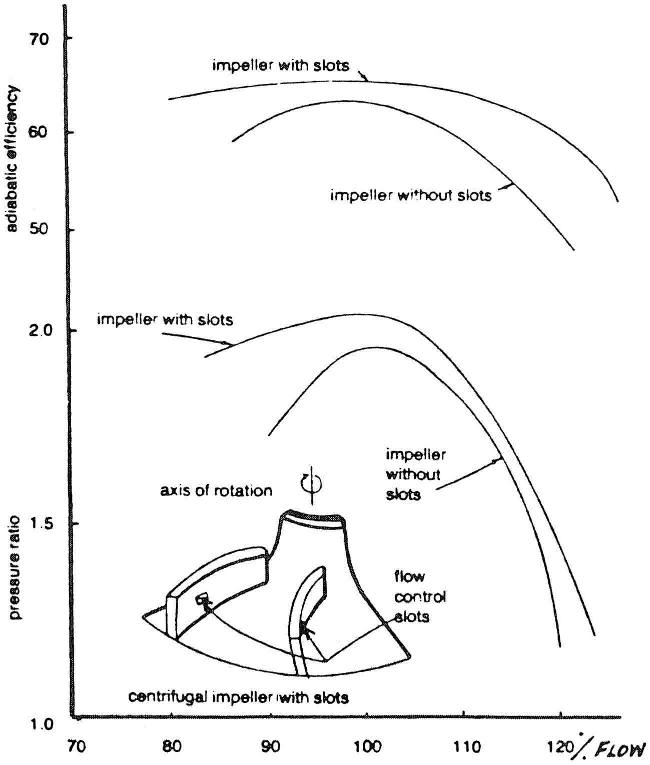


Figure 8. Laminar Flow Control in Centrifugal Compressor.

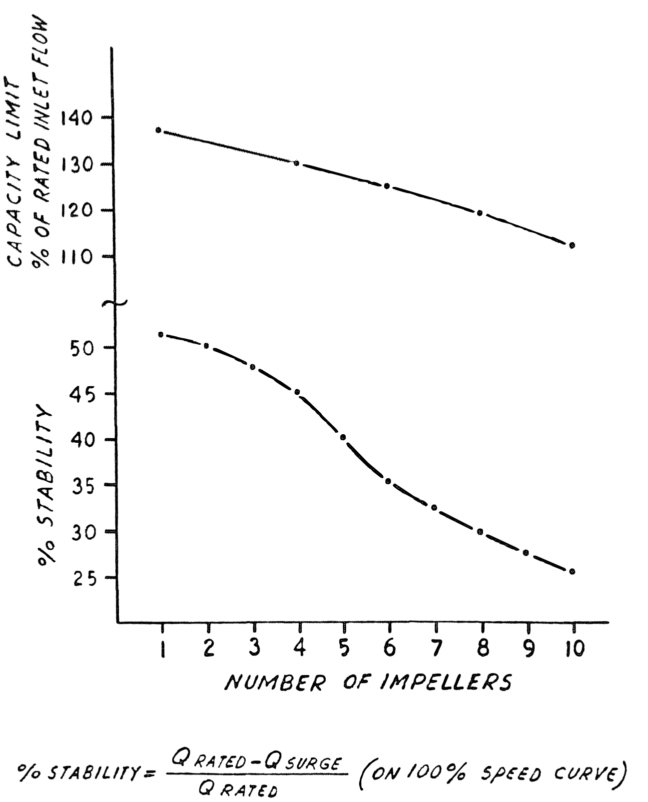


Figure 9. Effect of Number of Impellers on Stability and Capacity Limit.

impeller spacing. The shaft diameter may be constrained by torque and/or critical speed considerations, and the inside diameter limits of the impeller hub may be accordingly restricted. Practical economics and standardization occasionally will limit the number of types and sizes of wheels in a given machine. For example, in a six-wheel machine, there might be only two different wheel designs. Therefore, whereas the first impeller might be operating at optimum conditions, subsequent impellers of the same type and size usually would be operating at less favorable flow conditions, owing to the volume reduction across the stages. There are a great many other considerations which affect the overall stability. The relationship in Figure 9 is typical only, and the design can be changed to improve stability to some extent. However, more often than not, it will be at the sacrifice of some other important performance factor. The diminishing effect of the number of wheels on the location of the capacity limit is also shown in Figure 9. Just as the greater number of wheels in a casing reduces the stability, so does the number of sections of compression, or the number of casings in series. This effect is illustrated by the sample calculation shown in Figure 10. It can be seen that individual percent stability of the HP casing is better than that of the LP casing. However, as LP casing flow is reduced, the HP casing surges before the LP casing because of increase in intercasing pressure, which decreases acfm to that of the HP casing. Overall stability is less than the individual stability of either casing.

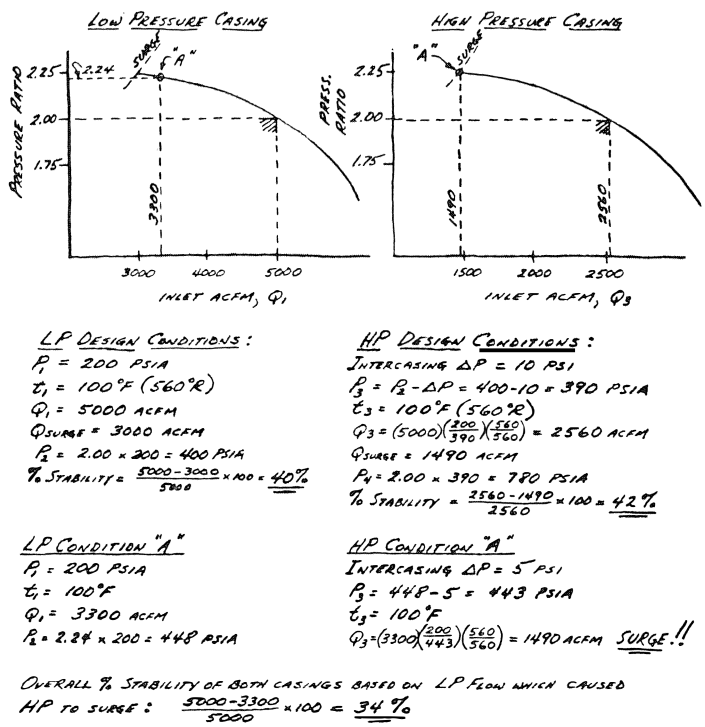


Figure 10. Typical Overall Stability vs. Individual Stability for Two Casings Operating in Series.

The information and curves presented thus far have been, for the most part, based on impellers with backward leaning impeller blades. Figure 11 [5] depicts the effects of impeller blade angle on the stable range, and shows the variance in steepness of the slope of the head-flow curve. The three curves are based on the same speed, and show actual head. The relationship of ideal or theoretical head to inlet flow for different blade angles would be represented by straight lines. For backward leaning blades, the slope of the line would be

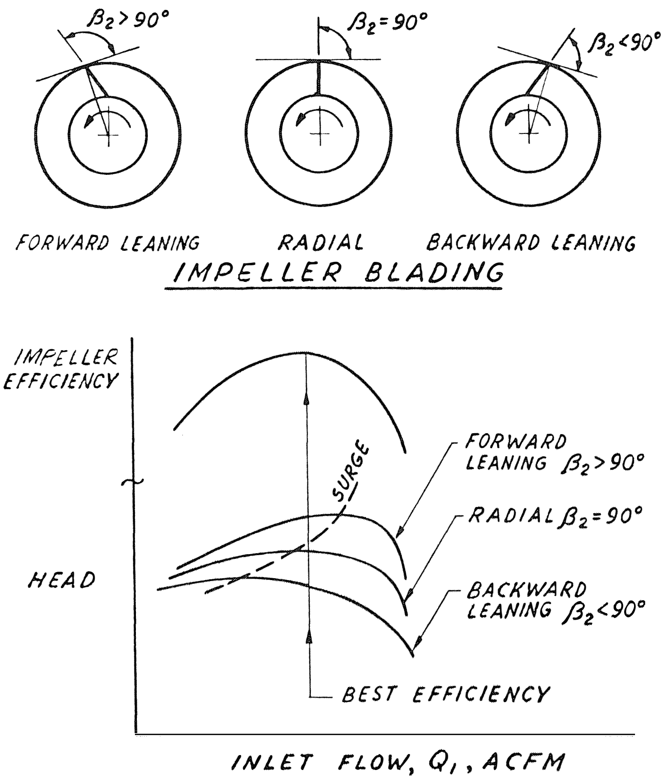


Figure 11. Effect of Blade Angle on Stability.

negative. The line for radial blades would be horizontal. Forward leaning blades would have a positively sloped line. For the average petrochemical process plant application, the compressor industry in the USA commonly uses a backward-leaning blade with an angle (β_2) of between about 55°-75° (or backward leaning angle of 15°-35°), because it provides a wider stable range and a steeper slope in the operating range. This impeller design has proven to be about the best compromise between pressure delivered, efficiency and stability. Forward leaning blades are not commonly used in compressor design, since the high exit velocities lead to large diffuser losses. A plant air compressor operating at steady conditions from day to day would not require a wide stable range, but a machine in a processing plant can be the victim of many variables and upsets, so more stability is highly desirable. Actually, the lower curve (Figure 11) appears to have a more gentle slope than either the middle or upper curve. This comparison is true in the overall sense, but it must be remembered that the normal operating range lies between 100% Q and Q at surge, plus a safety margin of, usually, about 10%. The right-hand tail ends of all three curves are not in the operating range. The machine must operate with a suitable margin to the left of where these curves begin their steep descent or tail-off, and in the resultant operating range, the curve for backward leaning blades is steeper. This steeper curve is desirable for control purposes. Such a curve produces a meaningful Delta-P for a small change in Q. The blade angle by itself does not tell the overall performance story. The geometry of other components of a stage will contribute significant effects also.

Most centrifugal compressors in service in petroleum or petrochemical processing plants use vaneless diffusers. A vaneless diffuser is generally a simple flow channel with parallel walls, and does not have any elements inside to guide the flow. Figure 12 shows velocity diagrams at the eye and exit of an impeller, and illustrates the trajectory a particle in the gas flow would take through a vaneless diffuser at the design condition (compressor rated point).

When the inlet flow to the impeller is reduced while the speed is held constant, there is a decrease in V_{R_2} and α_2. As α_2 decreases, the length of the flow path spiral increases. The effect is shown in Figure 13. If the flow path is extended enough, the flow momentum at the diffuser walls is excessively dissipated by friction and stall [6]. With this greater loss, the diffuser becomes less efficient and converts a proportionately smaller part of the velocity head to pressure. As this condition progresses, the stage will eventually stall. This could lead to surge.

Vaned diffusers are used to force the flow to take a shorter, more efficient path through the diffuser. There are many styles of vaned diffusers, with major differences in the types of vanes, vane angles and contouring, vane spacing, etc. Commonly

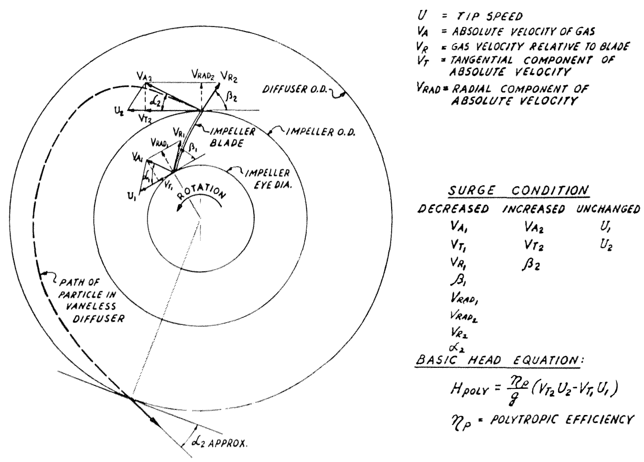
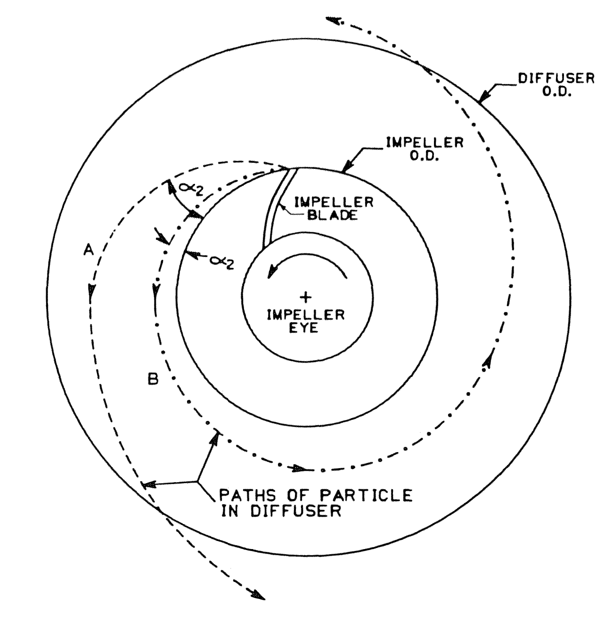


Figure 12. Design Condition Velocity Triangles.



A - NORMAL CONDITION. GOOD FLOW ANGLE. RELATIVELY SHORT FLOW PATH. NOT MUCH FRICTION LOSS.

B - TENDENCY TOWARD SURGE. SHALLOW FLOW ANGLE. LONG FLOW PATH. LARGE FRICTION LOSS. POSSIBILITY THAT PART OF THE FLOW FIELD WILL RE-ENTER THE IMPELLER.

Figure 13. Flow Trajectory in a Vaneless Diffuser.

used vaned diffusers employ wedge-shaped vanes (vane islands) or thin-curved vanes. The latter type of vane is illustrated in Figure 14. In high head stages there can be two to four stages of diffusion. These usually are vaneless spaces to decelerate the flow, followed by two or three levels of vaned blades in order to prevent build-up of boundary layer, thus causing separation and surging the unit. Figure 14 indicates the flow pattern in a vaned diffuser. The vaned diffuser can increase the efficiency of a stage by two to four percentage points, but the price for the efficiency gain is generally a narrower operating span on the head-flow curve with respect to both surge and stonewall. Figure 14 also shows the effect of off-design flows. Excessive positive incidence at the leading edge of the diffuser vane occurs when α_2 is too small at reduced flow, and this condition brings on stall. Conversely, as flow increases beyond the rated point, excessive negative incidence can cause stonewall [6]. Despite its narrowing effect on the usable operating range on the characteristic curve, the vaned diffuser has its application in situations where efficiency is of utmost importance. Although seldom used, movable diffuser vanes or vane islands can be used to alleviate the shock losses at off-design conditions. However, as the adjusting mechanisms required are quite complicated, they generally are applied only to single-stage machines.

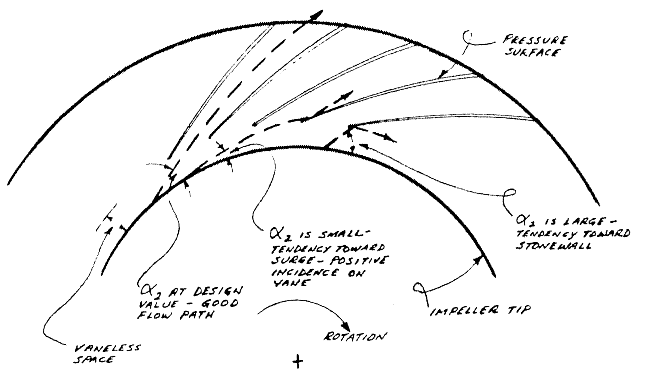


Figure 14. Vaned Diffuser.

It should be noted that the illustrations of the flow paths in Figures 12 through 14 are somewhat simplistic. Each flow path is indicated by a single streamline. The actual flow field is far more complex, with flow separation and recirculation present. Nevertheless, these figures should help with a practical understanding of the effects of changes in velocity triangles. Figure 15 [7] presents a visualization of the flow between two blades in the radial portion of an impeller with radial blades. This figure illustrates the complexity of the flow field as stalling begins. Illustrating the flow field in three dimensions would, of course, be far more complex.

Stationary guide vanes direct the flow to the eye of the impeller in an orderly fashion. Depending upon the head requirements of an individual stage, these vanes may direct the flow in the same direction as the rotation or tip speed of the wheel, an action known as positive pre-swirl. This is usually done to reduce the relative mach number entering the inducer, in order to prevent shock losses. This, however, reduces the head delivered but improves the operating margin. The opposite action is known as counter-rotation or negative pre-swirl. This increases the head delivered but also increases the inlet relative mach number. Negative pre-swirl is rarely used, since it also decreases the operating range. Sometimes the guide vanes are set at zero degrees of swirl; these vanes are called radial guide vanes. The theoretical effect of these settings is shown in Figure 16 [5]. Movable inlet guide vanes are

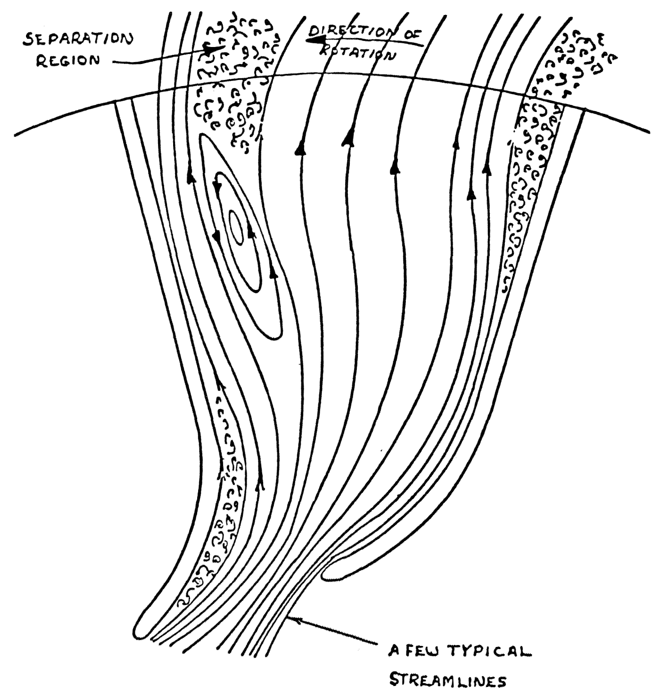


Figure 15. Impeller Flow Map.

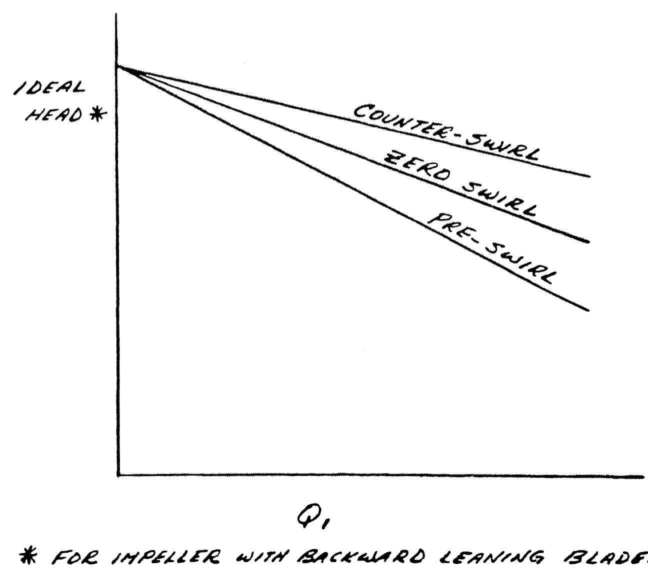


Figure 16. Effect of Guide Vane Setting (Stationary or Variable).

occasionally employed on single-stage machines, or on the first stage of multi-stage compressors driven by electric motors at constant speed. The guide vane angle can be manually or automatically adjusted while the unit is on stream to accommodate off-design operating requirements. Because of the mechanical complexity of the adjusting mechanism and physical dimensional limitations, the variable feature can only be applied to the first wheel in almost all machine designs. Hence, the effect of changing vane angle is diluted in the stages downstream of the first. Although the flow to the entire machine is successfully adjusted by moving the first stage vanes, the remaining stages must pump the adjusted flow at a fixed guide vane angle. A typical performance map of a constant-speed compressor with variable inlet guide vanes is represented in Figure 17.

Incidentally, a butterfly throttle valve in the suction line to the machine will produce nearly the same effects as moving the first stage guide vanes. However, throttling is not as efficient as moving the guide vanes, so that in many cases the added cost of the movable vane mechanism can be justified by horsepower savings.

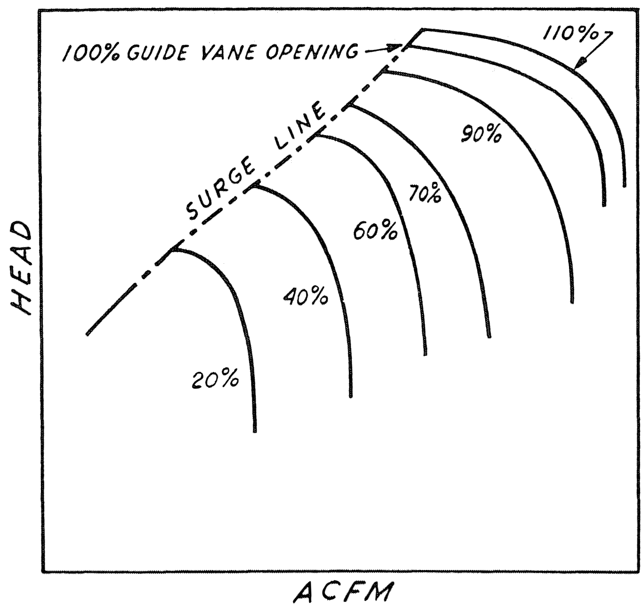


Figure 17. Constant Speed Machine With Variable Inlet Guide Vanes.

Effects of Gas Composition

Figure 18 [6] shows the performance of an individual stage at a given speed for three levels of gas molecular weight. The heavy gas class includes gases such as propane, propylene, and standardized refrigerant mixtures. Air, natural gases, and nitrogen are typical of the medium class. Hydrogen-rich gases found in hydrocarbon processing plants are representative of the light class.

The following observations can be made with respect to the curve for heavy gas:

1. The flow at surge is higher.
2. The stage produces slightly more head than that corresponding to medium gas.
3. The right-hand side of the curve turns downward (approaches stonewall) more rapidly.
4. The curve is flatter in the operating stage.

It is the last point (4) that often presents a problem to the designer of the antisurge control system. It should be noted that the flatness gets worse as stages are added in series. Since the RTS is small, there is a large change in Q corresponding to a small change in H. The control system, therefore, must be more responsive. It should be obvious that curves for lighter gases have a more desirable shape.

External Causes and Effects of Surge

The following are some of the usual causes of surge that are not related to machine design:

1. Restriction in suction or discharge of system.
2. Process changes in pressure, temperatures or gas composition.
3. Internal plugging of flow passages of compressor (fouling).
4. Inadvertent loss of speed.
5. Instrument or control valve malfunction.
6. Malfunction of hardware such as variable inlet guide vanes.
7. Operator error.
8. Maldistribution of load in parallel operation of two or more compressors.
9. Improper assembly of compressor, such as a mispositioned rotor.

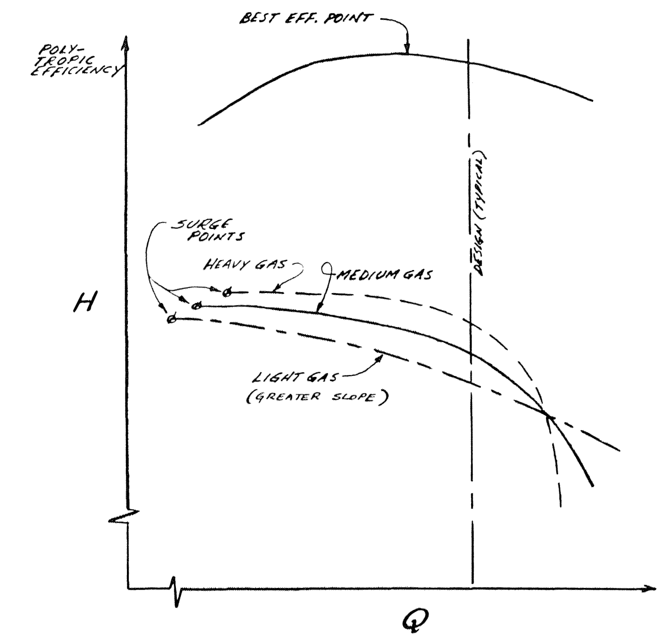


Figure 18. Effect of Gas Composition.

The effects of surge can range from a simple lack of performance to serious damage to the machine or to the connected system. Internal damage to labyrinths, diaphragms, thrust bearing and rotor can be experienced. There has been a reported case of a bent rotor caused by violent surge. Surge often excites lateral shaft vibration, and could produce torsional damage to such items as couplings and gears. Externally, devastating piping vibration can occur, causing structural damage, shaft misalignment, and failure of fittings and instruments.

The effects of the size and configuration of the connected system, as well as different operating conditions, on the intensity of surge can be astonishing. For example, a compressor system in a test set-up at the factory may exhibit only a mild reaction to surge. At the installation, however, the same compressor with a different connected system may react in a tumultuous manner. Section B describes several important piping and valving considerations. Surge can often be recognized by check valve hammering, piping vibration, noise, wriggling of pressure gauges or an ammeter on the driver, or lateral and/or axial vibration of the compressor shaft. Mild cases of surge sometimes are difficult to discern.

B. PRACTICAL ASPECTS OF SURGE CONTROL SYSTEMS

Surge Detection and Control

A surge control system consists of a set of detection devices (located around the compressor), which anticipate the surge, and control devices, which act to prevent surge from occurring.

Surge detection devices may be broken into two groups: static and dynamic. Static surge detection devices are those which attempt to avoid stall and surge by the measurement of certain compressor parameters and thereby ensure that a pre-decided value is not exceeded. When a parameter meets or exceeds the limit, some control action is taken. To this date, static surge detection devices have been widely used.

It is important to note that surge or stall itself is not measured by present day instrumentation. This is the reason for introduction of dynamic surge control concepts, which measure the actual onset of surge. It is probably the dynamic detection device that will meet the requirements and hopes of many engineers for a control device that can detect incipient surge and hence prevent its further development. However, much research on this type of device still is needed.

Before energy became such a critical factor, the allowance margin between actual physical surge and control operation was adjusted as required for all of the practical problems encountered. Very little motivation existed to narrow the margin. Even with the energy waste associated with wide allowance margins, very few operators would be willing to sacrifice reliability to narrow the margin. The challenge to the modern designer is to save energy without sacrificing control reliability. While analog control systems presently are common, they are, and will be, displaced by more modern microprocessor-based digital systems. However, many analog systems are in service and will continue to be used.

An anti-surge control system must be used to prevent surge by maintaining the minimum flow at a value safely away from the capacity at which surge occurs. This is accomplished by allowing some gas to recirculate through an anti-surge valve and recycle line from the compressor discharge back to its inlet. For air, and occasionally for other contaminant-free gases (O_2, N_2, etc.), the anti-surge valve vents gas to the atmosphere for surge prevention. Figure 19 illustrates surge and control points B and C, and process operating points A and D at 100% discharge pressure. The anti-surge control maintains 80% flow through the compressor, even though the process flow requirement is less than 80%. For example, if the process requires only 60% flow (Point A), the anti-surge control maintains 20% flow through the recycle line. Flow through the compressor is equal to the process flow (60%) plus the recycle flow (20%), or 80%, as illustrated by Figure 20. Recycle flow would, of course, be zero whenever the process was using 80% flow or more. The high energy cost for compression would justify minimizing recycling, when operating at reduced throughput, by using a variable speed drive which, in turn, complicates the anti-surge control system.

System surge lines are affected by changes in molecular weight, compressor fouling, changes in inlet conditions, and speed of the unit. Therefore, in many cases the preset value need not be exceeded before the compressor will surge. Optimum design entails considerably more than selecting primary flow elements, control valves, transmitters, and panel instruments, and integrating them into a control scheme based on a well-defined standard for designing antisurge control systems. Unfortunately, there is no so-called "universal best method" of anti-surge control. Before a control scheme is developed and components are selected, the operating requirements of the compressor and process must be known and evaluated. There are many variations in compressor and process characteristics; therefore, a control scheme that is ideal for one application might be far from adequate for another.

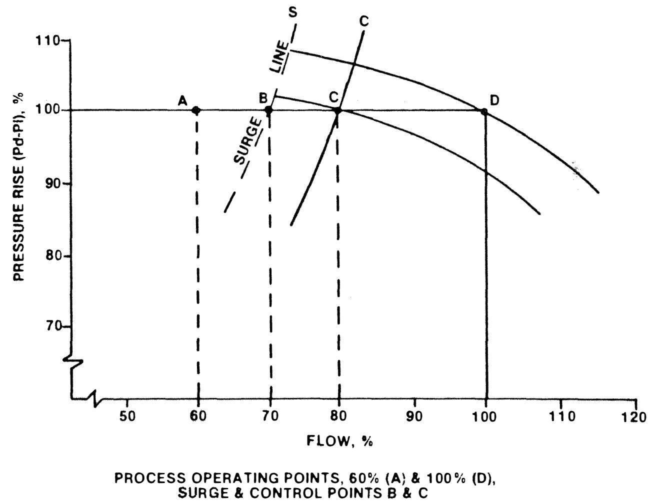


Figure 19. Process Operating Points, 60% (A) and 100% (D), Surge and Control Points B and C.

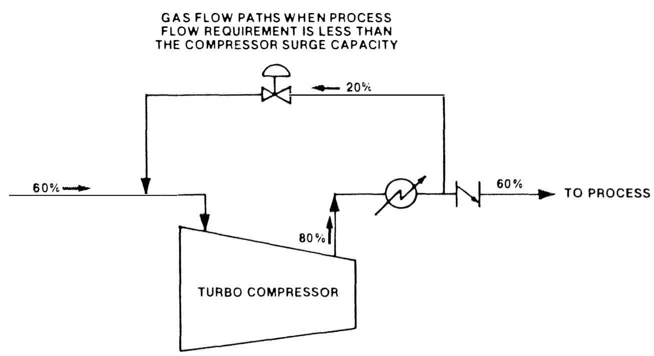


Figure 20. Gas Flow Paths When Process Flow Requirement Is Less than the Compressor Surge Capacity.

Methods of Capacity Control of Compressors

The two types of turbo compressors are centrifugal and axial flow. Methods of capacity control vary, depending on the type of compressor and its driver. This discussion will be limited to the following methods, which are most commonly used:

Type Driver	Type Compressor	Method of Capacity Control
Constant Speed	Centrifugal	Suction Throttle
Variable Speed	Centrifugal	Variable Speed
Constant Speed	Axial Flow	Variable Angle Stator Vanes
Variable Speed	Axial Flow	Variable Speed

Typical performance curves and surge lines for the above are shown in Figure 21. It must be emphasized that the word "typical," used to describe these curves, means just that. The actual surge line of many compressors is significantly different from the typical surge line illustrated. For example, instead of the typical shape (Figure 21b), the actual surge lines of many centrifugal compressors are shaped more like the surge line of an axial flow machine (Figure 21c or 21d).

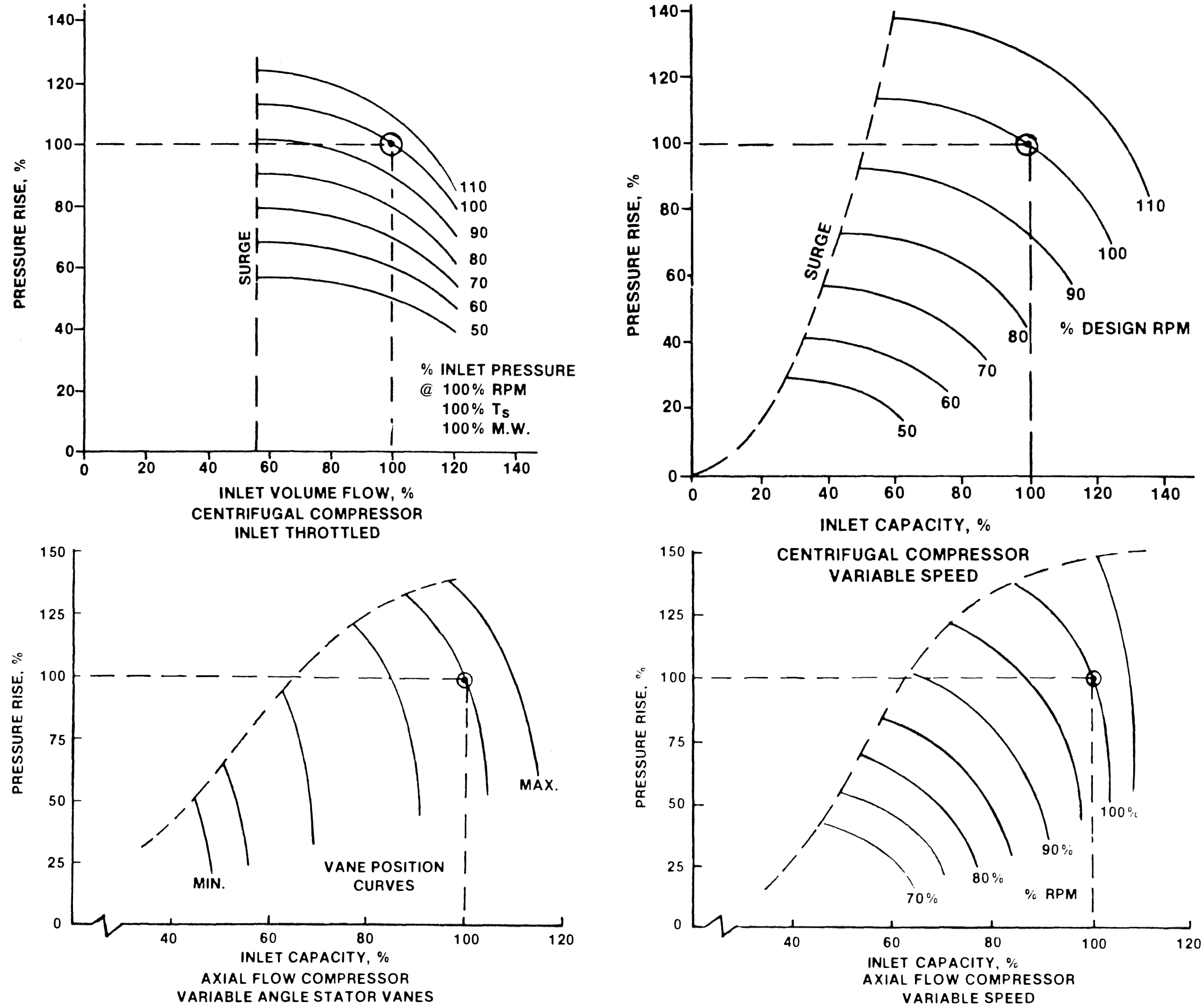


Figure 21. Head-Capacity Characteristics of Compressors.

Constant Speed Centrifugal Antisurge Control

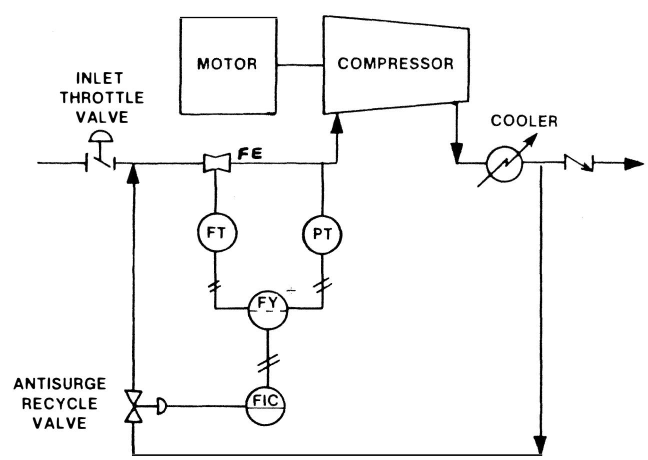
For this type machine, Figure 21a shows that surge occurs at a constant inlet volume flow rate, regardless of the inlet pressure. Therefore, inlet volume flow rate would be an appropriate measurement for antisurge control. Instruments can be used as shown schematically in Figure 22 to solve Equation 1.

$$Q = C' \sqrt{\frac{h}{P_i}} \tag{1}$$

where

- Q = volume flow rate at inlet conditions, (acfm or m^3/hr)
- h = head, inches of water differential produced by flow element (FE)
- P_i = inlet pressure, absolute
- C' = proportionality constant

Temperature compensation is not used in the control scheme (Figure 22); therefore, if inlet temperature increases,



ANTISURGE CONTROL SCHEME, INLET VOLUME FLOW

Figure 22. Antisurge Control Scheme, Inlet Volume Flow.

the controlled flow will be lower than the setpoint. Actual flow can be calculated by Equation 2:

$$Q_{actual} = \text{Setpoint Flow}\sqrt{T_{design}/T_{actual}} \qquad (2)$$

where

T = degrees Rankine or Kelvin

A decrease in mole weight will also result in a lower controlled flow, in accordance with Equation 3.

$$Q_{actual} = \text{Setpoint Flow}\sqrt{MW_{actual}/MW_{design}} \qquad (3)$$

These flow measurement errors are actually desirable, because gas temperature or mole weight variations also cause the surge point of most compressors to shift in the same direction. The surge point usually shifts more than the control point; therefore, temperature or mole weight variations normally change the safety margin (distance between surge and controlled flow points). Small to moderate gas variations normally can be accommodated by a setpoint which provides an adequate safety margin at the lowest gas temperature or highest mole weight condition. To handle extreme variations, it may be necessary to change the controller setpoint. The flow measurement in Figure 22 is pressure compensated. Therefore, the system will control a constant volume flow, regardless of the inlet pressure. This feature is needed because surge occurs at the same volume flow rate, regardless of the inlet pressure.

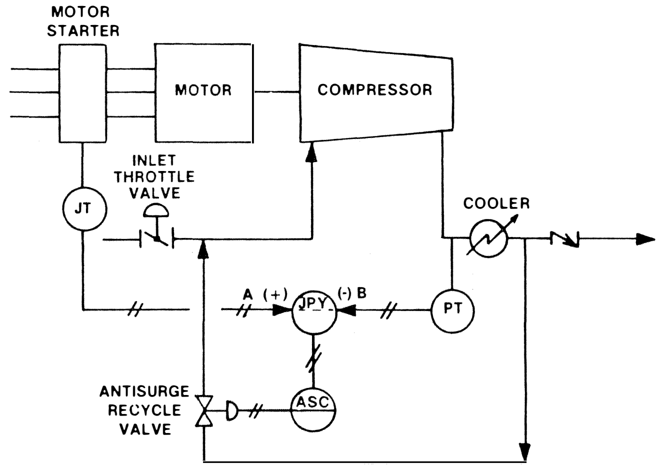
The inlet flow control system shown in Figure 22 includes some undesirable features. A relatively long inlet line is required to provide sufficient straight run of pipe between the throttle valve and flow element. Also, the flow element (FE) is relatively expensive, and it produces a pressure loss that results in slightly greater operating costs. These disadvantages can be eliminated by an antisurge control scheme based on measurements of power and discharge pressure, as shown in Figure 23. Antisurge control by this method is facilitated by the following characteristics of a constant speed, inlet throttled compressor:

1. At the surge point, the following parameters are unaffected by suction pressure variations:
   - Volume flow (acfm, m^3/hr, etc.)
   - Head, polytropic (Hp)
   - Ratio of compression (R_c); $R_c = P_d/P_i$
2. Power is proportional to weight flow (W), which in turn is proportional to inlet pressure (P_i).
3. P_d is proportional to P_i; i.e, $P_d = R_c \times P_i$

Based on the above, surge is a straight line relationship of P_d vs. W, or P_d vs. J, as illustrated in Figure 24. As with the inlet flow system previously described, this J vs. P_d system also is subject to some error if inlet temperature or mole weight changes. For example, a temperature decrease at constant inlet pressure will cause surge to occur at higher volume flow, higher discharge pressure, and higher power.

Variable Speed Centrifugal Antisurge Control

The "Flow/Delta-P" antisurge control system is widely used with variable speed centrifugal compressors. The name is derived from the fact that control action is based on measurements of compressor inlet flow and differential pressure ($P_d - P_i$). Two Flow/Delta-P type control schemes are illustrated by Figures 25 and 26. The controller setpoint in the typical system shown in Figure 25 is manipulated by the "$P_d - P_i$" signal, and the flow signal is the process input to the



CENTRIFUGAL COMPRESSOR ANTISURGE CONTROL, J VS. Pd

Figure 23. Centrifugal Compressor Antisurge Control, J vs. P_d.

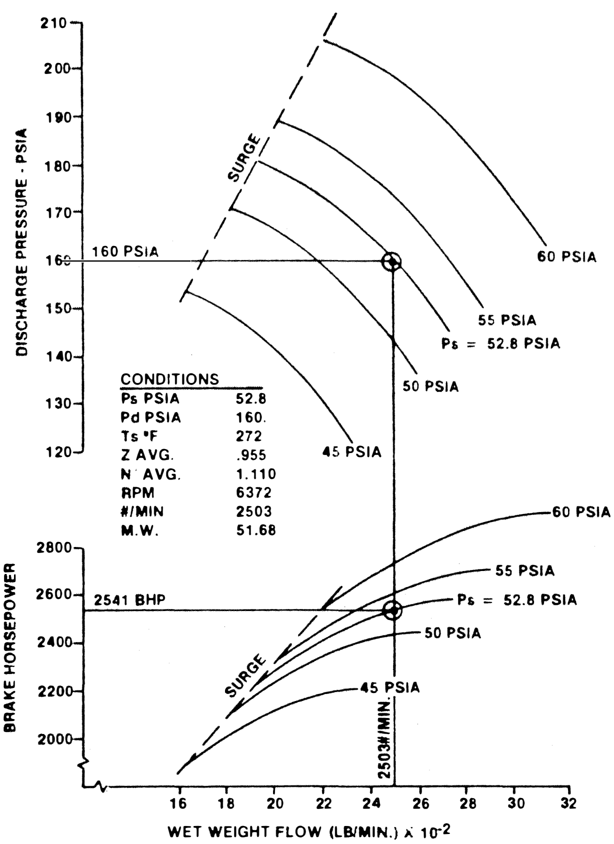


Figure 24. Wet Weight Flow vs. Brake Horsepower and Discharge Pressure.

controller. In the alternate system shown in Figure 26, the process signal to the controller is a value which represents the compressor operating point ($P_d - P_i$ vs. flow signal), and a constant controller setpoint is used. More complete information is provided in References 8 and 9.

The theory of the Flow/Delta-P system is based on the following flow element and compressor surge line characteristics.

1. Differential (h) produced by the flow element is proportional to flow squared.
2. Pressure rise ($P_d - P_i$) produced by the compressor is proportional to rotative speed squared.
3. The flow rate at which surge occurs is proportional to rotative speed.

Table 1 illustrates the above theory. For the purpose of this illustration, 100% pressure rise ($P_d - P_i$) and 100% flow are used for the surge point at 100% speed. Thus, at 90% speed; % $P_d - P_i = 100\ (90/100)^2 = 81\%$; % flow signal $= 100\ (90/100)^2 = 81\%$. Therefore, at 90% speed the setpoint will be 81%, and flow will be controlled at 90%.

The Flow/Delta-P type control scheme is widely used and in many cases is the most suitable method of antisurge control for centrifugals. Often, however, the system has to be modified to suit particular needs.

Table 1. Surge Line Values.

Speed	*Flow Rate*		$P_d - P_i$	
(%)	*(%)*	*(% Signal)*	*(%)*	*(% Signal)*
100	100	100	100	100
90	90	81	81	81
80	80	64	64	64
70	70	49	49	49

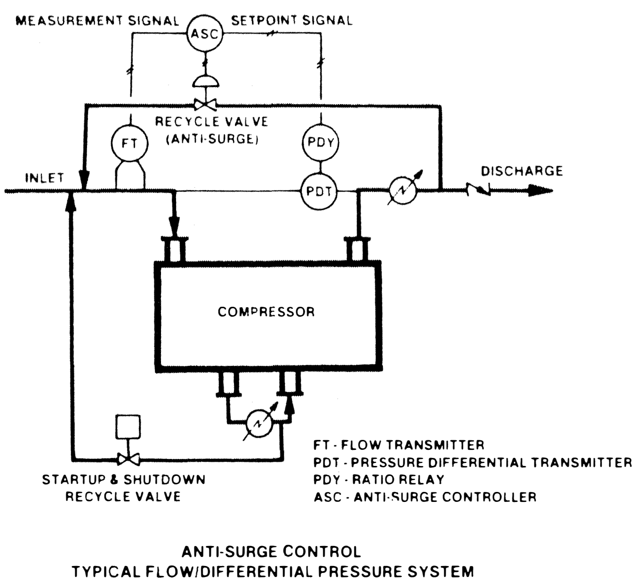


Figure 25. Typical Flow/Differential Pressure Type Anti-Surge Control System.

Discharge Volume Flow (P and T Compensated) Antisurge Control, Variable Speed Compressor

A pressure and temperature compensated discharge volume flow antisurge control system is shown in Figure 27. This particular centrifugal operated in series with an axial flow compressor. Because of very large interstage coolers, separators, piping and space limitations, it was deemed impractical to use an inlet flow element, so alternate methods were investigated. Calculated values of various parameters (i.e., weight flow, inlet and discharge volume flow, pressure ratio, power, etc.) at the surge point were examined for various operating speeds. Calculated discharge volume flows for all surge points, from 75 to 95% speed, varied less than ±1%. Thus, a careful examination of various calculated parameters indicated discharge volume flow to be not only the most practical but also a very effective variable to use for surge prevention. Field surge tests of the actual machine verified the calculated predictions. Surge and control lines are shown in Figure 28 on the compressor performance curve. The flow measurement is temperature compensated; therefore, the control line does not change with temperature variations. In this case, temperature compensation decreased the control error at speeds above 95%, but slightly increased the error at speeds below 95%. Because of the benefit at normal speeds, temperature compensation was justified. With this system, the inlet flow element was eliminated, the discharge flow element was smaller and less expensive, the pressure loss due to the flow element was less significant at the discharge, and a conventional flow computing system was used.

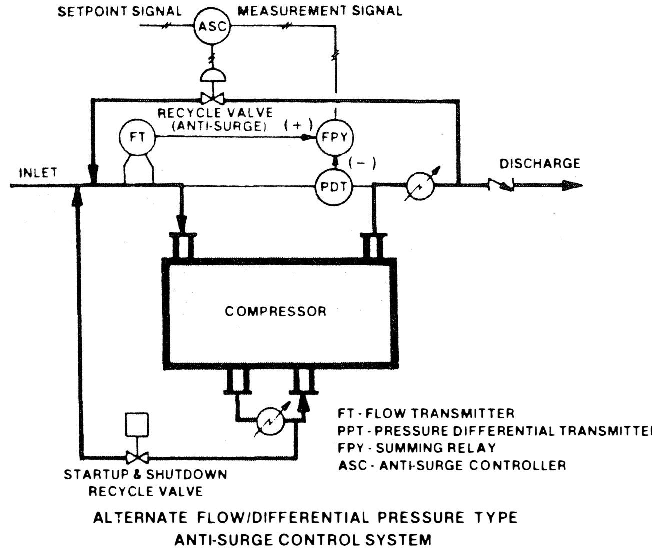


Figure 26. Alternate Flow/Differential Pressure Type Anti-Surge Control System.

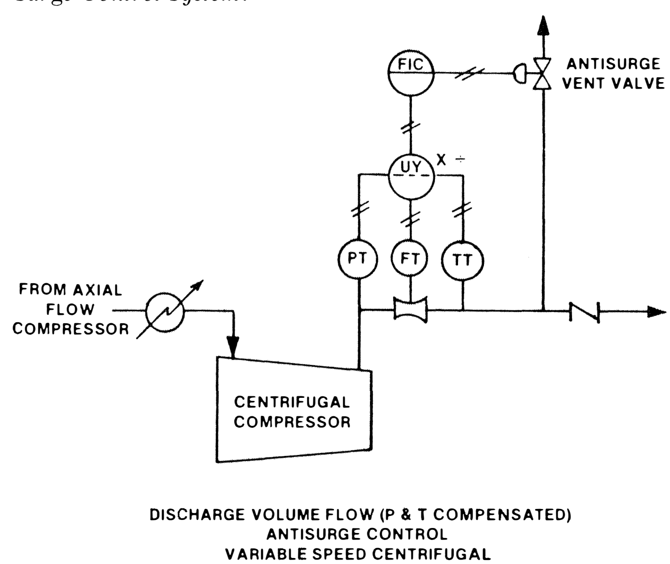


Figure 27. Discharge Volume Flow (P and T Compensated) Antisurge Control, Variable Speed Centrifugal.

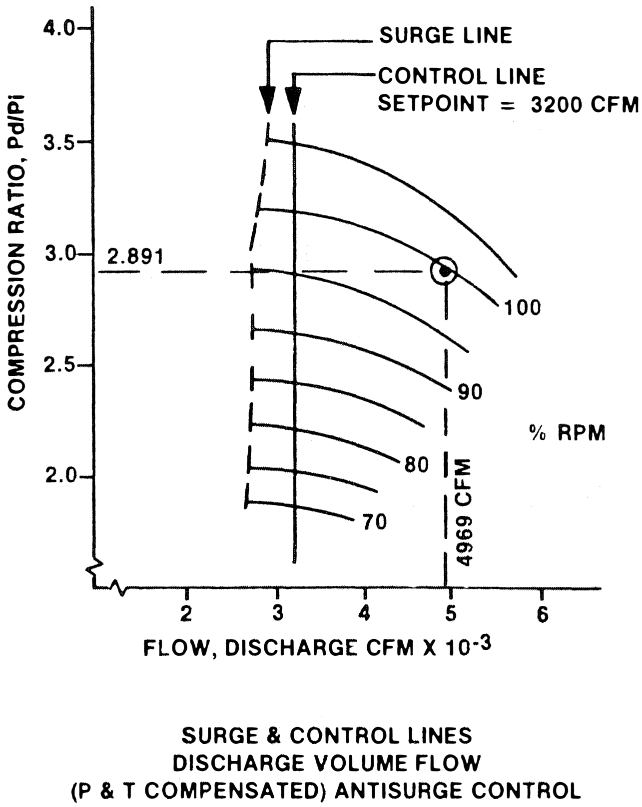


Figure 28. Surge and Control Lines, Discharge Volume Flow (P and T Compensated) Antisurge Control.

Discharge Volume Flow (Non-Compensated) Antisurge Control, Variable Speed Compressor

A similar system, but without pressure or temperature compensation, is shown in Figure 29. The surge and control line curves for the compressor and control system are shown in Figure 30. The system is very simple, yet it provides very effective control. Several things account for the surge and control lines having a similar slope over the normal 85 to 105% speed range:

1. Flow units at normal conditions (i.e., 1.03 kg/cm^2 absolute and 0°C) are used.

$$\mathrm{nm^3/hr} = C' \sqrt{\frac{h \times p}{T}} \tag{4}$$

   In this case, the system measures only the differential, head (h) across the orifice. The controller maintains "h" constant at its setpoint value. Therefore, the controlled flow (nm^3/hr) varies in proportion to P/T. Discharge pressure and temperature decrease with speed. On a percentage basis, discharge pressure decreases much faster than temperature, so the slope of the control line is primarily a function of P_d.

2. The capacity of the compressor relative to its compression ratio influences the slope of the surge line. In this case, the relationship is such that the surge line slope is very similar to the flow vs. P_d slope of the control line (in the normal speed range). By comparison, the compression ratio of this machine is approximately four times that of the compressor used for the preceding example, but the flow capacity is lower by a factor of eight. The antisurge control scheme shown in Figure 29 is not always best suited for all variable speed machines, but, because of its simplicity and effectiveness, it should always be considered.

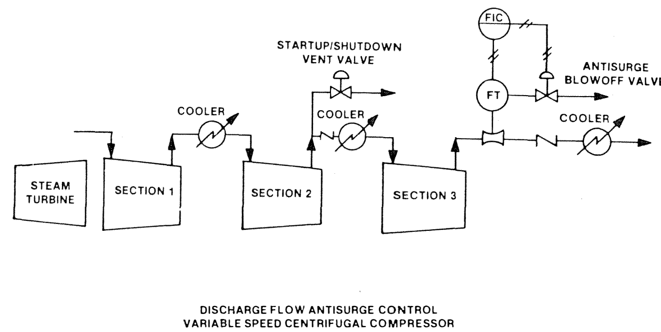


Figure 29. Discharge Flow Antisurge Control, Variable Speed Centrifugal Compressor.

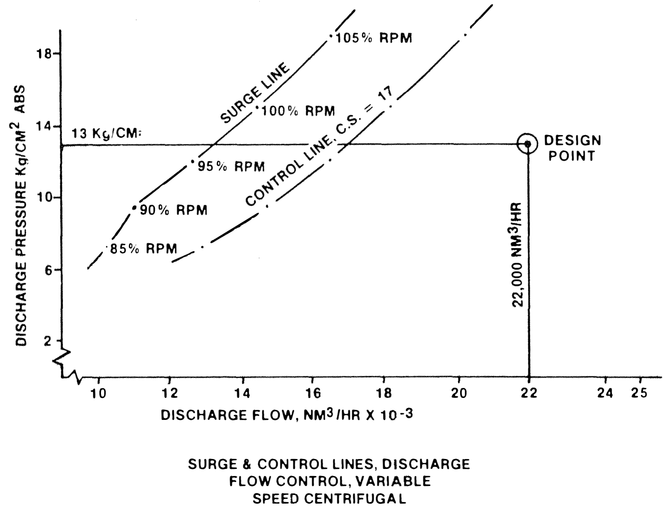


Figure 30. Surge and Control Lines, Discharge Flow Control, Variable Speed Centrifugal.

Flow/Delta-P Antisurge Control

The results of a Flow/Delta-P type control are illustrated for normal and startup conditions in Figures 31 and 32. The startup gas is more than twice as heavy as the normal gas (12.60 M.W. at startup vs. 5.97 normal). These two illustrations show the effect that a heavier gas has on the compressor, plus the inherent self-compensating effect of the Flow/Delta-P control scheme. At any given speed, the machine produces a much higher discharge pressure on the heavier gas (Figure 32), and at 70% speed it produces more pressure than on normal gas at 100% rpm.

To compare the relative control safety margins for the two gases, consider the surge and control points at 12.5 kg/cm^2 discharge pressure (4.31 $P_d - P_i$). When operating on normal gas (Figure 31), surge occurs at 5900 m^3/hr and 100% speed. Control line "N" at this point is 6800 m^3/hr, or a safety margin of 900 m^3/hr.

At the heavy gas condition (Figure 32) and at the same discharge pressure (12.5 kg/cm^2) and $P_d - P_i$, surge occurs at 4000 m^3/hr and approximately 68% speed. Control line "N" at this point is 4700 m^3/hr, or a lower safety margin of 700 m^3/hr. Control line "N" is produced by the alternate control system of Figure 26 with a setpoint of 6500 m^3/hr. This setpoint could be considered the normal setpoint, and a higher setpoint of 7000

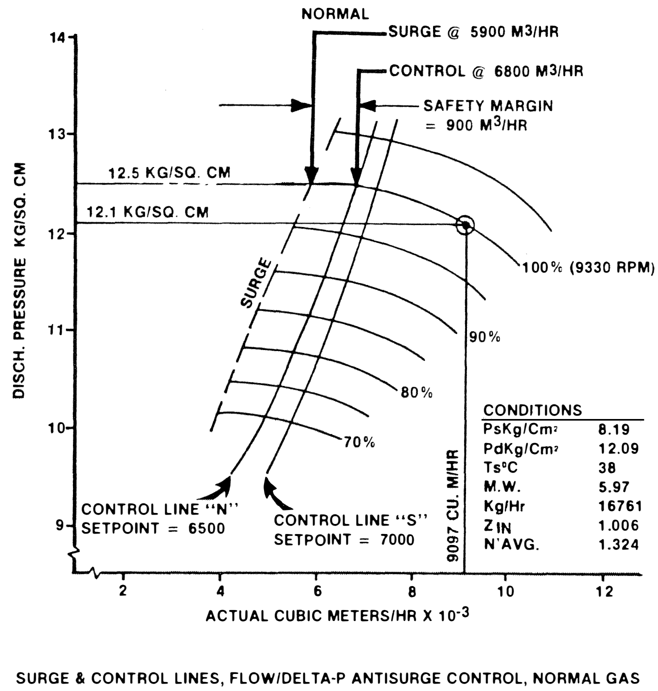


Figure 31. Surge and Control Lines, Flow/Delta-P Antisurge Control, Normal Gas.

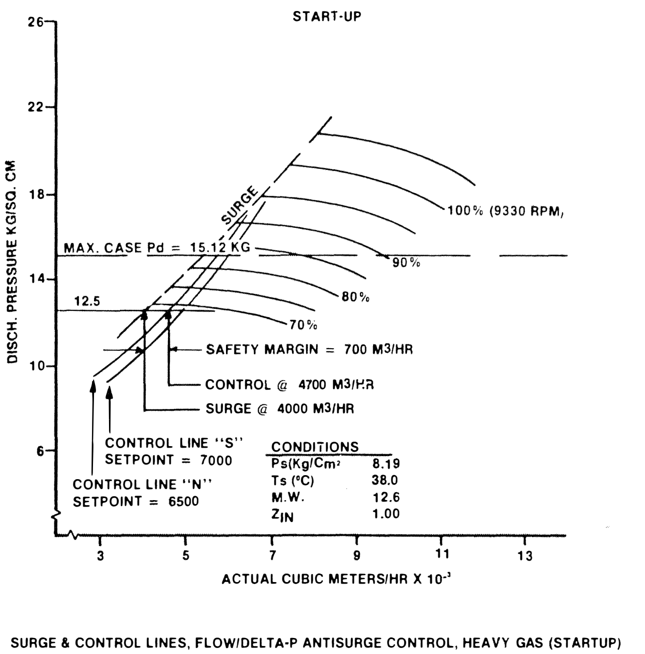


Figure 32. Surge and Control Lines, Flow/Delta-P Antisurge Control, Heavy Gas (Startup).

m^3/hr used to produce a greater safety margin during startup (control line "S"). Alternatively, if it is possible to operate at a setpoint of 7000 m^3/hr without causing inefficient operation at normal conditions, then such a setpoint would be practical for all operating conditions.

For the compressor used in this example, the control safety margin decreased as the mole weight increased. Also, it would decrease with a decrease in inlet temperature. The reader should be cautioned against drawing the wrong conclusion from this example. More often, the reverse actually is true; i.e., the safety margin increases due to a heavier gas or lower temperature. The procedure used to calculate the control lines is provided in the Appendix.

The Flow/Delta-P control is also very practical and effective for a constant speed, inlet throttled compressor. Depending on how the system is calibrated, errors due to inlet pressure variations will be small or non-existent. If the system calibration is such that the control line originates at zero flow and Delta-P ($P_d - P_i$), the error will be zero. This is explained by the following characteristics of the flow element and a constant speed compressor:

1. Inlet pressure variations do not affect the volume flow or the compression ratio at which surge occurs. The control, therefore, must maintain a constant volume flow regardless of inlet pressure.
2. At a constant volume flow, the differential head produced by the orifice varies directly with pressure, as explained by rearranging the standard Equation 5 for volume flow to Equation 6.

$$Q = C' \sqrt{\frac{h \times T}{p}} \tag{5}$$

or

$$h = \left(\frac{Q}{C'}\right)^2 \frac{P}{T} \tag{6}$$

3. Delta P ($P_d - P_i$) varies directly with inlet pressure in accordance with Equation 7.

$$P_d - P_i = P_i\,(R_c - 1) \tag{7}$$

Taking the above into account, at 50% P_i,

a. $P_d - P_i = 0.5\,(R_c - 1) = 50\%$ Setpoint.

b. $h = \left(\frac{Q}{C'}\right)^2 \frac{0.5}{T} = 50\%$

c. 100% volume flow is required for the orifice to produce 50% differential at 50% pressure, by Equation 6. Therefore, the system will, in fact, control 100% volume flow, regardless of the inlet pressure.

Modes Of Operation (Startup, On-Stream, Shutdown)

The complete antisurge control system must effectively prevent surge during startup and shutdown, as well as during on-stream operation. Discrete signals are used to override the analog control system and manipulate valves during startup and shutdown. As when designing the analog system, it is equally important to properly evaluate the compressor and process requirements when designing the startup and shutdown logic system.

Startup Control Action

For startup, it has been common practice to maintain the antisurge control valve fully open during most of the acceleration period. Typically, the logic override signal is removed at 80 or 85% speed to transfer control to the analog system for on-stream operation. This introduces a step control change. The immediate action by the antisurge controller is an attempt to drive the valve fully closed. The valve, moving toward its closed position, brings the compressor operation to the point where the antisurge controller must take over control and modulate the valve in a partially open position for stable compressor operation. If this recovery action is not fast enough, the valve will close too far, and the compressor will surge. If this is a problem, the system must include a feature to smooth the transfer action and to give the controller sufficient time to recover control.

Instead of activating the antisurge controller at 80 or 85% speed, it is better to have it in control during the entire acceleration period whenever possible. To facilitate this, the surge control line must never intersect the surge line. For example, the control line for a setpoint of 6500 m^3/hr (Figure 31) is always on the right side of the surge line. According to general practice, Figure 31 shows surge and control lines for the normal operating range only, but at zero Delta-P (i.e., P_d and P_s = 8.19 kg/cm^2), the control point is 2247 m^3/hr when calculated by Equation A1 of the Appendix. Therefore, the analog system should be in control throughout the startup, and a transfer at 80 or 85% speed is not necessary. However, for the control described for Figure 30, a startup override to open the valve at low speed would be necessary.

Another factor that must be considered is the fact that the stages of a multistage compressor become mismatched when operated at less than normal speed. Under some conditions, the compressor could surge even though the anti-surge valve is fully open. Usually, surge is initiated by a stage near the discharge when the machine is operating at rated speed. At lower speeds, a stage closer to the inlet, frequently the first stage, initiates surge. The reason for this is that at lower speeds, the gas volume is not reduced by each stage as much as it is reduced at rated speed. Consequently, at reduced speed, the inlet volumetric flow rate through the final stages is greater than their rated flow, and the volume entering stages near the inlet tends to become less than their rated flow.

At high flow rates, the gas velocity can reach the speed of sound at some location in the compressor. Since the maximum velocity of any gas is limited by its sonic velocity, this imposes a maximum flow limit on the compressor. This condition seldom occurs while operating at rated speed. However, when the speed of a multi-stage compressor is reduced, a speed finally is reached where it is no longer possible to keep the compressor out of surge by recycling or venting gas from the final discharge. The reason is that some stage near the discharge has reached its choked (sonic) flow limit, which backs up the flow in the earlier stages to the point where one stage reaches its surge point. To further explain what happens, assume a six-stage compressor with a rated inlet volume flow of 10,000 acfm. Also assume that, at rated speed, each stage makes a compression ratio of 2.0, and surge occurs at approximately 7700 acfm at 100% speed. Disregarding the effect of increased temperature, the discharge volume from Stage 1 therefore is 5000, which is the inlet volume to Stage 2. If we carry this on through, we find the inlet volume flows to Stages 3 through 6 to be 2500, 1250, 625, and 313, respectively, as shown in Figure 33. Now let us assume that at 70% speed, each stage makes a compression ratio of only 1.5 and that Stage 6 reaches its choke limit at a volume flow of 600. Also assume that each stage will surge at 60% of its rated capacity when operating at 70% speed. The 1.5 volume reduction per stage means that each stage will have an inlet volume flow 1.5 times greater than its discharge volume.

If we start with the 600 maximum (choked) limit of Stage 6 and multiply back through all stages, we determine that the volume flows into Stages 5, 4, 3, 2 and 1 are 900, 1350, 2025, 3038 and 4556, respectively, as shown in Figure 34. The 4556 volume flow at the inlet to Stage 1 is considerably less than its 6000 minimum surge limit, or, in other words, choked flow in Stage 6 has forced Stage 1 to operate below its surge limit. This condition might occur at a speed in the vicinity of 80% to 90%

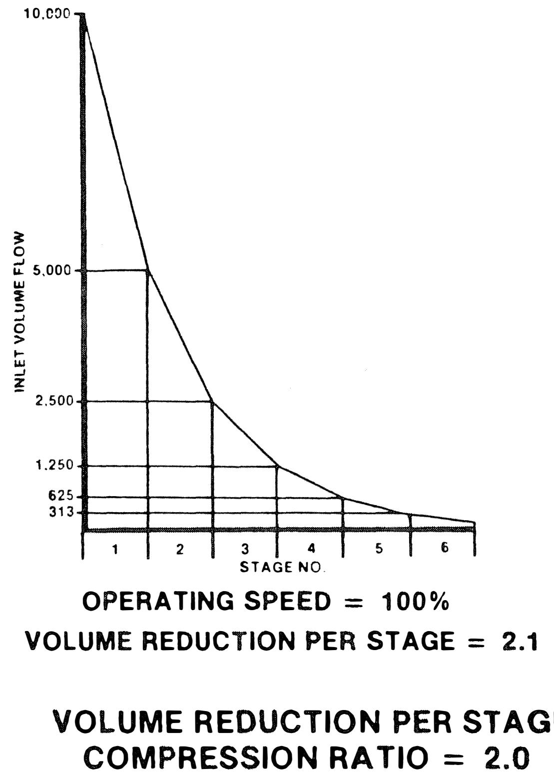


Figure 33. Volume Reduction Per Stage, Compression Ratio = 2.0.

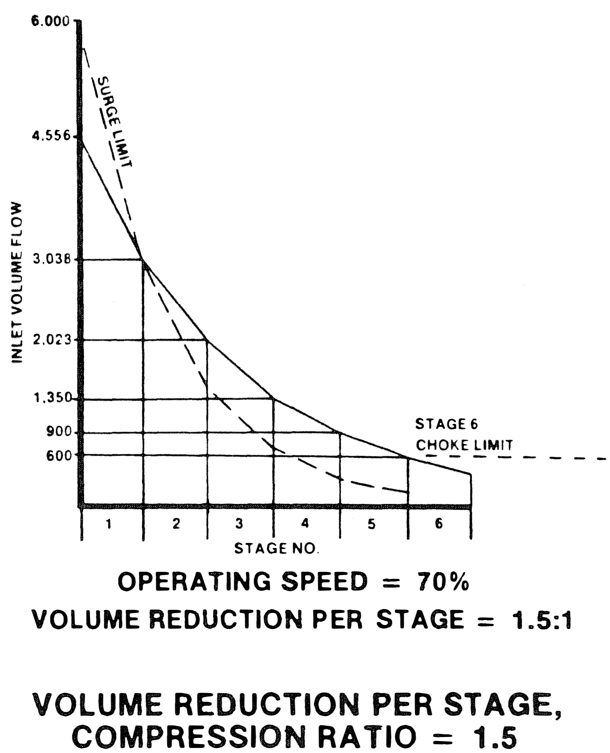


Figure 34. Volume Reduction Per Stage, Compression Ratio = 1.5.

on some compressors, and at much lower speeds on others. In order to accommodate this condition, a recycle valve at some intermediate stage (see startup and shutdown valve, Figure 26) must be open while the compressor is operating at reduced speed during startup or shutdown.

The type of gas handled is a factor in the condition just described. For a heavy gas, e.g., carbon dioxide, several valves located at intermediate stages may be required because its sonic velocity is higher. Also, a change in speed makes a greater difference in the volume reduction produced by each stage.

Stage mismatching also can alter the surge line shape. The shape can change with speed as well as gas variations. This is apparent in a comparison of Figures 31 and 32. The surge line on Figure 31 is slightly curved, but in the reverse direction from the surge line on Figure 32. The different shapes occur because Stage 5 initiates surge at all speeds on Figure 32, but on Figure 31 surge is initiated by Stage 5 at 90% speed and above, and by Stage 1 at 85% and lower speeds. On some machines, the stages which initiate surge shift several times within the operating range, and the reverse curvature is much greater than that in Figure 31. For the same compression ratio, an axial flow compressor has more stages than a centrifugal, so the surge initiating stage changes more times. This accounts for the reverse curvature typical of axial flow compressors. For a constant speed, axial flow machine the effect is less than for a variable speed axial, because only the stages with variable angle stators are affected. Typically, approximately 40% of the stages have variable stator vanes.

Shutdown Control Action

A very high potential for surge exists during an emergency shutdown, or any time the compressor is tripped while operating at normal conditions. Deceleration is very rapid, typically with speeds having decreased to 80% rpm approximately 1 to 1½ seconds after trip-out. The compressor's ability to produce pressure rise decreases approximately proportionally to speed squared. Thus, the need exists to lower the discharge pressure very quickly after trip-out; otherwise, severe surging can result. During deceleration, the recycle valve must handle all of the compressor flow, plus any excess gas accumulation in the discharge system. If there is an excessive quantity of gas compressed in the discharge system, the recycle valve must be capable of passing a very high flow rate during the first several seconds of the deceleration period in order to lower the pressure fast enough to prevent surge. The most important factor in preventing surge under these conditions is to have the least possible volume at the discharge. Sometimes this can be accomplished by locating a check valve as close as possible to the compressor discharge. However, in many systems the recycle takeoff point must be located downstream of a cooler and separator; this adds a considerable volume at the compressor discharge. In some cases it is possible to accommodate this by a fast opening (1 second or less), generously sized (125 to 200% of compressor rated flow) recycle valve. In other cases, it may be necessary to use an additional valve and check valve very close to the compressor discharge. If possible, this valve can vent gas to atmosphere. If atmospheric venting is not feasible, then the extra shutdown valve could dump hot gas back to the compressor inlet. The temperature rise from inlet to discharge decreases rapidly as speed decreases, so recycling hot gas back to inlet normally is not a problem during the relatively short coastdown time. It should also be noted that the high temperature effect of recycling hot discharge gas is often mitigated by the temperature drop (Joule-Thompson effect) resulting from the expansion across the recycle valve. This effect is significant, especially with natural gas at high pressures. If a temporary high inlet temperature is a problem for a particular shutdown characteristic, then the hot gas recycle valve could be closed approximately 10 to 15 seconds after tripout. For the rest of the coastdown period, the deceleration rate normally is slow enough that the normal recycle valve is able to prevent surge.

System Piping and Configuration Aspects

Piping and valving are important considerations in surge control system design. A very fundamental and important aspect is the sizing of the bypass line. When startup, shutdown, and normal operation are considered, there are a number of variations. Field experience has shown this straightforward task to be the cause of many field problems because the line was too small. Another consideration in line sizing is the dynamic or momentary requirements in the limiting type of control, such as the surge control system. Because steady state criteria would indicate a line sized to pass only the surge plus margin flow, there is a temptation to utilize the apparent economics of a smaller line size. In actual practice, the line should handle full compressor flow with a minimum pressure drop. Correct surge control valve selection and sizing is also of critical importance for achievement of good control sensitivity and a rapid response in the vicinity of surge. While other components (i.e., actuator, controller) also affect the overall control system response, valve sensitivity and response have the major impact.

The control valve is a key link in the control chain. In most cases, the valve should have linear flow and fast response. There is very little benefit in having a highly sophisticated controller set point computer coupled to a valve either too slow or too small to handle the problem. When the surge system is used for startup, a second valve may be required, as the "turn down" burden of a single valve may become too large. Frequently, surge control valves are selected too large, resulting not only in unnecessary cost but also in significant loss of control sensitivity, a loss that cannot be overcome through reduced signal sampling intervals, actuator changes, etc. Improper valve type selection can result in insufficient sensitivity and cause excessive noise, which usually limits valve durability.

When air is the compressed medium, the surge control valve normally can relieve to atmosphere, usually through an adequate silencer. If, however, the compressed medium is a process gas or a refrigerant, the valve will have to bypass the section being protected and return the flow to the proper suction nozzle. In such cases, some form of cooling is required to remove the heat of compression. A shell and tube exchanger or an air exchanger, as shown in Figure 35, may be required. The system shown in Figure 35a, which is a single recycle loop system, did not work well. This was especially true for high molecular weight operation where the low pressure (LP) casing was in surge while the high pressure (HP) casing was in stonewall. The situation was impossible to correct by opening the recirculation valve. It was impossible to start the system without mismatching the LP and HP casings during speed changes. The LP and HP signals to the recirculation valve also interacted and created a problem. The configuration shown in Figure 35b also was unworkable, because of reverse train rotation after trip. This was due to the large amount of energy stored in the discharge cooler. The emergency shutdown valve (ESD) was not fast enough to cope with this problem. Also, when the HP spill-back came open first, the LP casing went into surge. In addition, there was difficulty in maintaining system balance during start-up and loading. The preferred arrangement is shown in Figure 35c.

If the gas is sent through a condenser (as in the case of a refrigerant), direct contact cooling may be possible. On smaller units, a liquid level can be retained in the bottom of the suction

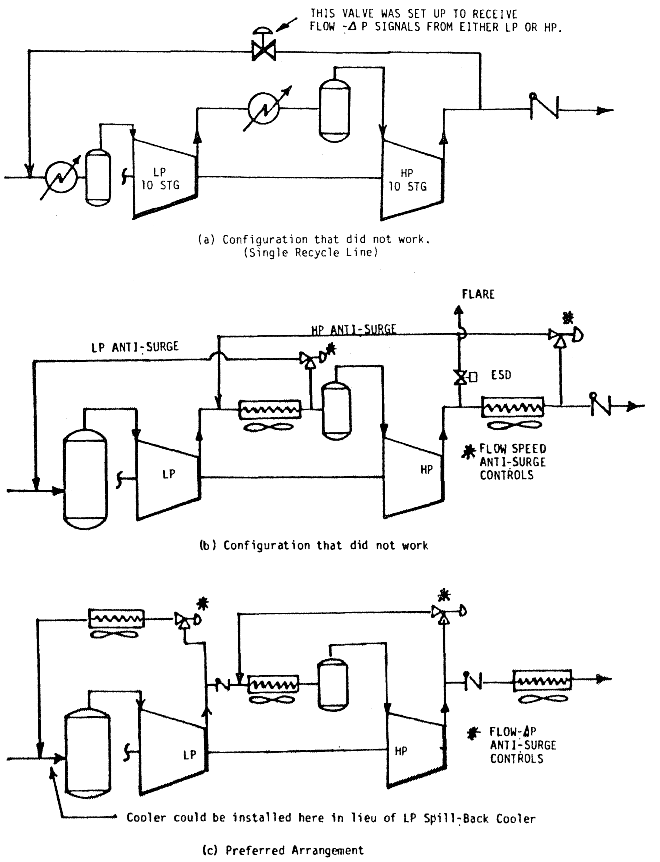


Figure 35. Equipment Arrangement.

drum with normal flow into the side above liquid level. The surge control by-pass flow passes through the bottom, contacting the liquid and thereby desuperheating. As the systems get larger, this method is not feasible, due to the shaking forces from the boiling liquid. For such service, a sparger or a modified steam desuperheater can be used, in which the liquid is sparged or sprayed into the bypass line ahead of the suction drum. A ring of temperature sensors monitors the spray and is set for the desired amount of residual superheat. Cooling a system on bypass can make startup on process gas much easier.

On multi-nozzle compressors or compressors with intercoolers, bypassing should be considered for individual sections. This will bypass the necessary flow to keep the affected stage out of surge, rather than waste energy by bypassing the entire compressor or train (if multibodied). The bypass scheme must be evaluated for both energy and stability, as there may be a tendency for some multiple loops to interact if proper care has not been exercised.

Figure 36 shows the location of a check valve in the system. Figure 36b is the required arrangement if parallel operation is anticipated. This system is also required if the compressor must start up against a stagnation pressure, as the pressure on the downstream side would keep the check valve from opening. Field experience and some experimental work [10] have shown some advantage to the control shown in Figure 36a, if the compressor is not in parallel operation and the downstream pressure would not prevent the check valve from opening. This configuration, however, is rather infrequently used in the petrochemical industry. The inherent stability of such a scheme is much greater and allows for more gain and responsiveness in the system, without the effort required for the scheme shown in Figure 36b.

Microprocessor based surge control systems can be made very sophisticated. Most of the variables external to the compressor are readily measured and can be digitized without problem or can be treated and combined in analog form. The signals may be processed either in analog form or digitally, with the latter having an advantage. For multiple component gas, measuring the molecular weight, "K" and "Z" is still somewhat of a problem. Sometimes a density meter can be used if the variations are not too great. This compensates for apparent molecular weight, but not "K" or "Z". There is no easy answer for this problem.

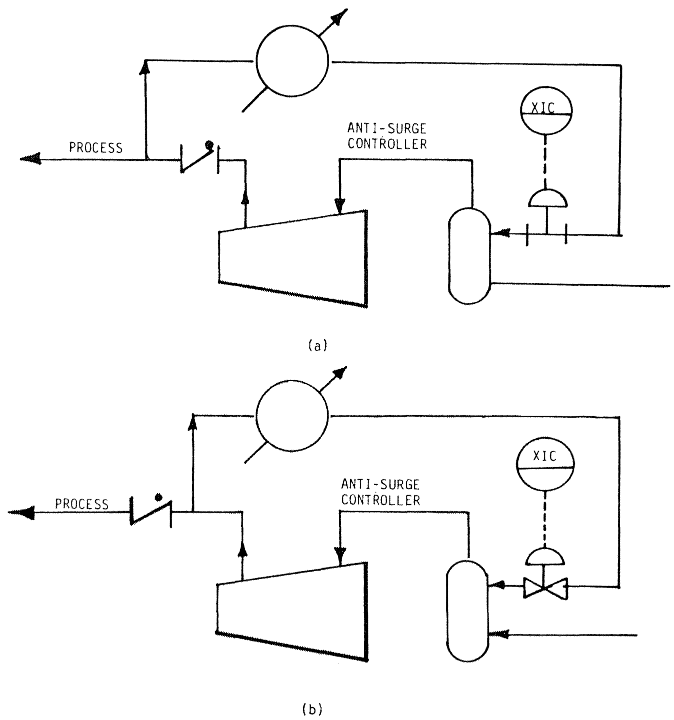


Figure 36. Location of Check Valve.

Response Time of Surge Control Systems in Upset Conditions

Often, surge control systems which are operating satisfactorily under normal conditions are inadequate when fast response time is required. A fast rise in head would be an example. Such a fast rise can occur if one compressor in a series combination shuts down because of malfunction or if the flow is decreased rapidly because of process malfunction. As can be seen from Figure 37, on shut down of one compressor in a series combination, the remaining compressor would experience an increase in Delta P, due to a back pressure effect. Without provisions in the surge control system, it would start opening the surge valve when the operating point approached the surge line. Because of the response time of the system (actuators, valve travel, etc.), the compressor most likely would experience surge. This can be avoided by adding anticipatory action, either by adding one way derivative action to the surge controller on a fast rise in head, or by fast decrease in flow. Surge valve opening would occur immediately after detection of such fast changes, reducing the response time and thus preventing surge.

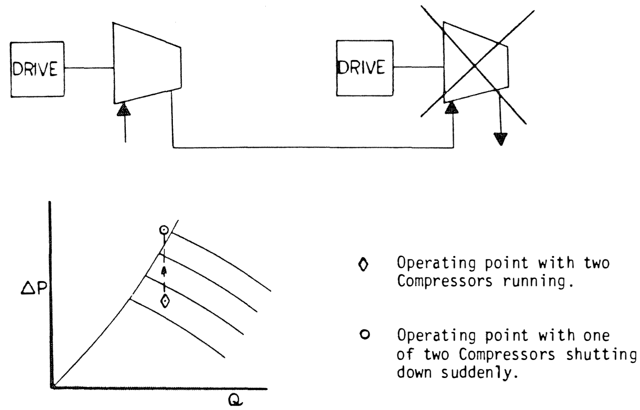


Figure 37. Series Operation of Compressors.

Stability of Surge Control Systems and Parallel Compressors Close to Surge

Parallel operation of compressors close to the surge line can lead to unstable operation of the surge systems if suitable control precautions are not taken. Only in rare cases are the surge lines of parallel compressors the same. Manufacturing tolerances and piping differences result in different surge lines, as shown in Figure 38. Two types of control design are possible. In the first type of design, the speed signal or other control signal is biased so that both operating points have an equal distance to the surge line. In the second design, the speed or control signal to both units is the same, which would result in a different location of the operating point in the compressor envelope. The surge control system is equipped with a feature to detect the distance from the operating point to the surge line. Any speed decrease of the unit closest to surge will be interrupted at a preset distance from the surge line. This preset distance is selected just to the right of the point where surge valve opening would be required. Any additional reduction of throughput will be achieved with the other unit until it also reaches this preset distance. This feature allows maximum operation without opening the recycle valves, which considerably decrease the plant efficiency.

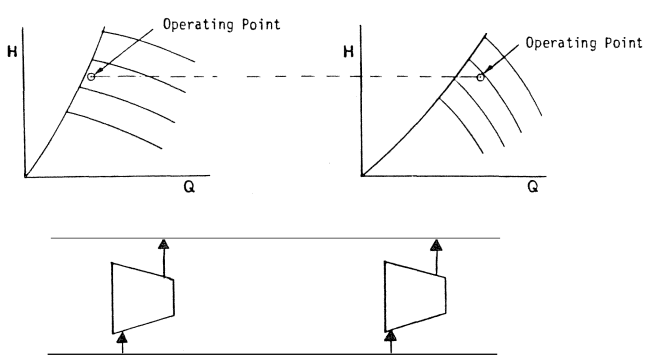


Figure 38. Parallel Operation of Compressors.

The first design is more complex than the second. The preferred design depends on the location of the efficiency lines. A comparison of combined efficiency of the parallel compressors will most likely determine the type of design preferred.

Surge system stability of parallel or series compressors is important, since oscillation of the surge control systems, because of their time constants, can lead to excursions into deep surge which are as damaging as the surge which is to be prevented. To prevent oscillation of compressors, their time constants should be as far apart as possible. One of the simplest ways of achieving this is by affecting the travel time of the recycle (surge) valve. Since, obviously, the opening time of the valve should be as short as possible, we are left with the adjustment of the closing time. This adjustment has proven effective in preventing oscillation in most applications. Due consideration also should be given to integral and proportional action of the surge control systems. However, it is much easier to find a stable condition empirically with valve closing adjustment than to use other methods, which often require dynamic modelling.

C. DYNAMIC SIMULATION OF COMPRESSOR SYSTEMS

Dynamic simulation is an engineering tool to evaluate a process design. It involves the use of a mathematical model describing the behavior of the system in an equation form that is then solved using a computer. The equations consist of algebraic and differential relationships, whose origins come from laws of physics and from empirical data. The computer representation then can be used to predict how the actual system will behave in "what if" situations. The results describe both the steady state operation and the transient behavior during upsets. Because this analysis can be performed during the design phase, the knowledge gained can be used to reduce overall costs of design, construction, startup and operation of the system of interest. The development of a model for a specific application involves many people; e.g., process engineers, design engineers, clients, simulation analysts, and vendors. The exchange of information between these groups also serves a quality assurance function for a project.

The Need For Simulation Studies

With higher and higher trends in the capacity, pressure, speed and horsepower of centrifugal compressors, the design process becomes more involved. The compressor designers and the application engineers, who are responsible for the design of the entire process, are faced with the problem of designing machines which are often larger and more complex than ever before built. They must somehow provide reasonable assurances that the machine will operate as intended, once installed. This process of satisfying one's self (and the plant's owner) that the compressor system design will perform satisfactorily can be very subjective. Traditional methods of analysis cannot provide accurate predictions of how a compressor and a process will interact. Because of this, designs are often based upon the designer's experiences and opinions.

A tool which has been extremely valuable in evaluating the performance of a compressor system prior to installation is dynamic simulation. The simulation studies result in design recommendations concerning the number and location of recycles required, sizing of recycle control valves, and setpoint, gain and reset settings for control system instrumentation. The studies also may lead to a better understanding of how the compressors will interact with processes. Startup procedures are tested and amended as required. Reference 11 discusses the applications of simulation to three compressor designs.

The question of utilizing single vs. multiple recycle loops for multi-casing applications is difficult to answer conclusively based on intuition, as past experience may not be a sufficient guide. The dynamic behavior of such systems is influenced by process parameters, control system responses and the turbine-compressor performance curves. The impracticality of manual calculations and intuitive design approaches creates a need for computer aided design. Reference 12 describes a simulation study of a compression-liquefaction system that revealed the original design for a test facility to be severely undersized in two of the five casings. A retrofit program was implemented prior to operation, thus minimizing schedule delays.

Compressor System Simulation

Figure 39 illustrates the procedural steps of a typical simulation study. The steps can be grouped into five phases. In Phase I, the boundaries of interest are established, operating modes and limits identified, criteria of acceptable performance documented, and the types of upsets of interest identified. The mathematical representation is prepared and assembled, and pertinent physical data are gathered. In Phase 2, the mathematical model is converted into computer language, and the program is checked out. A simulated steady state operating condition is achieved and compared with known/expected operating conditions. In Phase 3, the simulation model is evaluated at transient conditions. The model behavior is analyzed and validated as representing the system performance. This validation involves comparison of past knowledge from similar systems, review of behavior by specialists, comparison with expected behavior of physical laws, and a review by personnel with many years of dynamic simulation experience. In Phase 4, the simulation model is used to analyze "what if" scenarios. A series of test runs is performed to document how the actual system is expected to behave, and the results are analyzed. In Phase 5, conclusions and recommendations regarding system performance are documented and submitted in a final report. The data from the test runs are assembled, edited and presented. The final report also documents important aspects of the simulation activity.

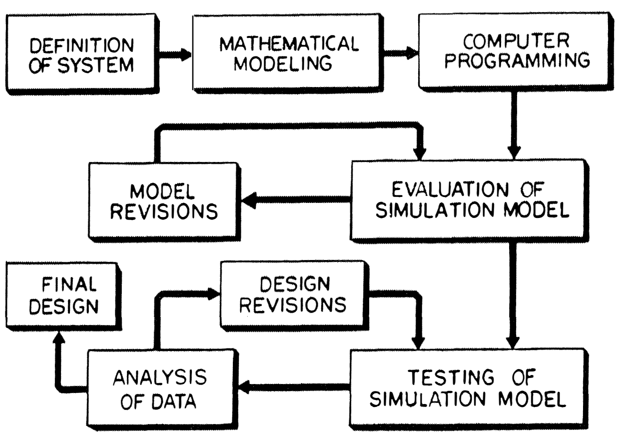


FIGURE 39 - SIMULATION STUDY DIAGRAM

Figure 39. Simulation Study Diagram.

Figure 40 illustrates how the mathematical model for a typical compressor study is organized [11]. The model for a compressor system can be considered to consist of three separate modules, among which information is passed:

1. Process Module—The process module consists of algebraic and differential equations describing the accumulation of mass and energy at various places in the system such as vessels or piping junctions (nodes), the transfer of mass and energy between the nodes, and thermodynamic characteristics such as enthalpies or phase equilibria. The process model can be relatively simple, such as the model for an injection system in which variations in temperature or molecular weight are not significant, and therefore only mass balances need be considered; or may be highly complex, such as refrigeration system models in which heat transfer, phase equilibria and temperature variations have to be accounted for, in addition to mass balances.
2. Compressor Module—The compressor module consists of the empirical head/flow characteristics of each section as provided by the manufacturer, thermodynamic compression equations, and a torque balance between the compressor and its driver. Reference 11 provides a typical compressor model block diagram. The compressor module receives suction and discharge pressure and temperature from the process module. It then calculates compressor flow and discharge temperature as outputs to the process module. In the calculation process, compressor head, speed and volumetric flow are determined. The compressor model may be a simple representation of a constant speed machine, or a very complex one in which variations in speed, efficiency and polytropic coefficients are considered. The degree of sophistication required is a function of both the type of system under study and the objectives of the study. For example, a model to study start-up would have to be more complex than a model used to study constant speed operation.
3. Control System Module—The control system module represents the behavior of the process control instrumentation and final control elements. Controllers, transmitters, computing relays and control valves usually are modeled mathematically. A tie-in to actual control hardware, e.g., distributed control systems, may be required to evaluate the response of software algorithms. Control valve dynamics (essentially the valve stroking period) are modeled, as are significant measurement lags.

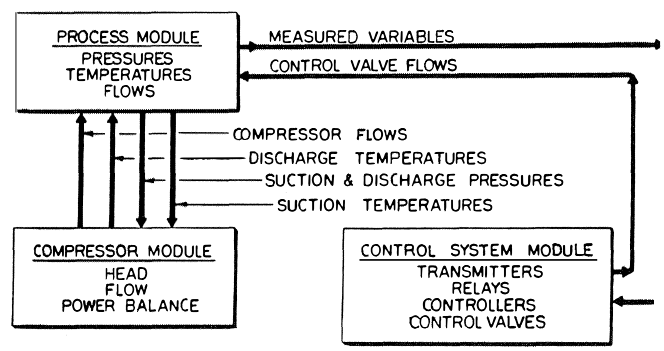


FIGURE 40 - COMPRESSION SYSTEM MODEL.

Figure 40. Compression System Model.

Example of a Simulated Compression System

Figure 41 is a simulation flow diagram for one train of a parallel train configuration that was used to handle gas from an oil field separation process. The incoming gas/water/oil flow from the field at low pressure conditions was separated and the gas routed to the suction header of the compression system. The gas was compressed from 140 psig to 700 psig for distribution to a central compression plant, which further increased the pressure for injection back into the oil reservoir.

The major objectives of the simulation study were to evaluate alternate process configurations during both trip and normal process upsets, to evaluate the effects of valve failures, and to determine the sensitivity of valve parameters to the prevention of surge. The methodology described in the previous section was followed. The model was subjected to a series of test runs. Of particular importance were the "Emergency Trip" and the blocked compressor discharge upset; e.g., closure of Block Valve 5 in Figure 41.

The simulation model results were instrumental in the decision to add a second check valve between the compressor discharge and the discharge cooler, and the addition of a hot bypass loop around the compressor. Also, control system parameters, e.g., valve parameters, controller parameters, etc., needed for the prevention of surge were determined. In a follow-up study, the performance of parallel trains during normal operation, upset conditions, and startup was analyzed.

Figures 42 and 43 illustrate typical results for the revised design under "Emergency Trip" conditions. Figure 42 shows the time history of important compressor variables, while Figure 43 shows that the compressor did not surge at high energy levels. In Figure 43, the variable SURVOL1 reflects the flow (acfm) that would indicate the boundary of surge. The variable ACFM1 reflects the actual flow through the machine. If ACFM1 is greater than SURVOL1, no surging occurs. Without the process changes identified above, this condition could not be met. Once a machine enters surge, the simulation model does not describe the resulting phenomena.

Figures 44, 45 and 46 illustrate results from a blocked discharge upset. Figures 44 and 45 show the variables previously described. As Figures 45 and 46 indicate, the compressor went into the recycle mode and operated on the control line set by the Flow/Delta-P controller.

Control Hardware Evaluation

Once the parameters for the anti-surge recycle loop, e.g., valve capacity, characteristics, response time, and controller algorithm and settings, have been determined, the next question centers around the type of control hardware to be used. A decision has to be made whether the control algorithm be

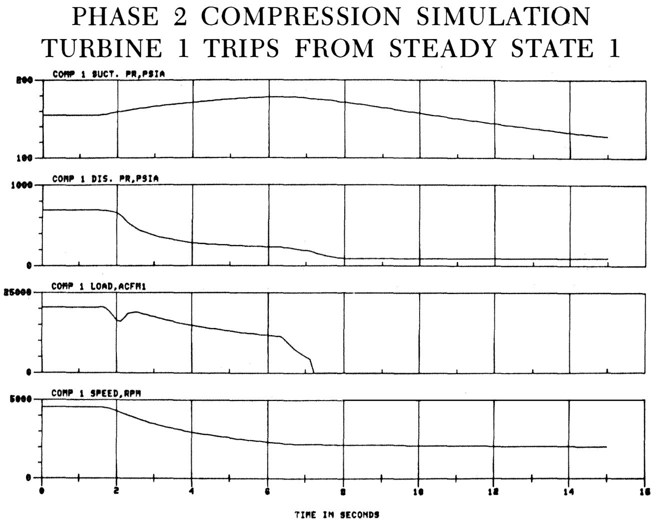


Figure 42. Simulation Results.

implemented with a continuous system, e.g., electronic analog, or with a sampled-data system, e.g., distributed digital control. Chiu and Pobanz [13] discuss the applicability of sampled data systems to fast response loops. The simulation here involved the actual tie-in of hardware for the antisurge controller described in the simulation example above.

The simulation model was run in real time. It provided "field like" transmitter outputs for the flow and delta-pressure measurements and accepted "field like" transducer inputs from the actual control hardware. Figure 47 illustrates the setup. The interconnections between the input/output (I/O) of the computer and the control hardware were the same as field signals (4-20 ma, 1-5 volts). When the train discharge block

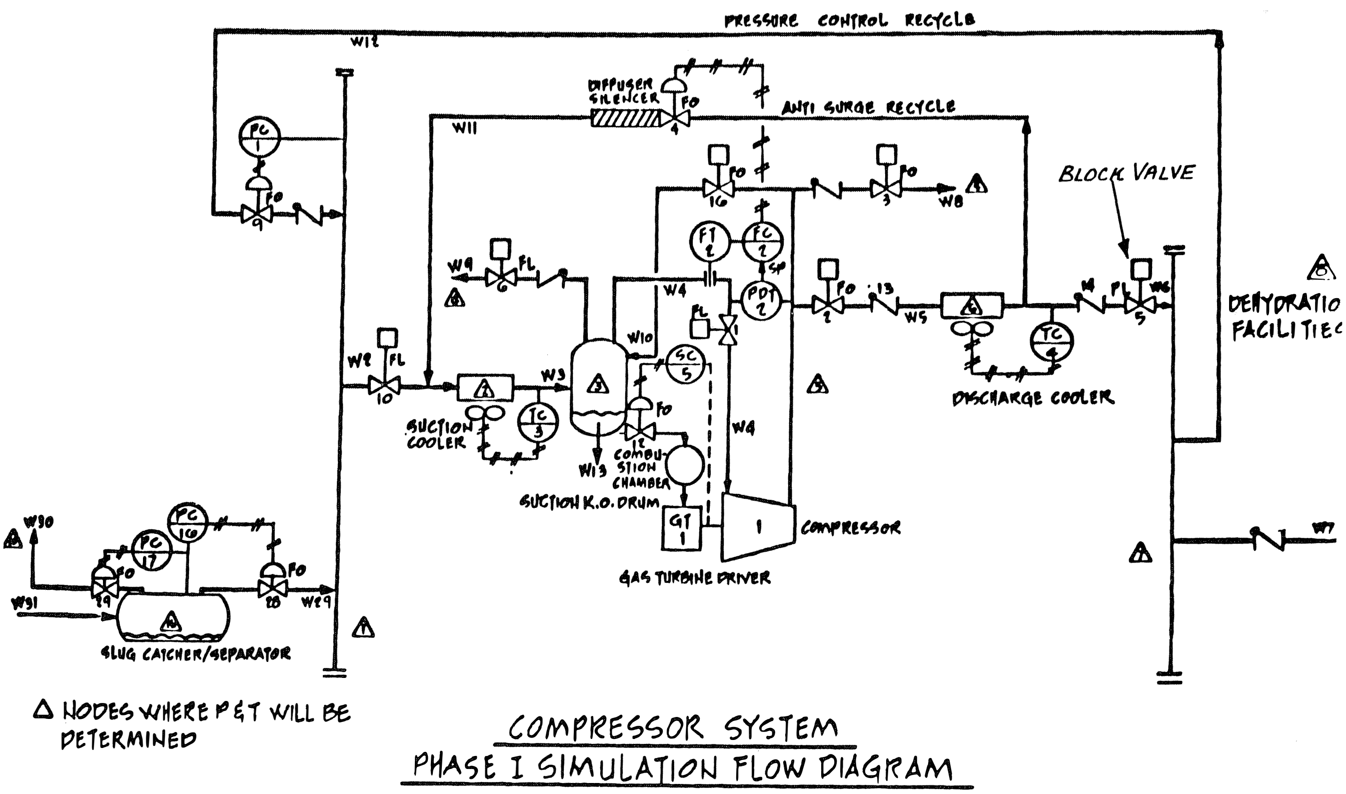


Figure 41. Compressor Flow Diagram.

PHASE 2 COMPRESSION SIMULATION
TURBINE 1 TRIPS FROM STEADY STATE 1

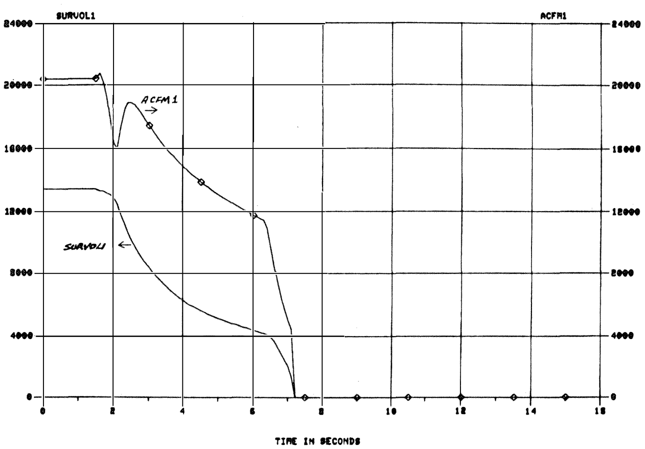


Figure 43. Simulation Results.

valve (Valve 5 in Figure 41) closed, the discharge pressure increased and the compressor moved toward the surge region. The response of the anti-surge recycle loop put the compressor in recycle flow at a flow set by the control line.

PHASE 2 STEADY STATE FULL CAPACITY
BLOCK VALVE 5 FAILURE AT TRAIN 1

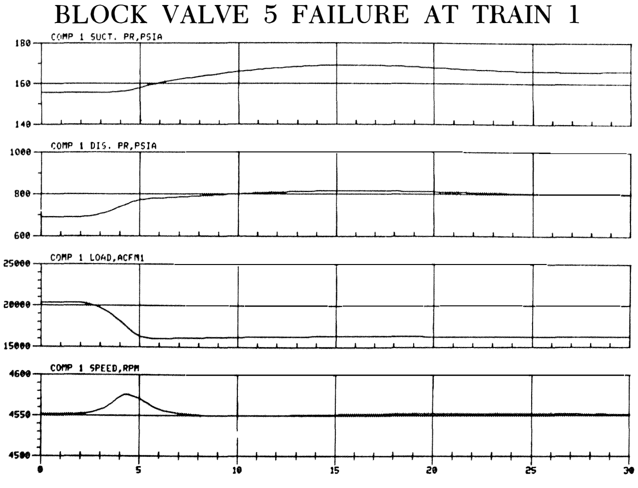


Figure 44. Simulation Results.

PHASE 2 STEADY STATE FULL CAPACITY
BLOCK VALVE 5 FAILURE AT TRAIN 1

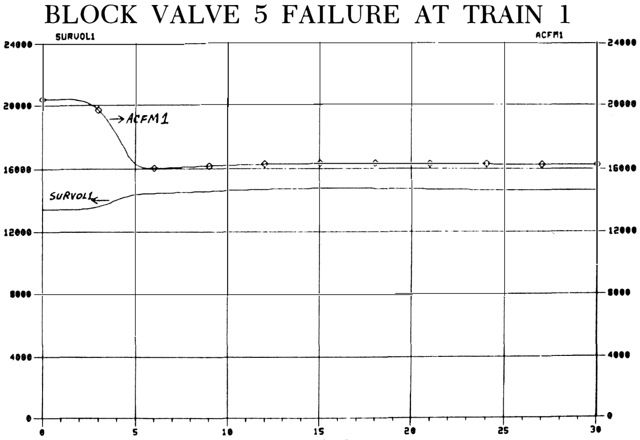


Figure 45. Simulation Results.

PHASE 2 STEADY STATE FULL CAPACITY
BLOCK VALVE 5 FAILURE AT TRAIN 1

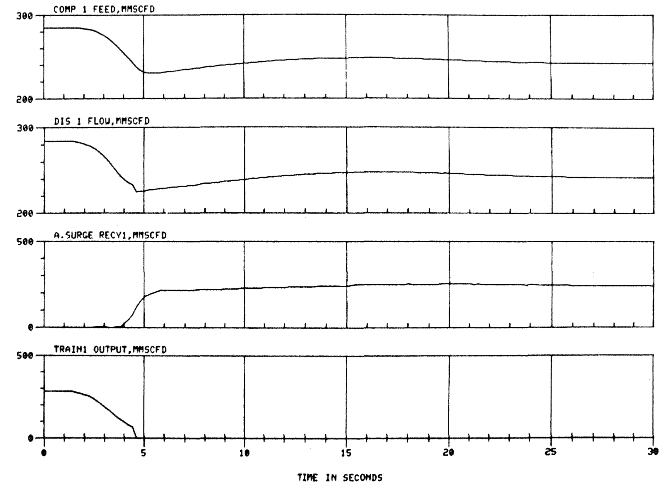


Figure 46. Simulation Results.

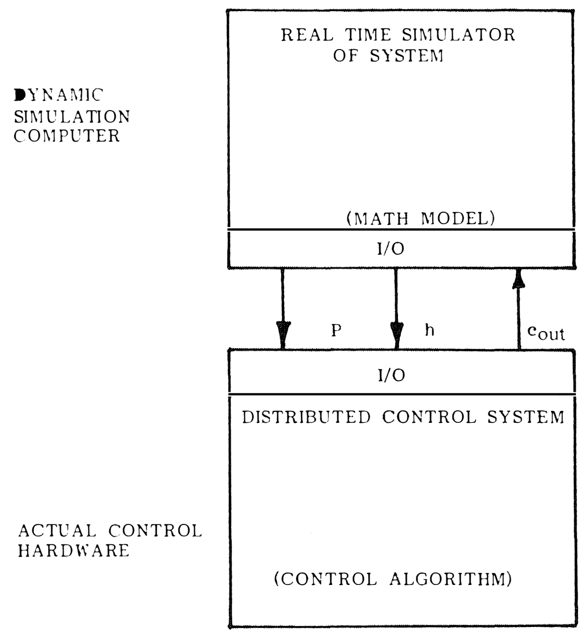


Figure 47. Distributed Control System Interfaced With Real Time Simulator.

The performance of this control loop depended upon the hardware/software implementation of the control algorithm. A comparative analysis for three different distributed control systems (DCS) and an electronic analog controller was made. Figures 48 through 51 summarize the results. Four time history traces are provided in each figure. The first one shows the block valve position. To initiate the upset, it was closed in one second. The second trace shows the compressor surge margin (actual flow less surge flow). When the trace crossed the center line, the compressor entered the surge region. Although some of the traces show the compressor coming out of surge, there is no assurance that the actual compressor would behave in the same manner. As previously stated, the simulation model does not reflect true surge behavior. The last two traces are from the output of the actual control DCS or analog hardware.

In Figure 48, the run was made using an electronic analog controller. As the block valve begins to close (Trace 1), the compressor discharge pressure rises. From the compressor map, this results in a decrease of flow through the compressor and a reduction of the surge margin (Trace 2). This reduction prompts the analog PI controller to act (Trace 4) and brings the compressor's operating point, before it reaches the surge line, back to the control line. The anti-surge controller acts before the operating point reaches the set point (control line), because of its anti-reset windup feature. From Trace 2, it can be seen that the controller provides a comfortable surge margin. This run is the basis for comparison of the remaining runs.

The three DCS systems involved in this comparison had the following sampling frequency:

System A	3 samples/second (fixed)
System B	2 samples/second (fixed)
System C	10, 3 and 1 samples/second and 1 sample/5 seconds (selectable)

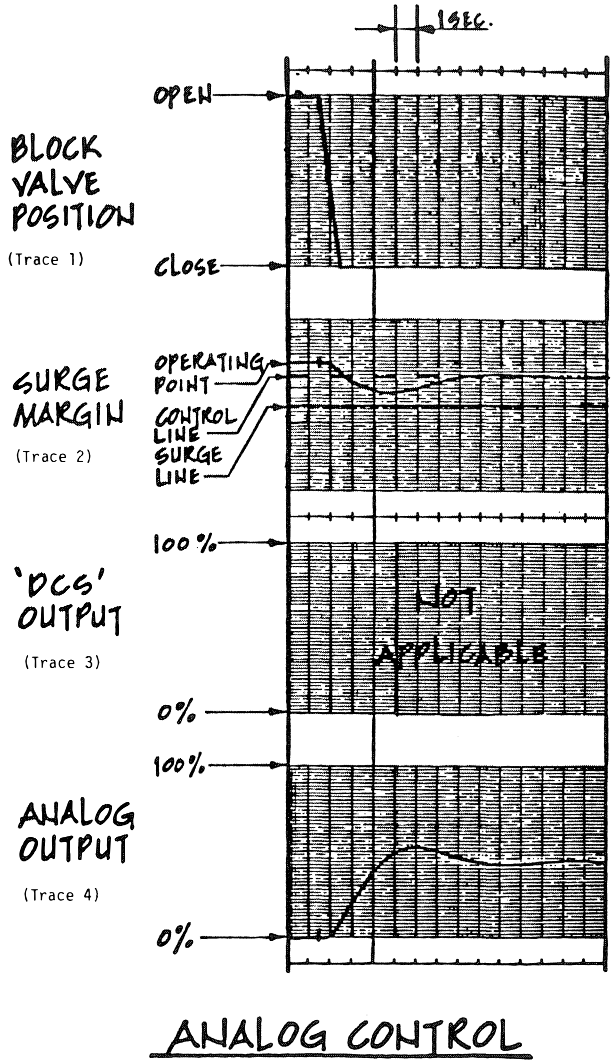


Figure 48. Analog Control.

The results are shown in Figures 49, 50 and 51, respectively.

In Figure 49, the run was made with System A, which had a sampling rate of 3 samples/second. It showed that the compressor will surge for a short time (Trace 2). In actual operation, once the compressor is in surge, the final outcome is unknown. The reason it came out of surge in this study is because of the extrapolation of the compressor map beyond the surge line. Therefore, the results must be used only to determine whether the compressor surged and to determine the margin from the surge line (if it did not surge). In Figure 50, System B, with 2 samples/second, was not fast enough to prevent the compressor from surging. In Figure 51, System C, with 10 samples/second, was selected. Run results showed that it did not surge the compressor; however, the surge margin was small.

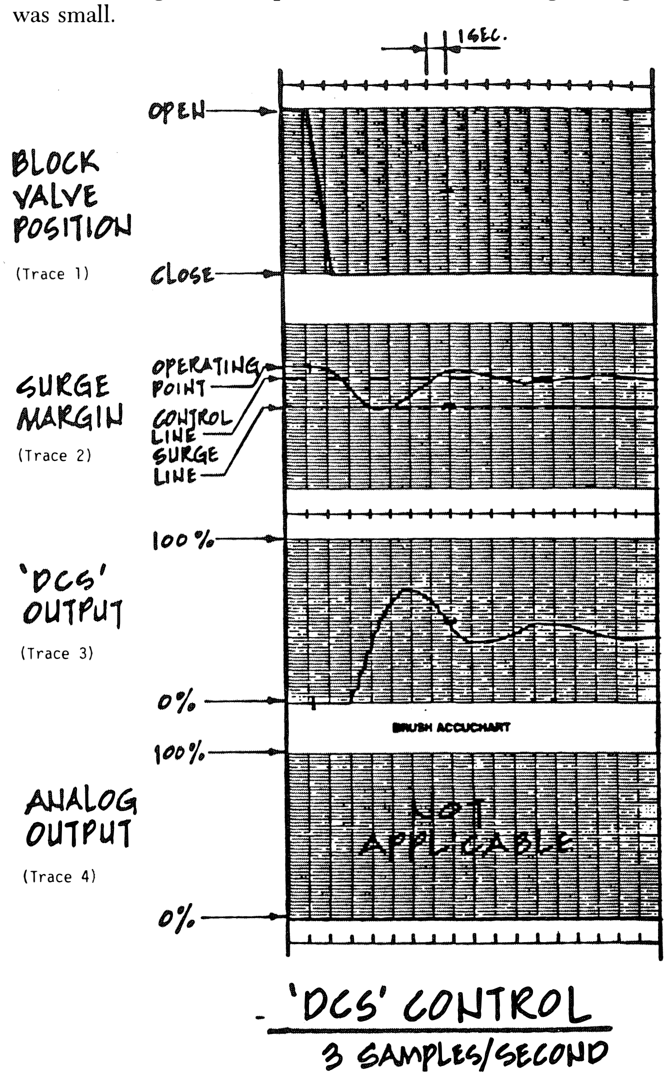


Figure 49. "DCS" Control, 3 Samples/Second.

In this evaluation, a simple proportional plus integral (PI) control algorithm was used. The controller parameters were the same for all four cases. This comparative analysis applies to a specific gas compression system. No attempt should be made to generalize these conclusions. Also, some DCS systems afford great flexibility in the algorithm calculation; this might result in an algorithm that performs better than the one used. Reference 13 also discusses other characteristics of a DCS system which contribute to its performance.

CONCLUSIONS

Compressor surge is a phenomenon of considerable interest; yet it is not fully understood. It is a form of unstable operation and should be avoided. It is a phenomenon that,

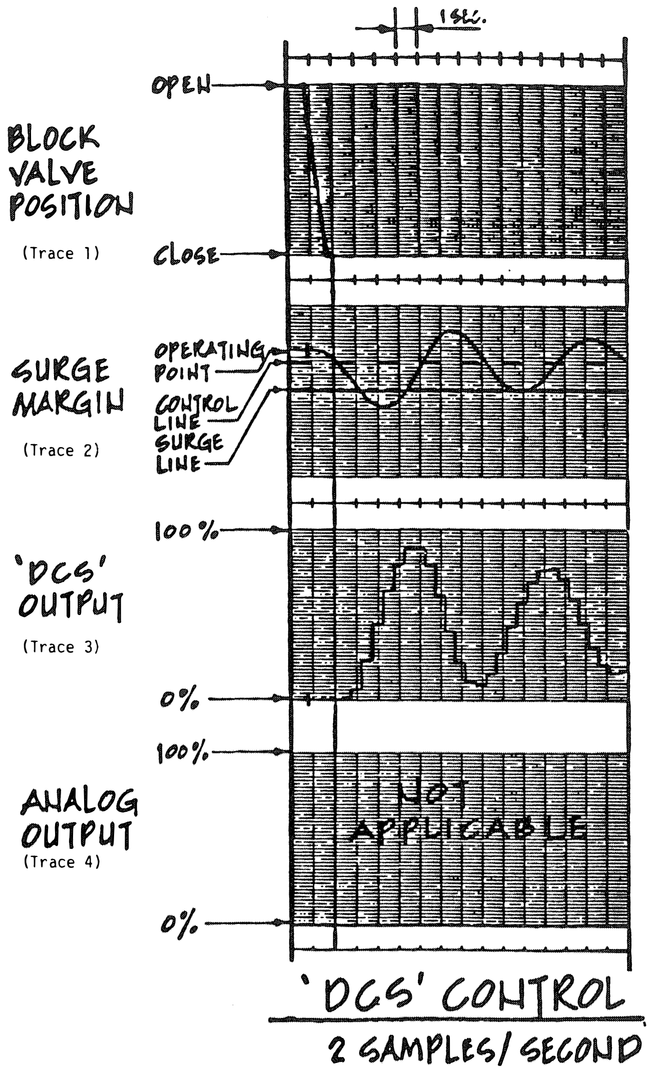


Figure 50. "DCS" Control, 2 Samples/Second.

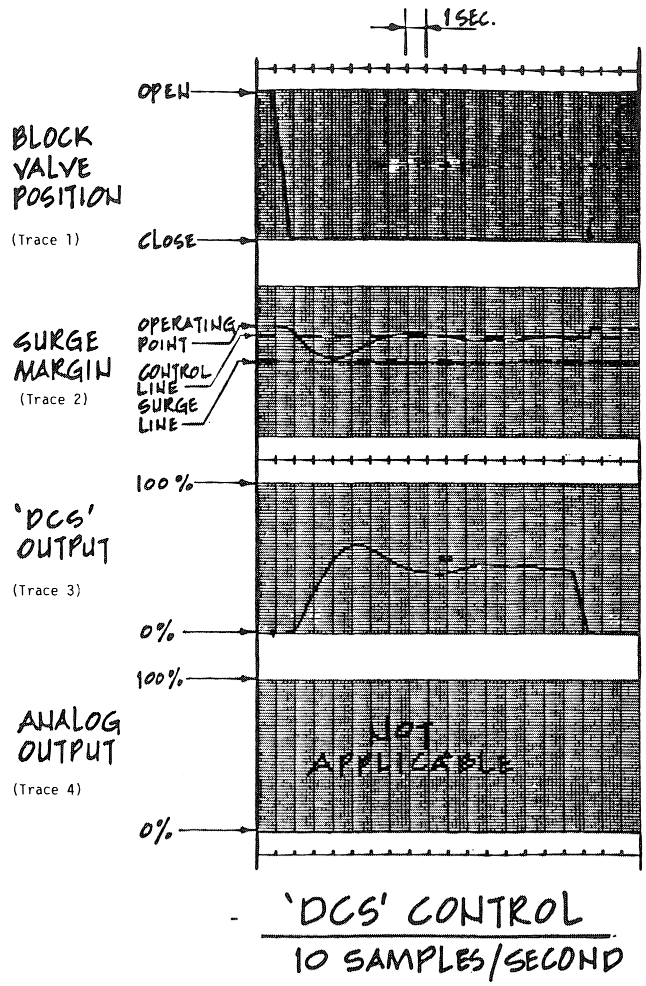


Figure 51. "DCS" Control, 10 Samples/Second.

unfortunately, occurs frequently in the process industry, sometimes with damaging results. Surge has been traditionally defined as the lower limit of stable operation in a compressor, and it involves the reversal of flow. This reversal of flow occurs because of some kind of aerodynamic instability within the system. Usually, a part of the compressor is the cause of the aerodynamic instability, although it is possible for the system arrangement to be capable of augmenting this instability. Compressors are usually operated at a working line, separated by some safety margin from the surge line. Surge is often symptomized by excessive vibration and an audible sound; however, there have been cases in which surge problems that were not audible have caused failures. Extensive investigations have been conducted on surge. Poor quantitative universality of aerodynamic loading capacities of different diffusers and impellers, and an inexact knowledge of boundary-layer behavior make the exact prediction of flow in turbomachines at the design stage difficult. However, it is quite evident that the underlying cause of surge is aerodynamic stall. The stall may occur in either the impeller or the diffuser. When the impeller seems to be the cause of surge, the inducer section is where the flow separation begins. A decrease in the mass flow rate, an increase in the rotational speed of the impeller, or both can cause the compressor to surge. Whether surge is caused by a decrease in flow velocity or an increase in rotational speeds, either the inducer or the diffuser can stall. Which stalls first is difficult to determine, but considerable testing has shown that, for a low-pressure-ratio compressor, the surge usually initiates in the diffuser section. For units with single-stage pressure ratios above 3:1, surge is probably initiated in the inducer.

The antisurge methods discussed, as well as most other proven surge prevention schemes, utilize conventional instrumentation to measure and control certain parameters (e.g. flow, pressure rise, etc.). Compressor and process applications vary so much that it would be difficult, if not impossible, to devise a universal standard antisurge control scheme. Each application must be evaluated in order to determine the required control functions. This requires not only a knowledge of instrumentation, but especially a good understanding of the compressor and load characteristics. The advent of the microprocessor has enabled consolidation of many control functions into one unit, but it has not decreased the amount of basic engineering that must be performed prior to programming. A reliable control which is not influenced by surge point prediction errors and other conditions such as compressor fouling and gas variations (mole weight, temperature, etc.) certainly is desirable, and should reduce the application engineering required for each machine. Efforts to develop such a control package continue, based on the theory that a measurable parameter exists in the region near surge. Additional research is needed to determine if the threat of surge can be detected early enough to facilitate control action in time to actually prevent it. The surge control system engineer has to take into account system valving, response rate, valve selection and location, bypass line sizing, and a host of other factors in order to develop a successful design.

Dynamic simulation of compressor systems is a tool used to evaluate process design and system behavior at the design stage. It involves computer based mathematical modeling. The simulation procedure can provide a better understanding of how the system will behave during process operation, emergency shutdown, and process startups and shutdowns. Dynamic simulation also permits an evaluation of control hardware. It is a design tool that, if properly used, can avoid costly problems with compressor operation.

APPENDIX: EVALUATING THE EFFECTS OF VARIABLE GAS CONDITIONS

The following should be done to evaluate the effects of gas variations:

1. Draw compressor performance curves for normal and alternate operating conditions.
2. Select an appropriate antisurge control scheme, and determine the calibration required for normal operation.
3. Plot control lines on the compressor performance curves for normal and alternate operating conditions.

Performance curves are produced by the compressor manufacturer. References 8 and 9 include information on how calibration constants are determined. The following explains how to calculate the data necessary for plotting the control lines. Performance curves for normal and startup conditions (Figures 31 and 32), and a Flow/Delta-P control scheme (Figure 26) will be used as an example. The following data apply:

1. Conditions at Compressor Inlet

Condition	Normal	Startup
P_i, kg/cm^2	8.19	8.19
T_i, °K	311	311
Z_i	1.006	1.000
$M.W._i$	5.97	12.60

2. Definition of Symbols

P_i — Pressure at compressor inlet, absolute
T_i — Temperature at compressor inlet, absolute
Z — Compressibility factor
M.W. — Molecular Weight
Q — Flow rate, m^3/hr, at inlet conditions
h — Head, inches of water column differential produced by the flow element
S — Span
FT — Flow transmitter
PDT — Pressure differential transmitter
ASC — Antisurge controller
FPY — Summing relay
g — Gain, summing relay input B
K — Bias, summing relay output
C′ — Constant used in flow equation

Subscripts

m — Maximum
sp — Setpoint
pdt — Pressure differential transmitter
i — Inlet
d — Discharge
n — Normal conditions

3. Calibration Values

Q_m — 10,000 m^3/hr
S_{pdt} — 10 kg/cm^2 differential
g — 0.951
K — 0.372

Calculation Procedure

The procedure consists of calculating Q vs P_d data, for plotting control lines, using equation A1.

$$Q = C'\sqrt{\frac{[h/h_m]\,T_i\,Z_i}{P_i \times M.W._i}} \quad \text{(A1)}$$

$$C' = Q_m\sqrt{\frac{P_n \times M.W._n}{T_n\,Z_n}} \quad \text{(A2)}$$

$$C' = 10{,}000\sqrt{\frac{(8.19)\,(5.97)}{(311)\,(1.006)}} = 3{,}953.21$$

Equation A3 is used to calculate "h/h_m" for each discharge pressure point.

$$h/h_m = \left[\left(Q_{sp}/Q_m\right)^2 + g\left(\frac{Pd - Pi}{S_{pdt}}\right) - K\right] \quad \text{(A3)}$$

Equation A3 is derived by rearranging Equation 4. Equation 4 defines the input/output relationship of summing relay FPY. When the antisurge controller (ASC) is operating, the output from FPY is maintained equal to the setpoint. The setpoint, P_d and P_i are known values, so with Equation 4 rearranged to Equation A3, the value of "h/h_m" can be calculated for use in Equation A1. A value of 0 to 1 represents full range of the flow and pressure rise terms in Equation A3. The equation is written in this manner so that the actual value of "h_m" is not needed for the calculations.

$$O = A - gB + K \quad \text{(A4)}$$

where

O = Output
A = Flow signal, i.e. "h/h_m"
B = $P_d - P_i$ signal
g = Gain, control line slope adjustment
K = Bias

With Equations A1 and A3 programmed into a hand held calculator, the data for plotting a control line can be calculated very quickly.

Any gas condition can be readily changed and a control line quickly calculated to evaluate the effect. Likewise, alternate lines for different setpoints can be produced, as in Figures 31 and 32.

Equation A1 is used for all calculations, but Equation A3 varies, depending on the particular control scheme and the values (P_d, R_c, etc.) on the "Y" axis of the compressor performance curve. Since Equation A1 is used for all calculations, it is programmed to perform a control line subroutine labeled "CLSR". Other programs, applicable to the particular instrument scheme, etc., are used to calculate "h/h_m," and the subroutine is used to complete the calculation. Table 2 shows the Hewlett-Packard 41CV program that was used to calculate the control lines in Figures 22 and 23.

REFERENCES

1. Bullock, R. O., Wilcox, W. W., and Moses, J. J., "Experimental and Theoretical Studies of Surging in Continuous Flow Compressors," NACA Report No. 861 (1946).
2. Emmons, H. W., Pearson, C. E., and Grant, H. P., "Compressor Surge and Stall Propagation," Trans. ASME (May 1955).
3. Japikse, D., "Stall, Stage Stall, and Surge," Proceedings of 10th Annual Turbomachinery Symposium, Texas A&M University (December 1981).
4. Dean, R. C. and Young, L. R., "The Time Domain of Centrifugal Compressor and Pump Stability and Surge," 1976 Joint Gas Turbine & Fluids Engineering Divisions Conference, ASME (March 22-25, 1976).
5. Fulleman, J., "Centrifugal Compressors, Chapter 8. In *Advances in Petroleum Chemistry and Refining*, Interscience Publishers (division of John Wiley & Sons), New York-London (1962).
6. Hallock, D. C., "Centrifugal Compressors . . . the Cause of the Curve," Air and Gas Engineering, *1*, 1 (1968).
7. Boyce, M. P., "A Practical Three-Dimensional Flow Visualization Approach to the Complex Flow Characteristics in a Centrifugal Impeller," ASME Paper No. 66-GT-83.
8. White, M. H., "Surge Control for Centrifugal Compressors," Chemical Engineering (December 25, 1972).
9. Gaston, J. R., "Antisurge Control Schemes for Turbocompressors," Chemical Engineering (April 19, 1982).
10. Brown, R. N., "An Experimental Investigation of a Pneumatic Close Loop Anti-surge Control for Centrifugal and Axial Flow Compressors." Masters Thesis—University of Wisconsin, Madison, Wisconsin (1966).
11. Stanley, R. A. and Bohannan, W. R., "Dynamic Simulation of Centrifugal Compressor Systems," Proceedings of the 6th Turbomachinery Symposium, Texas A&M University (1977).
12. Eby, R. S. and Pobanz, N. E., "Dynamic Simulation and Verification of a Compression-Liquefaction System for Material Withdrawn From a Uranium Enrichment Plant," Proceedings for the 1983 Summer Computer Simulation Conference (1983).
13. Chiu, K. C. and Pobanz, N. E., "Applicability of Distributed Control to a Fast Response Loop," Proceedings of the ISA '83 International Conference (1983).

ACKNOWLEDGEMENTS

This paper was intended to be a comprehensive tutorial on surge and surge control, created by incorporating and editing inputs from all contributors into one document. W. R. Bohannan contributed a major portion of Section A, along with contributions from other authors. J. R. Gaston contributed heavily to Section B, once again with significant material from the other authors. Section C was contributed by N. E. Pobanz. The overall paper coordinating and editing were done by M. P. Boyce and C. B. Meher-Homji, who would like to thank all authors for their contribution and time. Thanks are also due to Mrs. Billie Burton, Boyce Engineering International, Inc., for typing the final manuscript.

Table 2. Calculator Program (for HP 41-CV) to Calculate Control Lines.

Storage Registers	Step No.	Program		Subroutine	
00 SP, M^3/HR, 6500 & 7000	01	LBL	CLPd	LBL	CLSR
01 S_{pdt}, Kg/CM^2D = 10	02	RCL	02	RCL	04
02 P_i, Kg/CM^2 ABS = 8.19	03	--		X	
03 Q_m, M^3/HR = 10,000	04	RCL	01	RCL	02
04 T_i, °K = 311	05	/		/	
05 M.W. = 5.97 & 12.60	06	RCL	08	RCL	07
06 C′ = 3,953.21	07	X		X	
07 Z = 1.006 & 1.000	08	RCL	00	RCL	05
08 g = 0.951	09	RCL	03	/	
09 K = 0.372	10	/		SQRT	
	11	X^2		RCL	06
	12	+		X	
	13	RCL	09	END	
	14	--			
	15	GTO	CLSR		
	16	END			

Assign "LBL CLPd" to key "LN." To run program, key in P_d, then press "LN" to calculate Q.