STALL, STAGE STALL, AND SURGE

by

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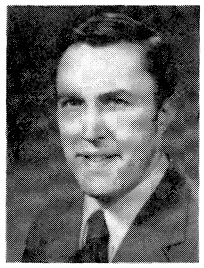

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ABSTRACT

An extensive introduction to stall, stage stall and surge is given with examples of the various stall phenomena including recirculation. Particular components including the inducer, impeller, vaneless diffuser, channel diffuser inlet, channel diffuser and volute are examined for their stall characteristics. Stage stall is discussed and illustrated for various compressor maps. In turn, system surge is examined closely. A phenomenological discussion of surge, the analysis of surge, designing to avoid surge, and surge experience is presented.

INTRODUCTION

The terms stall, stage stall and surge are often bandied about in the compressor world without careful definition. The terms are difficult to define until one gains an appreciation for the phenomena involved. Here we define the terms in limited form. Subsequently, the foundation will be developed so that a deeper appreciation for the basic processes can be established.

Whenever a diffusing flow (decelerating) is found along a physical surface, the possibility exists for the flow to be retarded so severely that it can no longer follow the surface. In this case, the streamlines adjacent to the wall will leave the wall and a reverse flow pocket will develop from that point along the wall surface. In other words, the momentum in the streamlines adjacent to the wall is insufficient to overcome the adverse pressure gradient and the viscous shear stresses along the wall. When the viscous shear effect and adverse pressure gradients overpower the streamlines, forcing them to deviate from the surface, the flow is said to have stalled. Thus, in *stall* the flow direction along the wall is reversed and approaching streamlines are deflected from the surface due to the overpowering effects of viscous shear and the adverse pressure gradient. The flow then becomes reoriented and large viscous shear stresses predominate, at least locally. Noise may be generated.

It is possible for several elements of a compressor stage to stall without the entire stage stalling or similar events occurring. When a stage either has a very strong stall in one of its elements, or a number of elements together collectively stall, so that the overall pressure ratio vs. flow characteristic is no longer stable (negatively sloped), then the stage has entered into a stage stall. In other words, *stage stall* is the condition where the basic flow characteristic of the stage alone is no longer stable. Random fluctuations in flow and pressure may occur.

Whenever a compressor system (including capacitors) is operated at a given speed and the flow is continuously reduced, there is usually a condition where the stage will no longer perform in a stable manner. Large oscillations in the discharge and inlet conditions will result. The stage and system then interact together in a manner which is often violent, giving a *surge*. Hence surge is a *system phenomenon* where a compressor is interacting unstably with an active system to give a strongly coupled, fluctuating, flow network with complete flow reversal throughout the stage on a cyclic basis.

CLASSES OF STEADY STALL

Stall, or flow separation which is a comparable term, can be created either by singular separation or by ordinary separation. It can be static (stationary) or dynamic. The stall conditions presented in the introductory section actually correspond to a singular separation, where the flow is retarded along the wall due to the adverse pressure gradient and viscous shear stresses so that a fluid element immediately adjacent to the wall can no longer proceed in its forward direction. The basic boundary layer equations tell us that the flow condition at this point is a singular point (mathematically speaking) and hence it obtained the name "singular separation." Perhaps the most familiar type of singular separation is the simple stall of an unswept aircraft wing where the flow is no longer able to continuously follow the upper surface of the wing and deviates substantially at the separation point.

A similar but fundamentally different type of separation is found in the ordinary (or three-dimensional) separation process. In this case, a skewing or secondary flow effect is present which redirects the flow so it no longer has a positive through-flow component. This process is a bit more complicated but it is still quite common. It exists in most duct bends including the simple case of a bend in a river. An example is shown in Figure 1. In centrifugal compressors, ordinary separation occurs frequently in vaneless diffusers where the streamlines adjacent to the wall no longer have a positive radial component, but yet have a very strong tangential component of velocity. In this event, the dividing streamlines are redirected back towards the tip of the impeller and a backflow condition results. Examples of this flow condition are given below.

Figure 1. Three Dimensional or Ordinary Separation on a Diffuser Wall Under the Influence of 45° Free Stream Swirling Flow.

The basic objective of a compressor is to increase the pressure of the flow passing through it while convecting the flow through the machine in a stable flow process. Hence, a highly efficient machine is one that can deliver a very high (static) pressure rise with very little kinetic energy at the discharge. Diffusion is essential for an efficient stage. Indeed, one must often force diffusion in several or all parts of the stage to critical limits if a highly efficient stage is to result. However, pushing diffusion to its limits enhances the chance for a stall in any one of many portions of the machine. In the remainder of this section, six examples of component stall are presented for centrifugal compressors in general.

Inducer Stall and Recirculation

Inducer stall is directly equivalent to simple airfoil stall. Figure 2 shows the velocity triangle approaching an inducer. If one reduces the flow rate while operating the machine at constant speed, then the inlet gas angle, β, will increase, thus giving an increase in incidence, i. As the incidence (or for a single airfoil, the angle of attack) continues to increase, the flow is required to accelerate more and more around the leading edge. The flow in turn faces a strong diffusion to bring the limiting (highly accelerated) surface streamline into balance with adjacent streamlines that did not accelerate so greatly. This subsequent strong rediffusion eventually (for sufficient incidence) separates the flow from the surface, thus creating a stall condition at an elevated level of incidence. A throat shock, at sufficient Mach number, can also cause inducer separation.

The existence of inducer stall does not necessarily lead to surge. For high pressure ratio machines, inducer stall may very well create conditions sufficient to permit the stage to enter into a strong surge. But at very low pressure ratios, it is possible to stall the inducer massively while the stage continues to operate stably. For example, the left hand extreme of the speed lines shown in Figure 3 represents a condition with inducer stall. Inducer stall can be detected at the inlet of the compressor by inserting a thermocouple near the shroud or by very carefully watching tip static pressures in that region. The thermocouples will read substantially higher than the inlet temperature due to the existence of backflow which heats up the thermocouple above the approaching air temperature. Likewise, the tip static pressure can show values higher than barometric due to the existence of an increased pressure from backflow, representing fluid that has had some work done on it. Frequently, the existence of the backflow or recirculation

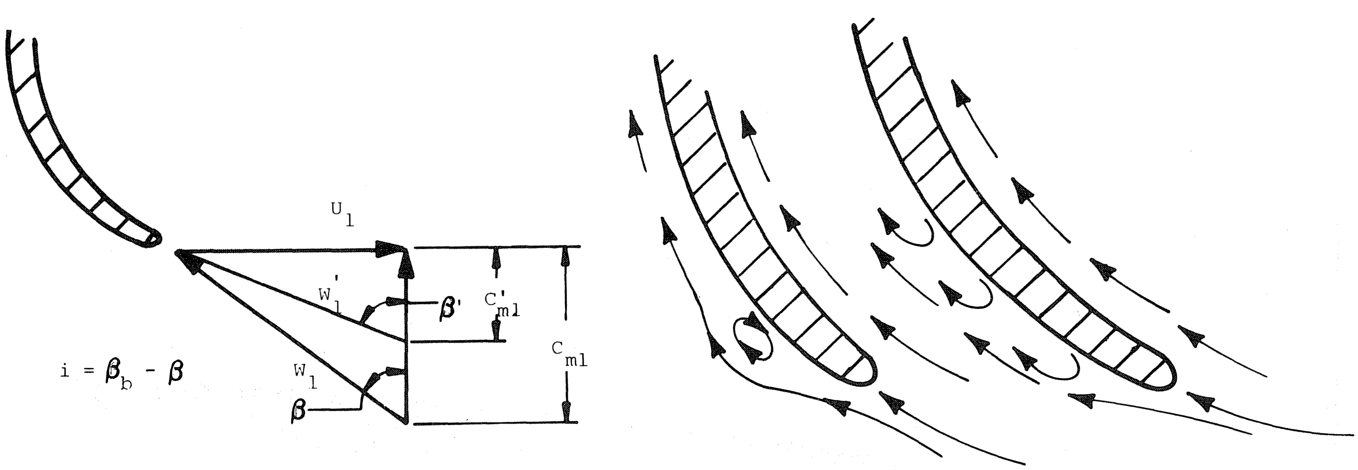


Figure 2. Inducer Velocity Triangle, Incidence and Stall.

TEST POINT	INDUCER TIP INCIDENCE, I_{1t} (deg)	IMPELLER EXIT SWIRL PARAMETER, λ_{2*}
	70 000 RPM	
15	19.3	5.3
16	16.9	4.0
17	12.8	3.0
18	15.9	3.6
19	6.5	2.2
20	9.4	2.5
21	11.4	2.8
22	12.9	3.1
23	9.1	2.6
	50 000 RPM	
26	0.8	1.4
27	6.7	1.9
28	10.8	2.2
29	14.3	2.8
30	17.2	3.8
31	19.7	5.7
32	3.4	1.6
33	8.7	2.2
34	6.7	1.9

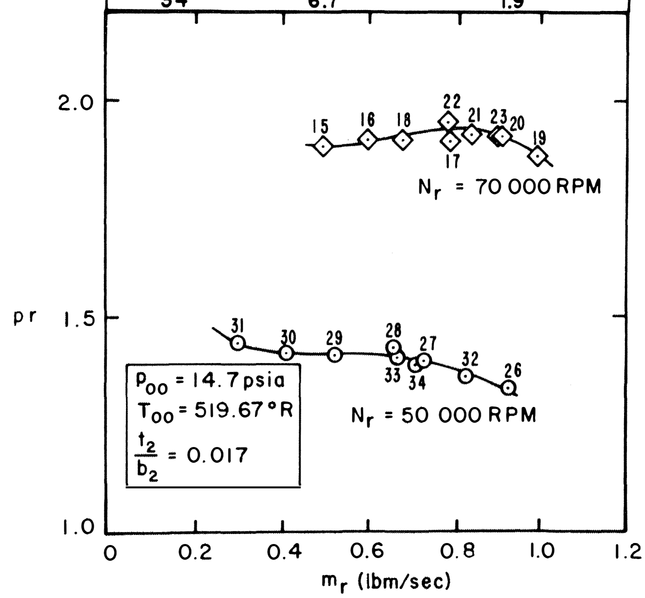


Figure 3. Stage Pressure Ratio versus Reference Mass Flow for a Turbocharger Compressor; Operation with Local Inducer and Diffuser Stall Zones.

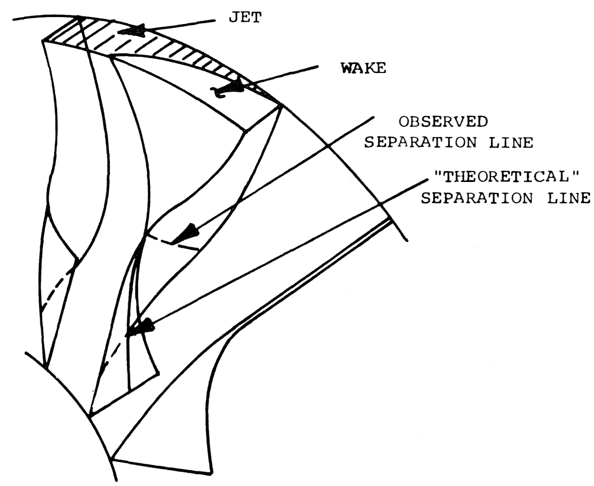


Figure 4. Impeller Jet/Wake Structure Showing Impeller Separation Zone [5].

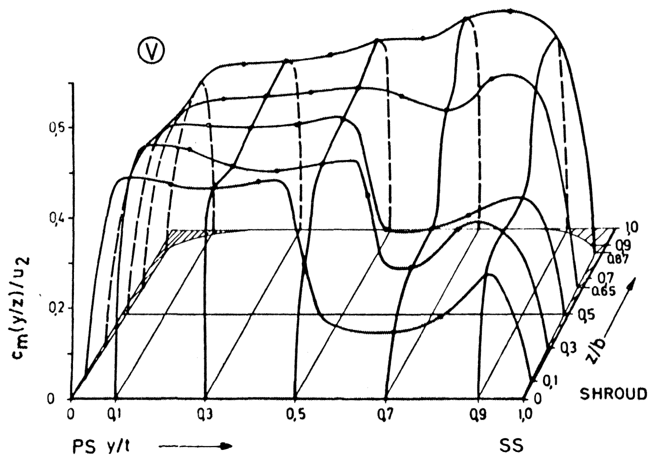


Figure 5. Velocity Distribution c_m/u_2 at Impeller Exit Showing Jet/Wake Structure.

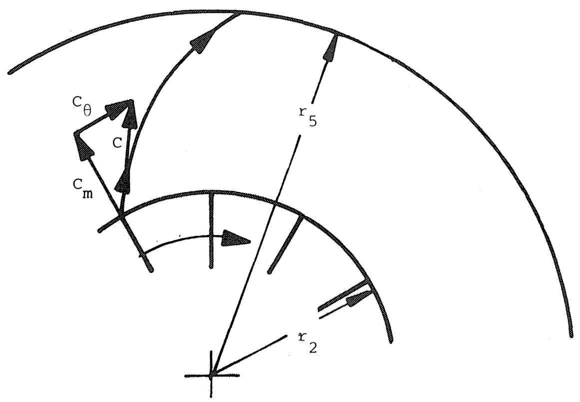
cell can be found a long distance upstream, equivalent to many impeller eye diameters.

Impeller Stall

As the flow proceeds through the impeller, conditions arise where the flow can again separate, but now later on in the impeller passage. Let us assume that either the inducer did not stall, or if it did, then the flow has reattached and has proceeded on in an orderly fashion. As the flow is turned from the axial to the radial direction in the centrifugal compressor, it begins to experience the presence of very strong Coriolis forces. These Coriolis forces provide an "effective cross-channel buoyancy" that operates across the passage and is sensitive to flow velocity rather than merely density as is found with common buoyancy. The Coriolis force, therefore, acts differently on different streamlines, depending on their own velocity. It tends to separate out the high velocity stream tubes from the low velocity stream tubes and isolate them from each other. Thus a wake region is formed in the impeller as shown in Figures 4 and 5. This wake will develop whether singular separation has occurred or not. If singular separation has not occurred, then we simply have strongly skewed boundary layers (possibly ordinary or three-dimensional separation) in the passage which have been swept into a very low momentum, or wake, region. However, skewed boundary layers are highly prone to separation and therefore they may very well separate again in the passage, giving a stalled or wake region. This separation is common in most impellers and represents a very substantial percentage of the flow field in the impeller. The strength of the cross-channel force field can be reduced by using high blade backsweep and therefore it is felt that the secondary flow field may very well be weakened in centrifugal pumps which are constructed with enormous blade backsweep. Pump impeller stall flow patterns, have also received investigation [1].

Vaneless Diffuser Stall and Recirculation

Either singular or ordinary separation can, presumably, occur in a vaneless diffuser as steady stall modes and, in addition, rotating stall can be found (discussed below). Existence of a singular separation simply indicates that the diffuser has been overdesigned and the flow field has been pushed too hard. However, this condition is rarely found. By contrast, ordinary separations are very common in vaneless diffusers. If one examines the velocity triangle in Figure 6, and realizes that this velocity triangle tends to conserve its shape through the vaneless diffuser due to the basic conservation of mass and angular momentum relationships shown in the Figure, then we see that the through-flow wants to follow closely to a log spiral (constant α) in the constant depth vaneless diffuser.



$$rC_\theta = \text{const.}$$
$$\rho C_m 2\pi rb = m$$
$$\tan \alpha = \frac{C_\theta}{C_m} = \frac{\text{const}/r}{m/2\pi rb\rho} = \frac{k\rho b}{m}$$

Figure 6. Elements of Ideal Vaneless Diffuser Flow.

However, streamlines close to the wall have less kinetic energy, and therefore less C_m, while still subject to the same radial pressure gradient. Therefore, they are overturned to the point where they no longer follow the nearly constant flow angle characteristic of the through-flow elements. In this case, the flow is swept back towards the impeller as shown in the flow visualization picture in Figure 7. The possibility for this backflow is strongly influenced by inlet distortion. Early studies of the performance of an incompressible vaneless diffuser flow, for example by Gardow [2] and Jansen [3], conclude that backflow (ordinary separation) might occur when values of the

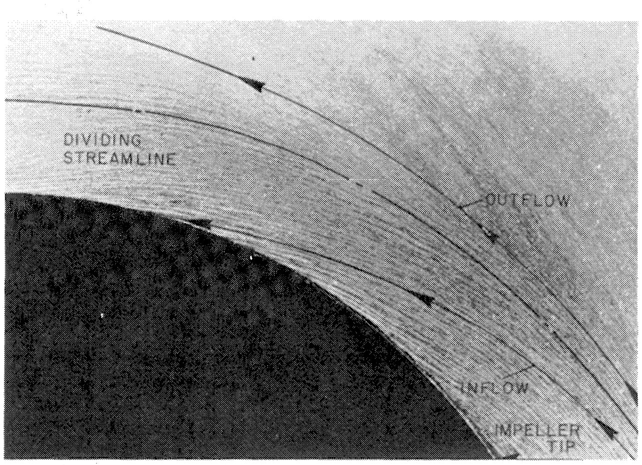


Figure 7. Unpinched Vaneless Diffuser Inlet Flow (Shroud Side) Showing Back Flow Into the Impeller Tip.

inlet flow angle exceed $\tan \alpha_{2m} = 4$. Thus a value of $\tan \alpha_{2m} = 4$ can be considered a necessary, but not a sufficient, condition for ordinary separation in a vaneless diffuser.

Regardless whether a singular separation or an ordinary separation exists in the vaneless diffuser, the breakdown of steady flow in a vaneless diffuser due to any form of stall debilitates the diffuser and enhances the chance for a stage stall. Conversely, existence of a separation in a vaneless diffuser does not necessarily imply that the stage will stall. For example, the flow conditions shown previously in Figure 3 correspond to a vaneless diffuser operating stably with at least three separation bubbles in the vaneless diffuser. Each of these was a form of ordinary separation. These separations are revealed in the flow surveys, Figure 8, as "inward flow" regions.

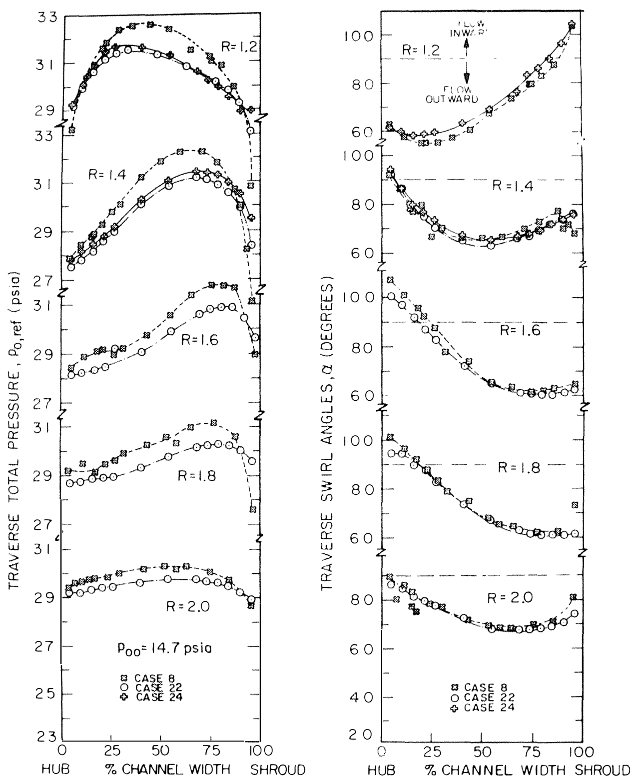


Figure 8. Total Pressure Survey and Flow Angle Survey Comparison for Full-Blade (Case 8) and Partial-Blade (Cases 22 and 24) Cases.

Vaneless and Semi-Vaneless Diffuser Stall For Channel Diffuser Configurations

Large turbochargers and some process machines use a channel or cascade diffuser to obtain increased recovery over a vaneless diffuser. The inlet region to this channel diffuser configuration is a very important component of the stage. It is often referred to as the vaneless and semi-vaneless space of the stage. In this region, increased pressure recovery will be found as the flow rate is reduced at constant speed. Regardless of how the passage is designed for design point operation, the recovery must increase as flow is reduced at constant speed. Extensive studies of stages that approach surge suggest that there may be a critical limit for the value of pressure recovery in this vaneless and semi-vaneless space beyond which a stage cannot stably operate. It has been suggested by Kenny [4] and corroborated by Dean and Young [5] that this value is approximately 0.4. In other words, $C_{p2m-4} = 0.4$ may be a limit for recovery in the vaneless and semi-vaneless space under subsonic operating conditions.

Channel Diffuser Stall

A common mode of separation in a channel diffuser is the singular separation, although some skewing of boundary layers in the corner regions may contribute to the diffuser stall. Nonetheless, at least in the case of simple channel diffusers operating under laboratory test conditions, separation conditions have been comprehensively mapped out for many classes of channel diffusers. For example, Figure 9 shows a pressure recovery map for a channel diffuser. The peak recovery line passes very close to the line of substantial transitory stall (an *unsteady* singular separation process) and lies beyond the line of first stall. Thus it is clear that the designer working with high performance channel diffusers is confronted immediately with the prospect of component stall. A conservative designer may choose to configure the channel diffuser with a high value of L/W in order to avoid regions of possible transitory stall.

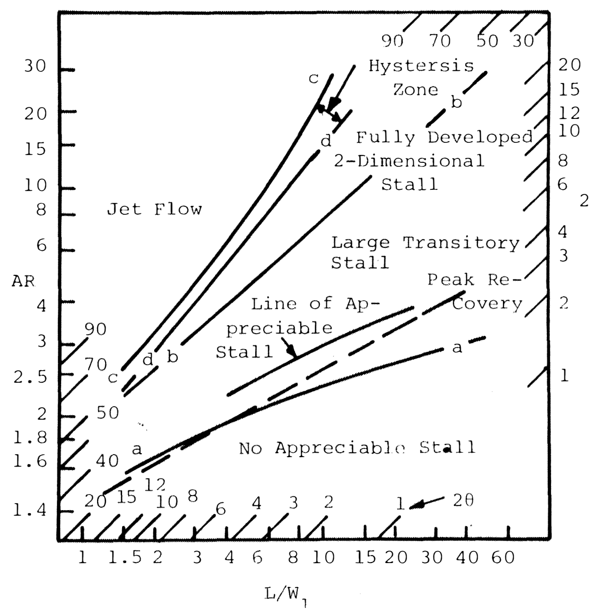


Figure 9. Channel Diffuser Regime Map Showing Various Stall Modes.

Stalling of the channel diffuser can play an important role in the eventual stalling and surging of an entire stage (see discussion later). Consequently, one characteristic that has been used to correlate the stage range is the measure of channel diffuser $\Delta 2\theta$. If we examine Figure 10, we can observe typical channel diffuser pressure recovery maps. Each contour map shown in Figure 10 corresponds to a different blockage level. The important characteristic is the change in the *relative* location of pressure recovery contours as a function of inlet aerodynamic blockage. It will be noted that the contour shifts to the right as one moves to ever increasing values of blockage. In other words, for a compressor operating along a speed line from a high flow to a low flow level, where the throat blockage is known to continuously increase, we move closer and closer to the ridge of peak recovery, which is also the region of

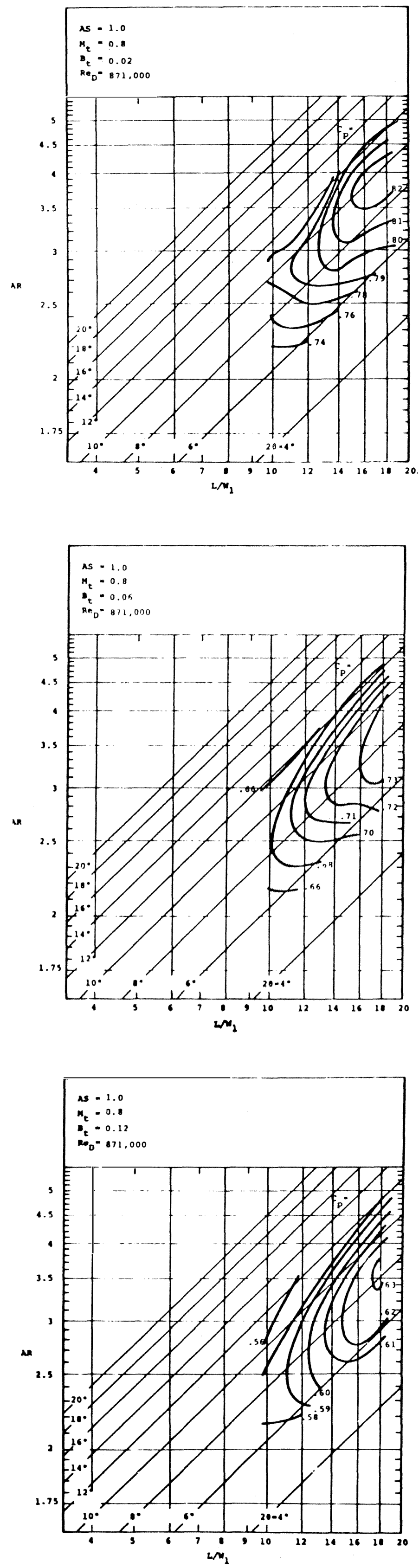


Figure 10. Channel Diffuser Recovery Maps as a Function of Blockage.

channel diffuser transitory stall! Thus the hypothesis has been suggested (put forth by R. C. Dean, Jr. at Creare in 1974 and documented by Japikse in 1980) that a measure of resistance to stage stall is the change in $\Delta 2\theta$ for a point located on the choke line vs. the stage stall line for the respective diffuser maps. One can obtain an estimated value of $\Delta 2\theta$ and plot it relative to the known range characteristic, as is shown in Figure 11. The correlation is observed to be generally good, with but a few exceptions. The stages which form an exception to this correlation were unusual stages whose range was extended substantially by a very specialized diffuser inlet shock structure, with a loss in efficiency. Those stages were analyzed and reported on in detail by Japikse [6].

Volute Stall

Perhaps the most neglected component of a centrifugal compressor from the standpoint of stall is the volute. The basic velocity triangles governing performance in the volute are shown in Figure 12. It is clear that the basic performance of the volute changes along a speed line as mass flow rate is decreased. Whereas the volute is probably sized so that the flow accelerates through it for points on the right hand side of a compressor performance map, the flow state changes so that the flow becomes nonaccelerating and then diffusing as the flow rate is further reduced. Hence the volute itself becomes essentially a wrapped-up conical diffuser which might eventually enter into a stall mode, either singular or ordinary. Stages have been diagnosed where stage stall and surge were found as soon as the volute changed over from an accelerating to diffusing mode [7]. However, the strong swirling flow in the volute should delay stall.

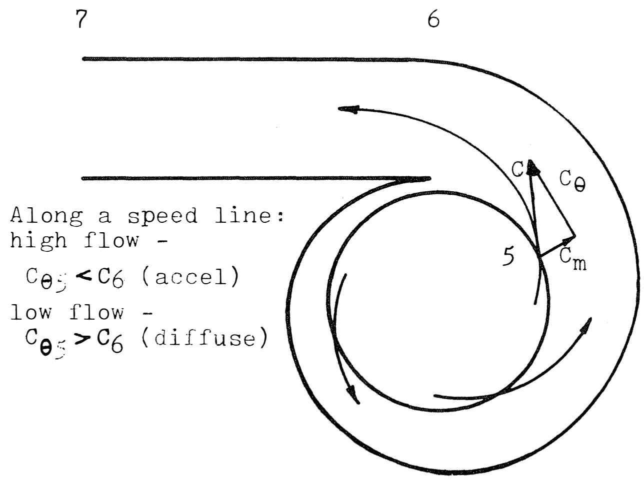


Figure 12. Elements of Volute Flow.

DYNAMIC STALL

The subject of dynamic stall is no less important than the question of static stall, but is far more difficult to report on in detail. The static stall conditions are stationary in nature and therefore can be associated with a fixed location in a machine. Dynamic stall conditions, by contrast, are not fixed and rotate in the machine system at some fraction of the machine rotational speed. Such dynamic conditions can exist in the inducer, in the impeller, or in a diffuser. Many studies have reported the probable existence of dynamic stall conditions. Only a few have studied them in sufficient depth to provide an initial understanding of their basic characteristics. By contrast, far more is known of dynamic stall conditions in axial compressors.

Dynamic instability, or dynamic stall, has been observed in the inducer region of centrifugal compressors by Dussourd and Putman [8] and Amann and Nordenson [9]. The latter distinguished two types of stall conditions. The first was a mild premature condition, frequently called "rough-running" which

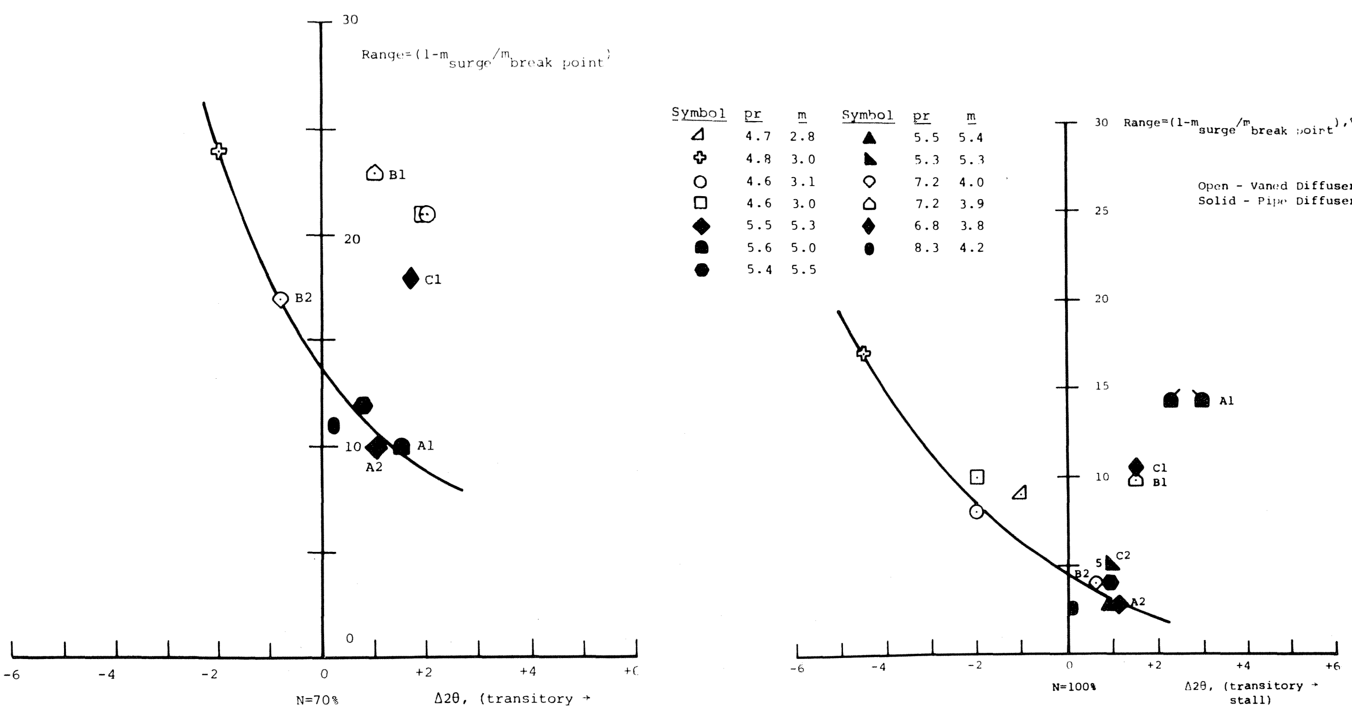


Figure 11. Compressor Range as a Function of Channel Diffuser $\Delta 2\theta$, Part Speed and Full Speed.

occurs at a much higher flow rate than the second stall mode. The second stall is precipitous and leads to a catastrophic breakdown or full surge.

At least one instance of a dynamic instability in the radial portion of a centrifugal compressor impeller has been reported by Abdelhamid and Bertrand [10]. However, this case may be the result of a dynamic instability in a subsequent vaneless diffuser that is, in turn, impressed upon the radial portion of the centrifugal impeller.

By far the most common portion of a centrifugal compressor which is plagued with problems of dynamic instability is the vaneless diffuser. Research work on the dynamic stall of the vaneless diffuser has spanned several decades with a famous initial study by Jansen [3] followed by a number of important subsequent studies. In fact, there are two important avenues of research pursuit that must be addressed to further our understanding of dynamic instability in a vaneless diffuser.

The first avenue began with the work of Jansen who argued, after very careful analysis, that a necessary but not sufficient condition for the onset of dynamic instability is the presence of backflow in the vaneless diffuser. In other words, the static stall condition discussed above must first occur before the rotating stall or dynamic instability can be initiated. Jansen solved the various governing equations to determine limiting conditions for this stall and argued that stall conditions were possible when the inlet flow angle to the diffuser was greater than 75° from the radial or less than 15° from the tangential direction. The actual critical level would depend on various flow conditions. This work was extended significantly by Senoo and Kinoshita [11] and also Senoo, Kinoshita and Ishida [12] who followed the lead of Jansen and critically evaluated conditions for the onset of reverse flow in a vaneless diffuser. They studied the influence of various inlet flow conditions and geometric conditions in the vaneless diffuser to find the critical flow angle for which reverse flow would be initiated. Detailed charts are presented in their work which enable a designer to estimate the influence of these conditions on the flow pattern. In general, they require inlet flow angles in excess of 10° to 30° from the tangential in order to assure stable flow. They avoided any discussion of surge or dynamic stall in their paper — yet Jansen in his discussion of the paper induced them to speak of their conviction in this matter. Here the authors indicated that they felt that reverse flow would quickly precipitate a surge or dynamic instability. Those authors suggest that if the reverse flow region can be constrained to lie within the diffuser and not develop into post diffuser elements (collector or volute or return channel) then the reverse flow zone will probably be a dynamic stall (rotating stall) and the system should not surge.

The second approach taken has evolved on a year-by-year basis with the work of A. Abdelhamid [10, 13, 14]. In their first reference [13], Abdelhamid, et al, have presented some very useful dynamic measurements in a vaneless diffuser. They observed that the dynamic stall condition develops in a gradual manner in the diffuser space as the machine flow rate is decreased. The pattern is sinusoidal in the circumferential direction and equal phase lines are radial at the onset of diffuser instability. The rotational speed of the pressure pattern and the amplitudes both increase slowly as the mass flow rate is decreased beyond the critical stable operating point. They deduced that the phenomena should *not* be attributed to rotating zones of separated boundary layer at the diffuser walls since the pressure fluctuations existed throughout the entire diffuser space (i.e., not just at the wall). However, they felt that it was still probable that a reverse flow zone at the diffuser wall was necessary to initiate diffuser instability. In a subsequent study [10], Abdelhamid became aware of many subtle differences between the dynamic stall characteristics that they were measuring and those presented in earlier studies. In addition, it was determined that the initiation of rotating stall depended very strongly on diffuser diameter and width. The rotational speeds of the oscillation patterns also depended strongly on the diffuser diameter but weakly on the diffuser width. An example of measured results is shown in Figure 13 where the onset of oscillations is clear in the head/flow characteristic, long before any surge condition was observed. Also

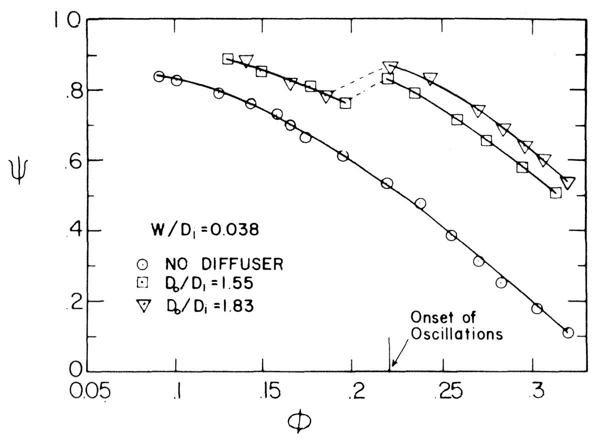


Figure 13. Effects of Diffuser Geometry on System Performance and Limits of Instability. $M_{tip} = 0.18$.

shown is the variation in the critical flow coefficient where the onset of oscillations occurs. In his following paper, Abdelhamid experimentally confirmed the existence of two different rotating unsteady flow patterns for the vaneless diffuser. A low rotational speed pattern was initially observed (as the flow rate is decreased) and then it is replaced by a high rotational speed pattern. The patterns are quite distinct in form. The critical flow angle (measured from the tangential direction) at the initiation of dynamic instability was found to increase as the diffuser radius ratio increased. This effect is shown in Figure 14. On the basis of this work, Abdelhamid has argued that

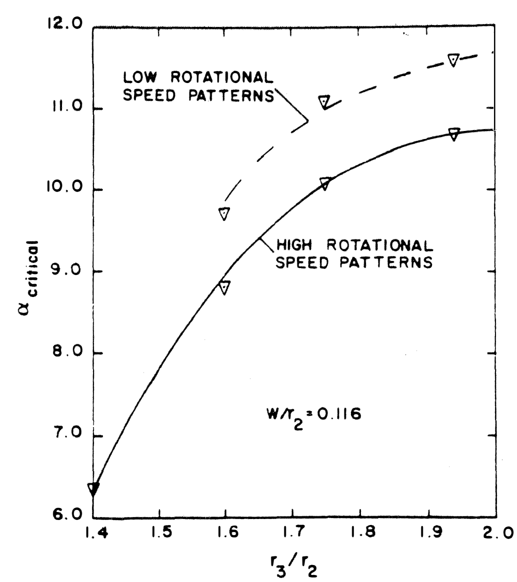


Figure 14. Variation of Critical Flow Angle at Diffuser Inlet with Diffuser Radius Ratio.

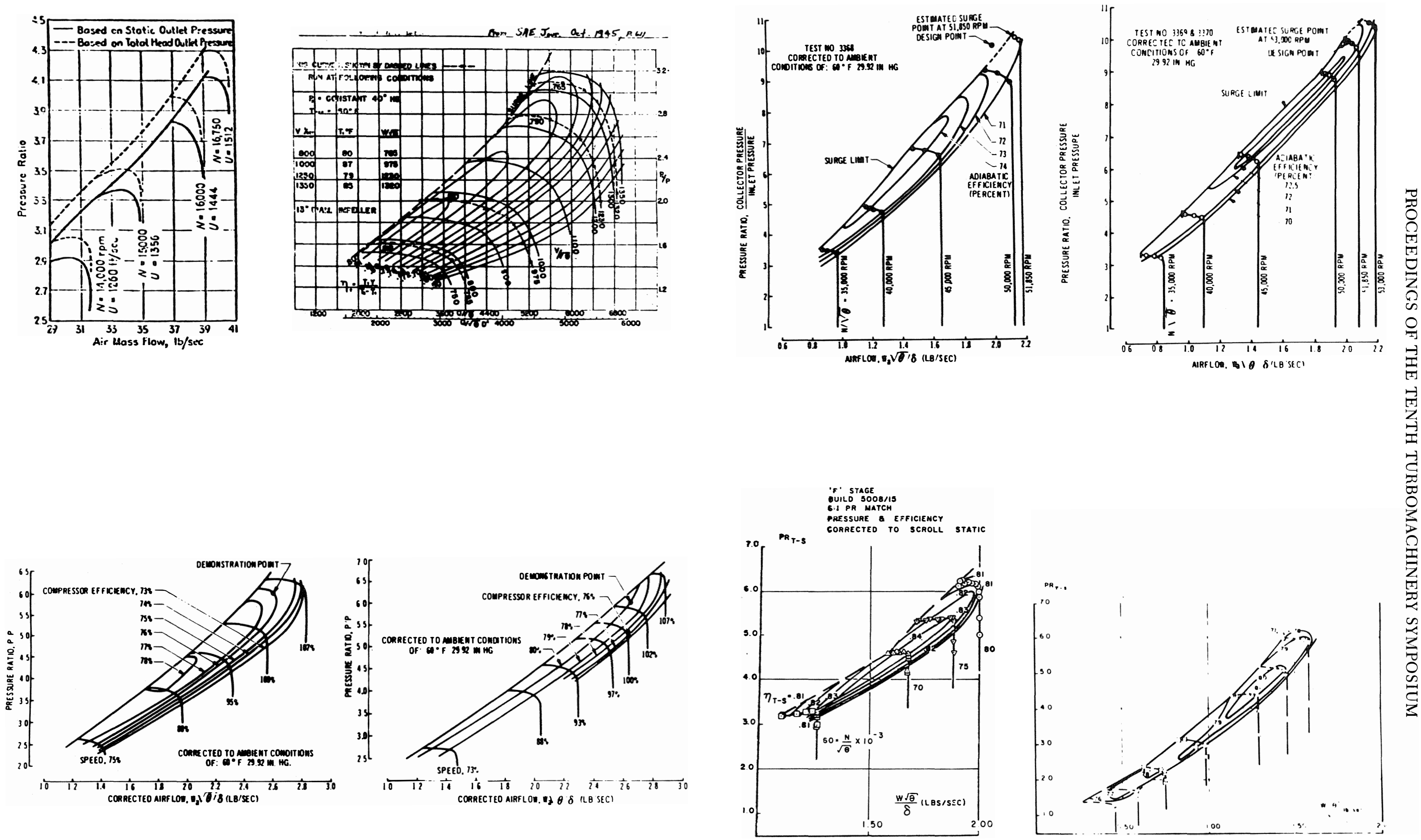


Figure 15. Typical Centrifugal Compressor Maps Showing $\partial pr/\partial m=0$ at the Surge Line [5].

dynamic stall can occur *without* reverse flow zones. He also feels that steady flows can exist with reversed flow zones in a vaneless diffuser. Clearly, the distinction between the two schools of thought and work is noticeable and will require further investigation.

STAGE STALL

In the preceding section, many examples of component stall were presented and comments were made about their presence in typical centrifugal compressors. It was indicated that many modes of stall often exist without the stage stalling or becoming unsteady. Indeed, we have seen cases with three, four, or perhaps even five regions where stall existed while the machine continued to operate stably. However, there comes a point, for any stage when operating at constant speed with ever reduced flow rates, where the stage cannot continue to operate stably and the stage itself can stall.

One possible criterion for stage stall is:

$$\partial pr_{TS}/\partial m = 0 \ .$$

It is interesting to examine a variety of compressor maps as shown in Figure 15 where it is observed that the limit to stable operation does occur approximately where the pressure ratio vs. mass flow characteristic is horizontal. Thus this criterion for stage stall seems relevant. In other words, a criterion for stage stall may be:

$$\partial pr_{TS}/\partial m < 0 \qquad \text{Stable}$$

$$\partial pr_{TS}/\partial m = 0 \qquad \text{Metastable}$$

$$\partial pr_{TS}/\partial m > 0 \qquad \text{Unstable.}$$

But we must recognize that the slope of the speed line is not determined by any one single component in the compressor but actually by all of the components listed above which exist in an actual machine. In order to determine the overall static pressure ratio, pr_{TS}, we must recognize the contribution of each component as shown below:

$$pr_{TS\ \text{stage}} = pr_{TS\ \text{inlet}} \times pr_{SS\ \text{impeller}} \times pr_{SS\ \text{diffuser}}$$

or

$$pr_{0-5} = p_{0-1} \times p_{1-2m} \times pr_{2m-5} \ .$$

We can manipulate this equation by logarithmic differentiation to give:

$$(1/pr_{0-5})\partial pr_{0-5}/\partial m = (1/pr_{0-1})\partial pr_{0-1}/\partial m +$$

$$(1/pr_{1-2m})\partial pr_{1-2m}/\partial m + (1/pr_{2m-5})\partial pr_{2m-5}/\partial m$$

which we can simplify to:

$$SP_{0-5} = SP_{0-1} + SP_{1-2m} + SP_{2m-5} \ .$$

By proper measurements in a centrifugal compressor, we can diagnose the relative importance of the different components which lead eventually to stage stall. A typical example is shown in Figure 16. It is interesting to note some of the important characteristics of such a plot:

a) the impeller itself is frequently close to a neutral stability condition,

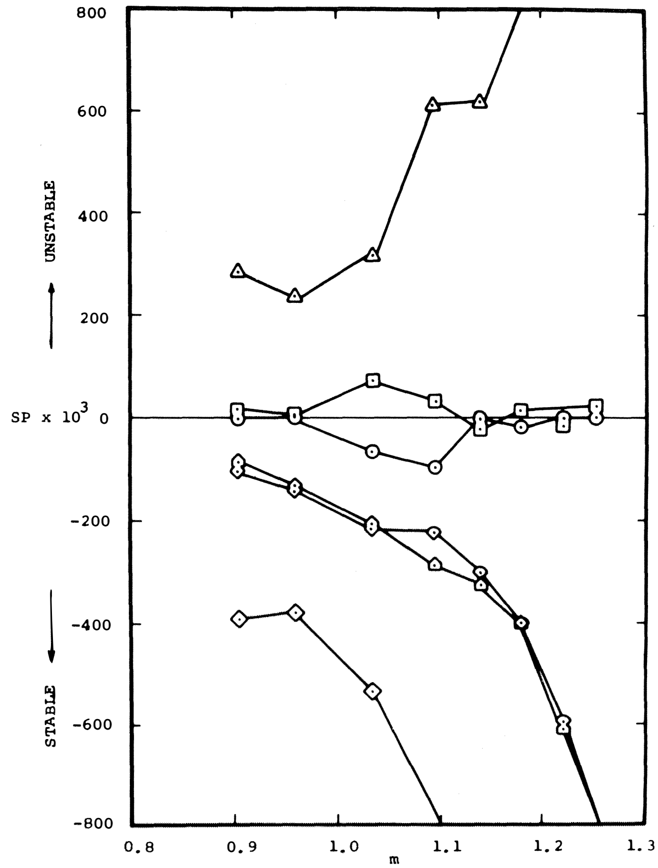


Figure 16. Compressor Stage Stability Plot from Stage Data Illustrating the Influence on Stability of the Stage Elements [5].

b) the channel diffuser is always destabilizing, and

c) the dominating stabilizing influence is the diffuser inlet region.

With this theory now established, we can begin to understand stage stall further. It has been stated previously that a stage can operate in the unstable condition and can do so for extended periods of time without any damage or even without significant noise or rough running conditions (although this frequently is not possible in many installations). However, if a compressor is operating in the unstable mode, its behavior will react as that of any dynamic system under such conditions. For example, if a significant perturbation is given to the flow and pressure of a stage, then it is quite possible for the flow process to break down and backflow to result through the stage. If this is a random process, and the system characteristics are appropriate, then a repetitious process might be avoided. However, if the greater system including the inlet elements and discharge elements (downstream receivers, heat exchangers, etc.) interact with the unstable stage so that the entire system is inherently unstable, then any perturbations will be unstable and a surge condition will result.

SYSTEM SURGE

System surge, or periodic surge, is a characteristic of the entire system where the system is no longer stable and the flow through the compressor breaks down periodically with periods of complete backflow.

When a stage surges, a noisy and often violent flow process results. The compressor is no longer able to meet its full pressure ratio characteristic (on a steady or average basis) and periodically breaks down as one or more elements in the compressor stalls and permits the flow to reverse direction. In some cases, the surge is mild with modest noise levels. In other cases, the surge is so violent that a single surge cycle can destroy a very complex and expensive machine.

Surge is a phenomenon that involves an entire system where components upstream and downstream of the compressor participate in significant flow oscillations. This is a self-excitation of a dynamic system for which stalling of the stage is the driver and the entire network is driven, providing a feedback loop through the piping to which the compressor is installed.

Figure 17 shows a schematic and system characteristics which we will discuss further. Consider for example a compressor operating at point A. If a disturbance occurs in the network such that the flow is suddenly changed to point B, then the situation will arise where the compressor attempts to deliver a higher pressure and a lower flow rate to the downstream elements (collector or other devices). The system can recover from this perturbation since the excursion is very brief and the discharge collector does not respond to the increased pressure but simply accepts the altered mass flow rate. Alternatively, if operation is at point C, and a perturbation occurs such that operation is moved to point D, then an altogether different series of events occurs. In this case, the stage pressure ratio will be significantly less than the collector pressure and the discharge collector will attempt to drive flow back through the compressor stage. Once this process initiates, it will continue until it has blown itself down to a minimum value and then recover along the loop as shown in the schematic. This process then may well repeat itself if the system is permitted to have a mean operating point sufficiently close to the surge line.

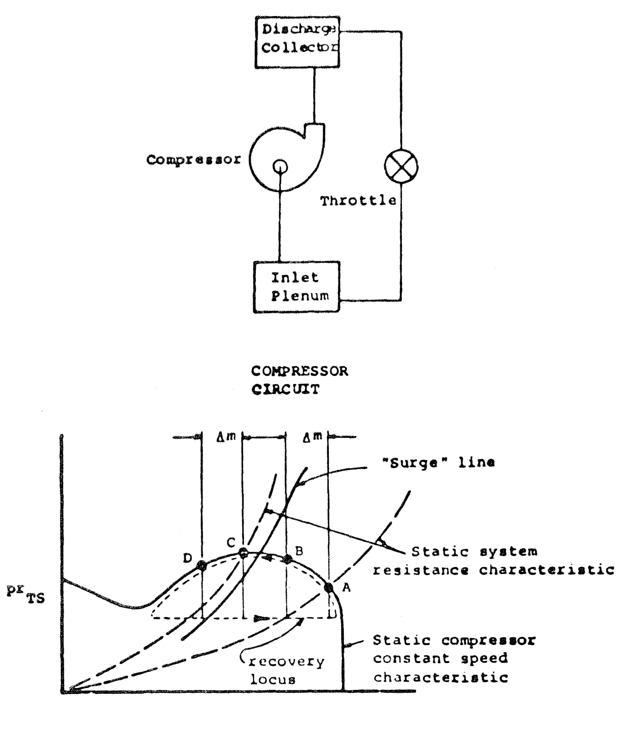


Figure 17. Compressor System Elements and Characteristics.

Detailed analyses of systems have been prepared where the compressor is modeled as an "actuator" and various resistances and capacitances are included in the system. One-dimensional stability analyses have been reported by Bullock, et al [15], Emmons, et al [16], Kazakevitch [17], Emmons [18], Taylor [19], and Greitzer [20, 21]. Such analyses can be helpful in understanding the general characteristics of a compressor system installation.

No theory is available today by which stage stall or system surge can be consistently predicted. However, we can offer the following conservative design rules:

a) avoid inducer stall,

b) prevent any form of steady stall in the vaneless diffuser by controlling the inlet flow angle and the passage contour to eliminate reverse flow on any streamline,

c) if a channel diffuser is employed, then:

   1. keep the pressure recovery in the vaneless and semi-vaneless space below 0.4, and

   2. operate the channel diffuser to the right of the transitory stall line on its recovery map,

d) carefully match a volute (if required) to the stage so as not to destroy the SP value for the stage as a whole,

e) be sure that the composite set of SP parameters yields an overall stage SP that has a negative value.

These rules, when used conscientiously in a design program, have permitted the author and colleagues to avoid surge in most cases.

RANGE EXTENSION

Amann, et al, [22] postulated that pressure field distortions at the tip of the impeller, due to the presence of a vane island diffuser, may very well lead to a flow field distortion that reduces range. Consequently, he constructed a circumferential equalizing chamber, Figure 18, and found improvement in

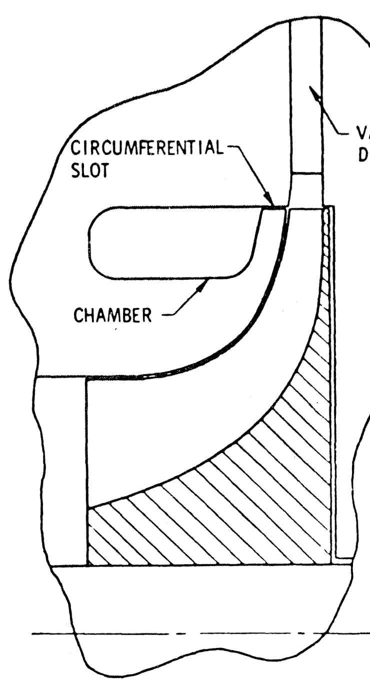


Figure 18. Impeller Exit Tip Treatment to Reduce Distortions and Enhance Range [22].

range. Rodgers and Mnew [23] introduced the rotating vaneless diffuser in order to reduce the shear forces along the walls (the diffuser rotates in the direction of the gas leaving the impeller). They reported that a test condition was established where separation was avoided at a flow rate for which an equivalent stationary diffuser was separated.

Variable geometry has been employed by many investigators to obtain changes in the operating flow range. Variable inlet guide vanes are a popular form of flow control made to work by either of two basic principles. One approach is simply to throttle the inlet substantially (with either a valve or an inefficient guide vane) and relocate the operating point of the compressor on the reference performance map by changes in inlet density. A more efficient approach is to change the inlet swirl, thus unloading the compressor according to the Euler turbomachinery equation. In this approach, substantial levels of swirl are required and only minor effects are obtained up to a preswirl angle of 50°. An early example is shown in Figure 19 from Anderson and Schulman. A subsequent example is shown in Figure 20 from Rodgers [24]. Variable geometry diffusers have been employed so as to rematch the diffuser to the changed flow condition. An example is shown in Figure 21 from Rodgers [25]. Improved flow range can be observed but the penalty leakage around the variable geometry diffuser vanes is also clear.

In addition, improvements in operating range have been reported by Jansen [26] through various modifications in the compressor shroud wall and also in the walls at the inlet to a

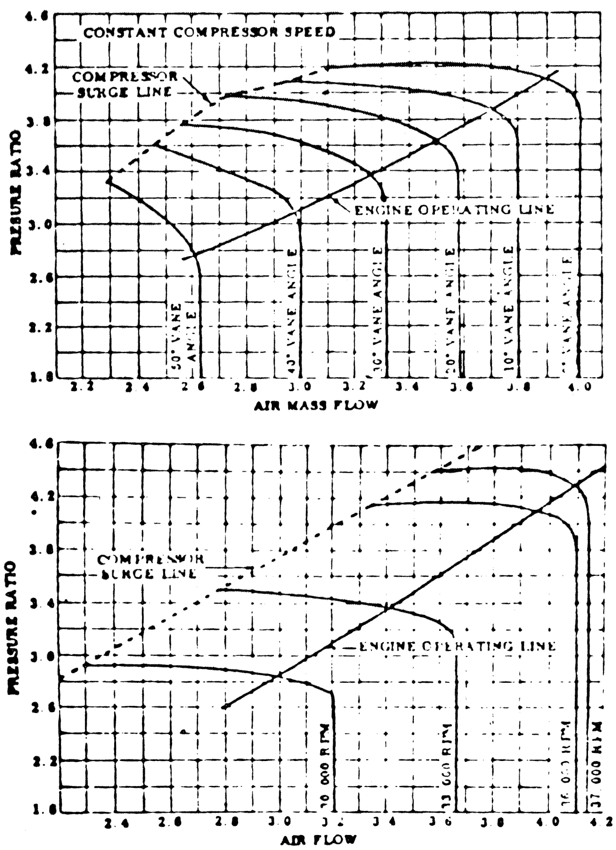


Figure 19. Influence of VIGVs on Compressor Map.
a. Without Preswirl.
b. With Preswirl.

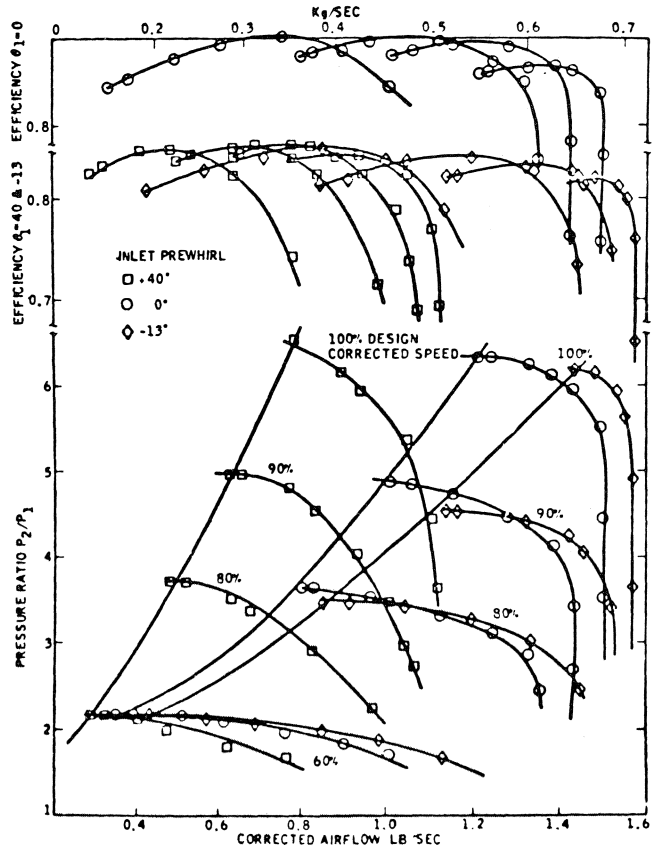


Figure 20. Typical Compressor Performance with Preswirl [25].

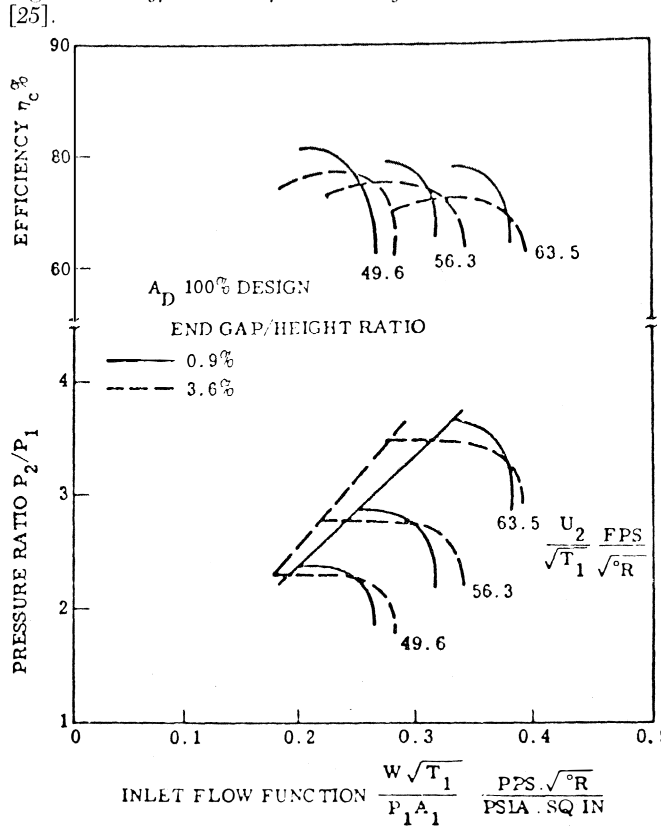


Figure 21. Typical Compressor Performance with Variable Channel Diffusers [24].

channel diffuser. Further work has been done, but not reported extensively, on the use of variable depth vaneless diffusers. Of course, the use of a flow bypass is well known in the field as a method to control stable operating range.

An interesting design example has recently been represented by Flynn and Weber [27]. They postulated that a condition leading to stage stall may be the coupling of an inducer stall with an impeller stall. They presented an interesting design evaluation, coupled with an impeller redesign to test this hypothesis. The authors objectively and correctly pointed out that their evaluation was a design concept evaluation and not basic research. By changing the blade-to-blade passage loadings in the impeller in a drastic manner (Figure 22), they were able to remove the possibility of an inlet separation coupling with an impeller exit separation, thus hopefully improving the stage operating range. The result is shown in the form of an operating map in Figure 23. Indeed, they were highly successful and their premise may be correct. However, there may be other similar ways of accomplishing the same objectives.

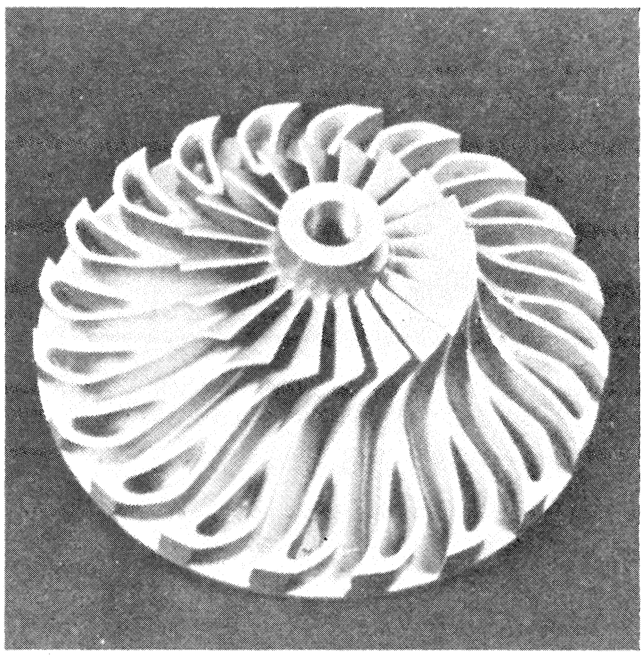


Figure 22. Modified Impeller for Separation Control [27].

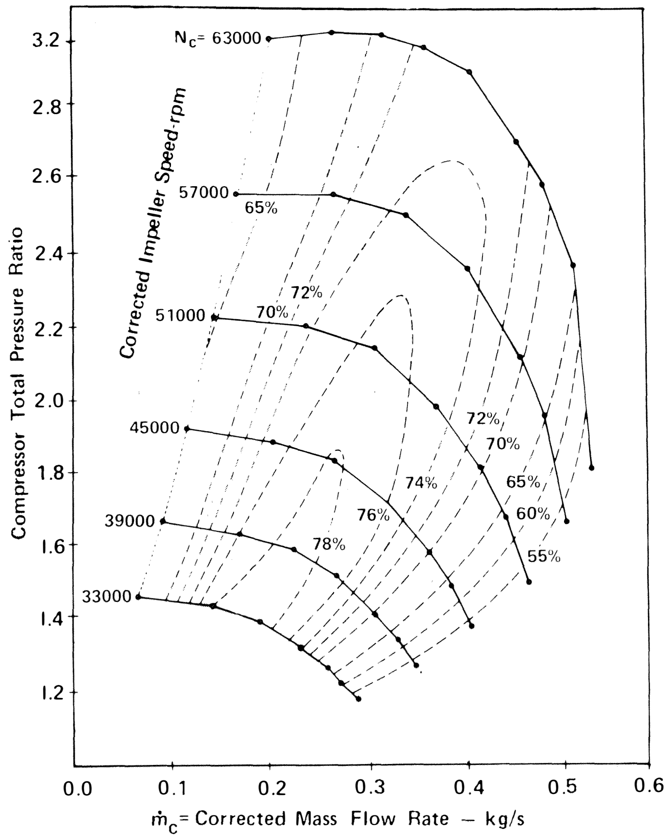


Figure 23. Performance Map for the Modified Impeller (see Figure 22) [27].

CLOSURE

A detailed review of stall, stage stall and surge has been given. Special comment has been made on design rules and range extension. Practicing machinery engineers will want to follow the advanced topics presented herein as current and future research leads to further insight.

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NOMENCLATURE

AR	Diffuser area ratio
AS	Diffuser throat aspect ratio
b	Passage height
B_t	Diffuser throat blockage
C	Absolute velocity
C_m	Meridional component of absolute velocity
C_θ	Tangential component of absolute velocity
C_p	Diffuser static pressure recovery coefficient
D	Diameter
D_i, D_o	Vaneless diffuser inlet and outlet diameters
i	Incidence
L/W	Diffuser length to width ratio
m	Mass flow rate
m_r	Referenced mass flow rate
M_t	Diffuser throat Mach number
N	Rotational speed
N_r	Referenced rotational speed
p	Pressure
pr	Pressure ratio
r	Radius
Re_d	Diffuser throat Reynolds number
SP	Stability parameter, $= (1/pr_{a-b})\partial pr_{a-b}/\partial m$ for process a to b.
T	Temperature
t_2/b_2	Impeller tip clearance to passage height ratio
U	Wheel speed
W	Relative velocity
W/D_i	Vaneless diffuser width to inlet diameter ratio per work of Abdelhamid
α	Absolute flow angle
β	Relative flow angle
ρ	Density
2θ	Total diffuser divergence angle
$\Delta 2\theta$	Relative location of operating point from transitory stall line on diffuser performance map in terms of 2θ separation
λ	Tangent of absolute flow angle
ϕ	Flow coefficient
ψ	Head coefficient

Subscripts

00	Inlet stagnation state
1	Impeller eye station
2	Impeller tip station
3,5	Diffuser exit station
b	Blade
TS	Total-to-static
SS	Static-to-static