REVIEW—Axial Compressor Stall Phenomena

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Stall in compressors can be associated with the initiation of several types of fluid dynamic instabilities. These instabilities and the different phenomena, surge and rotating stall, which result from them, are discussed in this paper. Assessment is made of the various methods of predicting the onset of compressor and/or compression system instability, such as empirical correlations, linearized stability analyses, and numerical unsteady flow calculation procedures. Factors which affect the compressor stall point, in particular inlet flow distortion, are reviewed, and the techniques which are used to predict the loss in stall margin due to these factors are described. The influence of rotor casing treatment (grooves) on increasing compressor flow range is examined. Compressor and compression system behavior subsequent to the onset of stall is surveyed, with particular reference to the problem of engine recovery from a stalled condition. The distinction between surge and rotating stall is emphasized because of the very different consequences on recoverability. The structure of the compressor flow field during rotating stall is examined, and the prediction of compressor performance in rotating stall, including stall/unstall hysteresis, is described.

Introduction

In normal operation of a compressor, the flow is nominally steady and axisymmetric, apart from the blade to blade pressure variations and the small scale unsteadiness associated with the moving pressure and velocity fields of the rotors (or impellers). However, if the performance map of a compressor is plotted in the usual form shown in Fig. 1, as pressure ratio versus mass flow for different rotational speeds,[1] a line can be defined which is commonly referred to as the stall line (or surge line) and which divides the map into two regions. To the left of the line, the flow is no longer steady. Large oscillations of the mass flow rate may occur (called *surge*) or severe self-induced circumferential flow distortions may rotate around the annulus (*rotating stall*), or a combination of both phenomena may appear.

Rotating stall induces large vibratory stresses in the blading of compressors and is therefore often unacceptable for structural reasons. In addition, there can be a large drop in performance associated with this flow regime (efficiencies below twenty per cent can be seen in the literature) [1][2] so that overall gas turbine engine cycles may not be self-sustained. In an engine the greatly decreased mass flow through the system can also cause turbine overtemperatures. Surge can be intolerable from the point of view of system operation and can also lead to high blade and casing stress levels [2]. Thus, no matter which type of instability appears when the stall line is crossed, the stall line generally represents a limit to the useful operation of the machine and is therefore to be avoided.

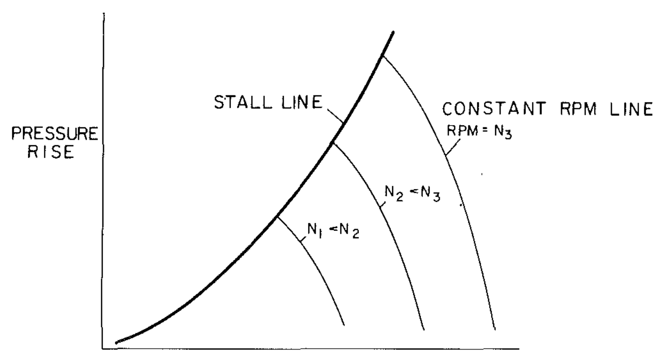

Fig. 1 Compressor performance map

The position of the stall line is a matter of great concern to the designer, and considerable effort is frequently expended to ensure that there is sufficient margin between the stall line and the operating line. Even if this is done, there are still certain situations under which stall will occur, such as rotor speed transients, flow distortions, and nonsteady inlet and exit flow pulsations. Thus, the problem of *recovery from a stall condition* also becomes extremely important.

Because of this the overall topic to be described can be separated, at least conceptually, into two main parts: (1) examination of the basic fluid mechanics associated with the *onset* of the instabilities that lead to rotating stall and/or surge, including such aspects as the impact of inlet distortion, effects of downstream system components on compressor stability, and stability enhancement, and (2) behavior *subsequent* to the onset of this initial instability including large

[1] Actually, corrected flow versus corrected speed.
[2] Numbers in brackets denote references.

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amplitude system transients, limit cycle surge oscillations, structure of the rotating stall flow field, and the effect of stall/unstall hysteresis on system recovery from rotating stall.

The choices of subjects to be covered in a survey of this type will reflect, to some extent, the background of the author. The present author's experience is mainly with aircraft engine turbomachinery. This review is therefore focused on the fluid mechanics of axial compressor stall, although it should be emphasized that many of the ideas (especially those concerning system behavior) can be applied to centrifugal compression systems as well.

Fluid Dynamic Instabilities in Compressors and Compression Systems

We can start this section by describing in a bit more detail the types of phenomena associated with instabilities in compression systems which arise due to the presence of stall. Looked at from the point of view of the individual diffusing passages in the compressor, stall generally implies separation of the flow from one or more of the passage walls. However, compressor blade rows consist of many of these diffusing passages in parallel, so that phenomena can occur which do not happen with a single airfoil or diffusing passage. One of the most striking of these is rotating stall. This is a flow regime in which one or more "stall cells" propagate around the circumference of the compressor with a constant rotational speed, usually between twenty and seventy percent of the rotor speed. In the cells the blades are very severely stalled. Typically there is negligible net through-flow, with areas of local reverse flow, in these regions. The cells can range from covering only part of the span (either at the root or at the tip) and being only a few blades in angular width, to covering the full span and extending over more than 180 degrees of the compressor annulus. It is this latter situation which most commonly occurs in multistage compressors at speeds near design and which is most serious. The part span stall of the front stages of multistage compressors at low speed attracted a large amount of interest in earlier years, but generally has much less severe consequences and is hence not of primary concern.

The basic explanation of the mechanism associated with the onset of stall propagation was first given by Emmons [3] and can be summarized as follows. Consider a row of axial compressor blades operating at a high angle of attack, such as is shown in Fig. 2. Suppose that there is a nonuniformity in the inlet flow such that a locally higher angle of attack is produced on blade B which is enough to stall it. If this happens, the flow can separate from the suction surface of the blade so that a substantial flow blockage occurs in the channel between B and C. This blockage causes a diversion of the inlet flow away from blade B and towards C and A (as shown by the arrows), resulting in an increased angle of attack on C and a reduced angle of attack on A. Since C was on the verge of stall before, it will now tend to stall, whereas the reduced angle of attack on A will inhibit its tendencies to stall. The stall will thus propagate along the blade row in the direction shown, and under suitable conditions it can grow to a fully developed cell covering half the flow annulus or more. In this fully developed regime, the flow at any local position is quite unsteady; however the annulus averaged mass flow is steady with the stall cells serving only to redistribute this flow.

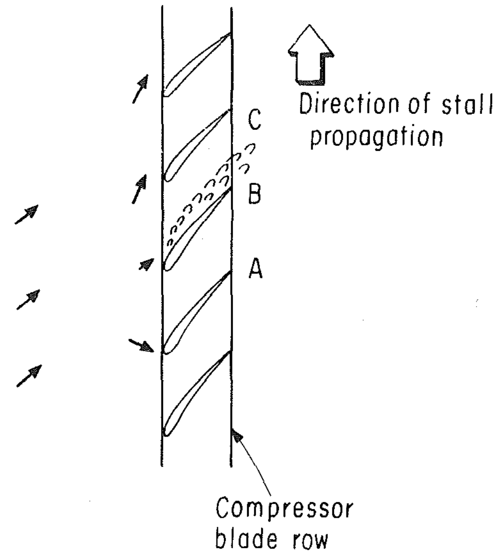

Fig. 2 Physical mechanism for inception of rotating stall

The onset of rotating stall is thus associated with an instability which arises due to the stall of the compressor blade passages.[3] For the overall compression system, this can be regarded as a localized instability. However, a more global *system* instability can also occur, leading to *surge*. In contrast to the behavior during rotating stall, the annulus averaged mass flow and the system pressure rise during surge undergo

[3] The point of stall can, however, be affected by other closely coupled components in the system as described below.

Nomenclature

a = speed of sound
A_c = compressor flow-through area
b = blade chord
B = nondimensional stability parameter;

$$B = \frac{U}{2\omega L_c} = \frac{U}{2a}\sqrt{\frac{V_p}{A_c L_c}}$$

C_x = axial velocity
$\bar{C}_x$ = mean axial velocity
C_x' = nonuniformity in axial velocity
L_c = effective length of compressor
$\dot{m}$ = mass flow
$N/\sqrt{\theta}$ = rotor corrected speed
P = static pressure
P_T = total pressure
ΔP = pressure change across compressor
ΔP_{stage} = pressure rise per stage
δP_T = total pressure distortion
Q = inlet dynamic pressure
R = compressor mean radius
U = rotor speed at midspan
V_p = plenum volume
W = relative velocity
β = relative flow angle
λ = stall cell blockage
ϕ = axial velocity parameter; $\phi = C_x/U$
ϕ^* = design value of axial velocity parameter
$\bar{\phi}_{\text{cessation}}$ = value of annulus averaged axial velocity parameter at full-span stall cessation
ϕ_u = (local) value of ϕ in unstalled region of compressor
θ = circumferential coordinate
ρ = density
ψ_{TS} = pressure rise parameter;

$$\psi_{TS} = \frac{P_{\text{exit}} - P_{\text{inlet}}}{\rho U^2}$$

ω = Helmholtz resonator frequency;

$$\omega = a\sqrt{\frac{A_c}{V_p L_c}}$$

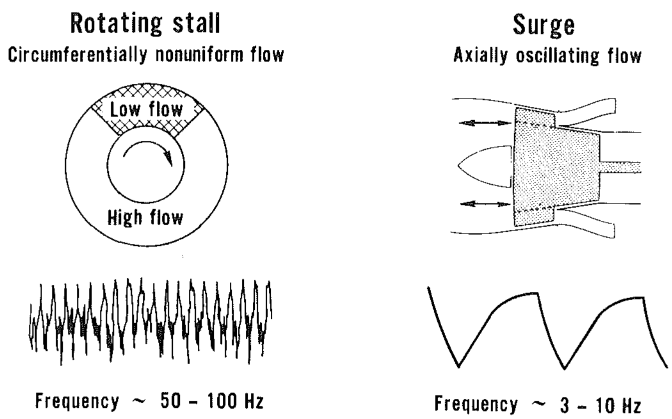

Fig. 3 Modes of system response resulting from stall

large amplitude oscillations. The frequencies of these oscillations are generally at least an order of magnitude below those associated with passage of a rotating stall cell and depend on the parameters of the entire system. In addition, during the surge cycles the instantaneous mass flow through the compressor changes from values at which (in steady state operation) the compressor would be free from stall, to values at which one would find rotating stall or totally reversed flow. Because of the low frequency of the oscillations, the compressor can pass in and out of these flow regimes in an approximately quasi-steady manner.[4]

The two types of instability are indicated schematically in Fig. 3. Sketches of the transient signatures that would be given by typical high response instrumentation, such as a hot wire at the compressor inlet for the rotating stall situation and a pressure probe in the combustor (or other volume downstream of the compressor) for the surge cycles, are also indicated in the figure. Rough magnitudes of the time scales associated with the different phenomena (in an aircraft gas turbine context) are shown as well. It is to be emphasized that although the illustration indicates the phenomena only in this particular context, these instabilities are inherent in pumping systems involving all types of turbomachines.

The two phenomena (surge and rotating stall) are seen to be quite distinct. However, they are not unrelated, since often the occurrence of the local instability (associated with the onset of rotating stall) can trigger the more global type of system instability (leading to surge). Thus one has to consider the possibilities of either type of instability and develop methods for their prediction. We will examine the techniques which have been applied to prediction of rotating stall first.

Prediction of the Onset of Rotating Stall

Correlations for Stall Inception. As noted, a key problem for the designer is the prediction of the point at which stall occurs. This problem has been attacked by many investigators at several quite different levels of approach. The most empirical are the correlations that have been developed for stall onset. The basic concept is to find a parameter (or parameters) which correlates the onset of stall[5] for a number of different blade geometries, compressor designs, etc. In a design procedure for a low hub-tip ratio fan, for example, the parameter could be applied at different span locations along the blading, using the local flow conditions generated by use of one of the many axisymmetric compressor flow field calculations, to see whether any section would be operating under too adverse a condition, while for a multistage compressor the parameter might be applied only on a meanline or averaged basis.

Several of the early correlations of this sort are described by Horlock [5], but one of the well-known examples of this type of approach, which is still much in use, is the work of Leiblein [6]. He developed a parameter which he called the diffusion factor (or D-factor). This was related to the adverse pressure gradient to which the boundary layer on the suction surface of the airfoil was subjected. The D-factor was defined by Leiblein as

$$D = 1 - \frac{W_2}{W_1} + \frac{\Delta W_\theta}{2\sigma W_1}$$

where W_1 is the inlet relative velocity, W_2 is the exit relative velocity, ΔW_θ is the change in circumferential velocity component and σ is the solidity.

It was found that the total pressure loss correlates quite well with D, and, based on Leiblein's cascade results, one can see a rather sharp rise in loss occur as D is increased past a value of roughly 0.6. This can therefore be taken as a *very approximate* criterion for the onset of stall in a cascade (see also [7] for further work using this approach). Although much of the work done by Leiblein was based on two-dimensional cascades, the use of the D-factor has been carried over to axial as well as to centrifugal compressors [8]. Features such as the differences between the flows in a cascade and the flows at the tips of axial compressor rotors, for example, are "recognized" by noting that different limiting values of the D-factor are used for the rotor tips than for other sections.

The correlations based on D-factor, as well as other improved correlations for the stall point, have been intensively investigated by the aircraft engine companies. Virtually all of this information, however, is held as proprietary and there is very little in the open literature, particularly as regards multistage compressors and transonic fans. Several features that are significant, however, are that the "limiting" D-factor, or other loading parameter, (i.e., the value at onset of stall) tends to increase as the aspect ratio and/or the non-dimensional tip clearance[6] decreases. An example of the first of these trends, as presented by Smith [9], is given in Fig. 4 which shows nondimensional stage pressure rise versus flow coefficient from two compressors with aspect ratios of 1.96 and 5. The two compressors each have four stages and the same non-dimensional tip clearance[6]. The lower aspect ratio compressor has a higher pressure rise per stage at the stall point, and stalls at a lower flow rate. Similar results have been found by Fligg [10]. Effects of tip clearance are also illustrated in the above mentioned paper by Smith [9] as well as in [11] and [12].

There are several other factors such as Reynolds number, tolerances and deterioration etc., that can also have an effect on the stall point. These are also accounted for in practice by using correlations, and they will not be discussed here save for the remarks that there can be a substantial loss in stall margin as the blade Reynolds number drops below roughly 100,000.

Although the use of correlations such as D-factor may appear to be an overly simple approach, it is one that is, at present, in common usage for multistage axial compressors. A recent example of such an approach can be found in [13]. More theoretical approaches have not yet led to the definition of stall point with additional precision, but this is not to say that the correlative procedures are all that is needed to put this

[4] In view of this behavior E. S. Taylor [4] has paraphrased P. T. Barnum to describe the distinction between surge and rotating stall, respectively: "You can operate a compressor to stall all of the blades some of the time, or to stall some of the blades all of the time."

[5] Defined here as the condition at which the steady axisymmetric flow becomes unstable.

[6] Tip clearance/staggered spacing.

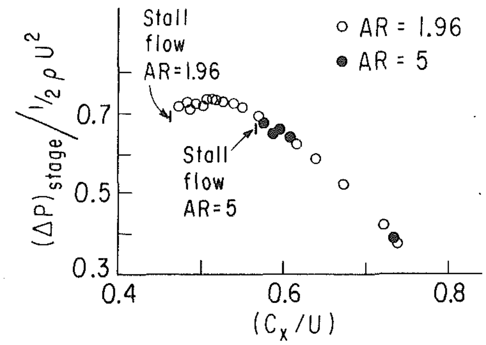

Fig. 4 Performance of high and low aspect ratio compressors (data of Smith [9])

problem to rest. Indeed, a comment in an overview of this topic by a representative of one of the engine companies states flatly "we still cannot predict accurately the onset of engine instability" [14]. (However, it should be recognized that requirements for stall prediction in aircraft gas turbine engines may be quite stringent compared to those in other areas of axial compressor usage.)

Although it is important to mention these correlations, and to give several relevant references, the main emphasis in this review is on the fluid dynamic phenomena associated with compressor stall, and the correlative procedures do not really provide insight into this aspect. Therefore we now consider approaches to the stall prediction problem, which are tied more closely to the flow instabilities occurring in the compressor. These can be divided into two types: linearized (small disturbance) stability analyses for predicting the *onset* of rotating stall, and nonlinear treatments which follow the growth of small perturbations to a fully developed (finite amplitude) state.

Stability Analyses for Rotating Stall Onset. The first of the linearized stability analyses was by Emmons [3], who investigated the conditions under which a small amplitude, circumferentially nonuniform, two-dimensional flow perturbation in a cascade would grow. Since then there have been many extensions of this investigation, of which two recent examples are the papers by Nenni and Ludwig [15] and by Fabri [16]. An excellent summary of the earlier work is given by Emmons, Rockett, and Kronauer [17] and a comprehensive bibliography up to 1967 has been compiled by Fabri [18]. A useful overall introduction to the topic is given by Stenning [19].

In these analyses a small amplitude perturbation is superimposed on a given mean operating condition of the compressor or cascade. The linearized equations of motion are solved to yield the forms of the flow perturbations in regions upstream and downstream of the compressor blade row (cascade). The wavelength of the flow nonuniformities is taken to be much larger than the blade pitch (as is experimentally found to be the case) so that an actuator disk[7] model of the cascade can be used. Suitable matching conditions are applied across the cascade to link the flow quantities upstream and downstream. In general the conditions used (for the two-dimensional case) have been continuity of mass across the cascade, an inlet/exit flow angle relation, and an inlet angle/total pressure loss relation, although some investigators have found it more convenient to work in terms of circulation and shed vorticity [15] rather than total pressure. From this procedure one can determine the eigenvalues of the system of equations which define the stability of the flow field, or one can examine the growth in time of an initially prescribed small perturbation (for times until the disturbances become too large for the linearized theory to apply).

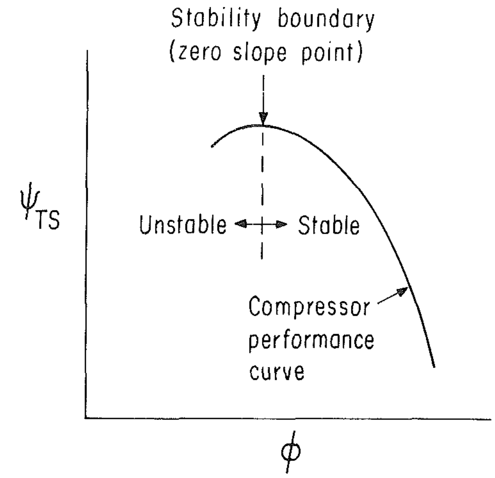

Fig. 5 Onset of compressor flow instability (rotating stall inception point)

These calculations yield results in terms of the critical *slope* of a function such as the mean compressor performance curve, or cascade loss curve, etc. at which the axisymmetric flow becomes unstable. However, these slopes are extremely difficult to obtain accurately, especially for the cases of greatest interest—multistage compressors or transonic fans—and this is a major reason for the relatively little usage of these methods by designers. In other words, if one wishes to use one of the instability calculations as a predictive tool, one must be able to predict the *slopes* of the compressor constant speed characteristic at off-design conditions, and, at present, this cannot be done with an adequate degree of precision.

Putting aside this difficulty, there is still the question of how well existing stability criteria describe the onset of rotating stall. As an example let us consider one of the best known of the (two-dimensional) criteria, due initially to Dunham [20], which states that rotating stall inception will occur at the peak (zero slope point) of the exit static pressure minus inlet total pressure compressor characteristic. This is illustrated schematically in Fig. 5. In this figure the horizontal axis is axial velocity parameter, ϕ, $(=C_x/U)$, and the vertical axis is nondimensional pressure rise, $\psi_{TS} = ([P_{exit} - (P_T)_{inlet}]/\rho U^2)$.

This criterion, which has also been derived by other investigators, has been applied with some success and does appear to furnish a rough "rule of thumb." However, counter examples in which it does not hold can readily be found. As illustration of this, Fig. 6 shows data from a representative sample of low speed multistage compressors.[8]

Curves V, VI, and VII do appear to show approximately zero slope (within the accuracy of the data), but curves I-IV have a negative (i.e., stable) slope right up to the stall points. This situation, where there does not appear to be a zero slope region of the compressor characteristic, is even more apparent in high speed multistage compressor data. (The author has not seen any multistage data in which the converse is true—i.e., in which operation significantly on the positive slope part of the characteristic is found.)

Having said this, however, it should be emphasized that in the opinion of this author these types of stability calculations can still be useful. This is true on several counts, not the least of which is that they provide an overall physical un-

[7] Defined as a representation of a blade row as a plane across which the mass flow is continuous but the total pressure, circumferential velocity, pressure, etc. can be discontinuous.

[8] Three Stage Compressors: I, II, V, VII, [21]; IV [22]; VI [23]; Four Stage compressor III, [24].

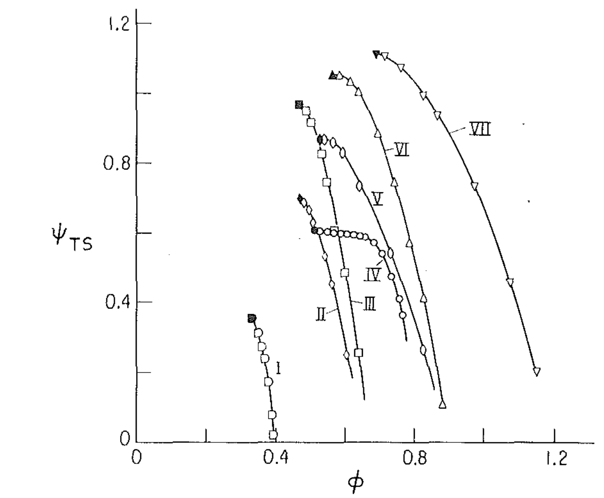

Fig. 6 Multistage compressor performance (low speed rigs); references identified in footnote 8

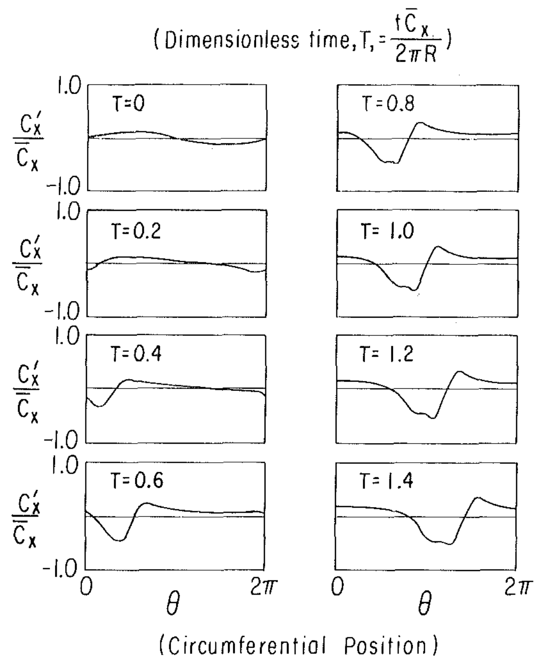

Fig. 7 Growth of small disturbance (calculations of Takata and Nagano [25])

derstanding of the breakdown of the axisymmetric flow in axial pumps or compressors.

An important advantage of the stability analyses is that they can be used to show effects that correlations do not predict. One of these effects is the coupling, or interaction, between blade rows, between different compressor spools, or between a compressor and components downstream in the compression system. Such coupling is much stronger with asymmetric flows than in the axisymmetric situation. This feature, which is central to the understanding of nonaxisymmetric flows in compressors, occurs because of the much larger length scales of the flow nonuniformities than in the axisymmetric situation. Specifically, it is the "single lobed" (i.e., one per circumference) flow perturbation which appears to be of most practical interest in multistage compressors. For disturbances of this type the relevant length over which adjoining rows can be considered to interact is of the order of the radius of the machine. Thus, adjacent blade rows, or even the two spools of a gas turbine engine, can be considered to be closely coupled as far as circumferentially nonuniform flow is concerned, and, as emphasized by Dunham [20], it is not correct to analyze the stability of one row in a compressor without considering the influence of the other rows. This coupling can be taken into account by the stability analyses and although, as stated, they may not be quantitatively precise, they can still give valuable guidance as to when row-row, spool-spool, etc., interactions will be important.

Nonlinear Investigations of Rotating Stall Onset and Growth. Although the linearized analyses have, in the past, been used to try to predict some of the features of fully developed rotating stall, it has become clear that in this flow regime the stall cells are definitely not small perturbations and linear analysis is inapplicable. Thus this type of investigation is only useful for the problem of stall inception, and to follow the subsequent development of the rotating stall, one must use a nonlinear model. This has been done by Takata and Nagano [25], as well as by Orner [26]. These models use time marching techniques to determine whether a small disturbance will grow or decay by calculating the development of the flow to some eventual steady-state solution. This could consist of a flow with a large amplitude disturbance propagating around the compressor, which is taken to be indicative of compressor operation in rotating stall.

So far these calculations have only been applied to an isolated rotor or to a single stage. (It should be noted that these procedures can give no more information than the linearized stability analyses in regard to stall inception in a uniform flow. However, they do seem to offer potential for assessing the effect of inlet distortion on stall inception, as is discussed below.) An example of the results of this type of calculation for an isolated rotor is given in Fig. 7 taken from Takata [25]. The figures show the normalized axial velocity nonuniformity, $C_x'/\bar{C}_x$, versus circumferential position, θ, at different nondimensional times, T, = (time • $\bar{C}_x$)/$2\pi R$. It can be seen that the initial sinusoidal perturbation grows into a large amplitude disturbance which propagates round the circumference.

Calculations of this type can show some of the features of rotating stall such as the hysteresis in the stall/unstall process. Other aspects, however, do not appear to be modeled very well, and a general comment on these nonlinear models is that, in spite of their complexity, many of the central features of the fully developed stalled flow may not yet be represented. For example, the present calculation procedures are either two-dimensional, assume that the flow occurs along axisymmetric stream surfaces so that flows perpendicular to these surfaces are supposed zero, or only include three-dimensionality in a potential flow representation. Experiments, however, indicate that strong radial flows can occur. In addition, the models of unsteady blade row performance that are used are quite rudimentary. They are essentially one-dimensional and rely heavily on extrapolation of the loss and turning characteristics (to the negative flow regime according to the experimental measurements). Further, the calculations are carried out with a constant mass flow, whereas in the actual situation the mass flow will change (along a throttle line) as the flow develops from unstalled to fully developed rotating stall. In addition, the steepness of the downstream throttle characteristics can also have significant effect [27]. In view of the path dependent behavior shown by some features of rotating stall (i.e., the hysteresis) it may be that these latter aspects should also be taken into account.

Because of the above, the author feels that while it is useful to pursue calculations of this type, they should be closely compared with experimental data to assess the effects of the assumptions. For example, one result of these calculations which is not satisfactorily explained is that for an isolated rotor the number of cells that appears from the calculation is equal to the number of lobes in the initial perturbation,

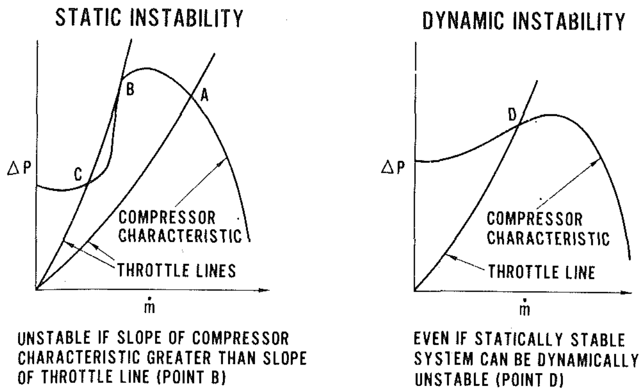

Fig. 8 Types of compression system instability

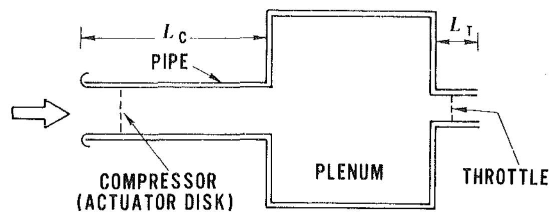

Fig. 9 Equivalent compression system for stability analysis

whereas tests with isolated rotors have shown that the number of cells is rather a function of the overall mass flow through the stage [28].

Overall Compression System Instability

We have so far discussed the prediction procedures for the inception of rotating stall in a compressor with uniform inlet flow, and we can now examine the criteria for the onset of the more global type of instability leading to surge. This is a system phenomenon and one in which, in contrast to rotating stall, the compressor appears to participate in an essentially one-dimensional manner.

We can sketch briefly the basic mechanism for compression system instability. The simplest view of this, and one that is found in many texts, can be derived with reference to a system consisting of a compressor and downstream throttle. The operating point of the "system" is the intersection of the compressor and the throttle pressure-flow characteristic curves. From considerations of the pressure rise through the compressor and the drop through the throttle it can be shown that the system will become unstable when the slope of the compressor pressure rise curve is steeper than the slope of the throttle curve [29].

This *static stability* argument is illustrated on the left of Fig. 8. For a small perturbation in mass flow (a decrease, say), if the system is operating at point A a pressure imbalance will arise to cause fluid accelerations that return the system to operation at the initial point. Point A is thus a stable operating point. At point B, however, where the throttle line is tangent to the compressor characteristic, the pressure forces associated with a small decrease in mass flow will cause the system to depart further from the initial operating point, so that point B is an unstable operating point.

This criterion is, however, too simple to describe the real phenomenon, since it only considers the static stability of the system. In fact it is generally the *dynamic stability* criteria which are violated first.[9] Thus as indicated on the right side of the figure a compression system can be statically stable (according to the foregoing slope criterion) and still exhibit instability.

[9] The terms dynamic and static instability can be made more quantitative by the following illustration. Consider a simple second order system described by the equation

$$\frac{d^2x}{dt^2}+2\alpha\frac{dx}{dt}+\beta x=0$$

where α and β are constants of the system. If $\beta > \alpha^2$, the condition for instability is simply $\alpha < 0$, but, independently of α, instability will occur if $\beta < 0$. It is usual to denote these two types of instability as dynamic and static respectively. Static stability ($\beta > 0$) is a necessary but not sufficient condition for dynamic stability.

The basic analysis of the dynamic system stability was also by Emmons, et al. [3], who treated a lumped parameter system similar to that shown in Fig. 9. This consisted of a compressor operating in a duct of small flow through area, a plenum volume (where the flow had no appreciable kinetic energy), and a downstream throttle. Basic criteria were developed for the stability of this system. Since then other investigators have looked at extensions of this problem to predict the system instability point. Although there are differences between the various analyses, the general result that emerges is that the system will be unstable near the peak of the (constant speed) pressure rise/mass flow characteristic, at some slightly positively sloped point, which is generally well before the simple static stability criterion is violated [19].

Although the calculation of this system stability point can readily be carried out, once again we encounter a situation where the critical parameters are the slopes of the steady-state (uniform flow) compressor speed lines at the stall point and, as stated, there is difficulty in predicting these.

We have discussed the two types of instability, and it is useful to now relate them. The global (system) instability is a basically one-dimensional phenomenon, involving an overall, annulus averaged, (in some sense) compressor performance curve. For typical volumes, lengths, and throttle characteristics this must generally be slightly positively sloped for system instability to occur. We have also seen that the axisymmetric flow through a compressor can be unstable to two- (or three-) dimensional infinitesimal disturbances, and that this local instability marks the inception of rotating stall. However, the onset of this rotating stall is very often associated with a precipitous drop in the overall ("one-dimensional") pressure rise mass flow curve of compressor performance. In other words, the inception of rotating stall can lead to a situation where the instantaneous compressor operating point is on a steeply positively sloped part of the characteristic, with a consequent violation of the dynamic and/or the static instability criteria. In this sense, the onset of the local compressor instability can trigger the more global compression system instability. Because of this, the compressor designer does not generally differentiate between the two modes as far as the applications of the stall correlations are concerned.

Multi-Element Stability Models. Up to now the discussion of compression system stability has been based on viewing the compressor as a single element. However under some circumstances this can be too simple, since there can be differences between the mass flow perturbations at the front of the compressor and those at the rear due to compressibility. Thus, different investigators have extended the overall lumped parameter analysis and modeled the compressor in a stage by stage (or even row by row) method, with separate volumes and actuator disks for each stage. Mass, momentum, and energy balances are written for each of the stages. Examples of such investigations are references [30]-[34], and a discussion and comparison of some of the basic models can be found in [31].

The determination of the instability point for these models can be found either by solving for the eigenvalues or by (digital) simulation of the system transients. In addition these models can explore the effects of pulsations that are imposed on the compression system. However, as with the simpler models, "the compressor stability boundary is strongly influenced by the stage characteristics. The overall results are thus dependent on the accuracy of the stage characteristics . . ." [34]. Hence the ability to predict the *slopes* of the steady-state (uniform-flow) compressor speed lines *at the stall* point is a key item in the successful use of these techniques.

Inlet Distortion Effects on Stall Inception

The discussion so far has been confined to stability predictions with a uniform inlet flow, but critical situations for stall are often those in which an inlet nonuniformity or distortion is imposed on the compressor. For an aircraft engine, for example, distortions can arise from inlet separations, armament firing, aircraft maneuvers, or take off in cross wind. In any event, if we consider a compressor operating in nonuniform flow at the same overall mass flow rate as one in a uniform flow, it is clear that in the former case some parts of the blading will be working at more unfavorable conditions than in the latter, and that some sections of the blading will be closer to the stall point. The overall effect of distortion is therefore to degrade the overall performance of the machine and, in particular, to decrease the stall free range of operation. Since the effects of distortion on compressor stall onset can be substantial, it is useful to discuss the general fluid mechanics of compressor operation with nonuniform flow.

In the investigations that have been carried out on the effect of inlet distortions, the flow nonuniformities are commonly divided into radially varying steady-state, circumferentially varying steady-state, and unsteady distortions. In reality the distortions encountered are combinations of two or possibly all three of these types, but significant progress has been made using the above simplifications since, in many situations, the principal loss in stall margin can be regarded as due to one of the three. For example, during changes of aircraft attitude the inlet distortion may vary significantly with time over a time scale of the order of tenths of seconds or longer, which is many times longer than the time for a fluid particle to move through the compressor. The distortion can therefore be considered as if it were steady-state. In analyzing the response of an axial flow compressor to an inlet flow distortion, it is also essential to recognize the strong interaction that exists between the compressor and the distorted flow field. Put another way, the compressor does not passively accept the distortion but plays an active role in determining the velocity distribution that will occur at the compressor face, which is what the individual compressor airfoils actually respond to.

We will very briefly discuss the pure radial distortion first. Typically this is analyzed using the axisymmetric procedures that have been developed for a uniform inlet flow, i.e., the distortion is essentially viewed as an off-design situation. (In doing this, one comment that can be made is that distortions encountered in practice can have substantially more severe gradients than do flows that arise solely from off-design operation. Hence, the calculations may have to be carried out using stations within the blade rows as well as at inlet and exit). The criteria that would then be applied to predict the onset of stall are based on the correlations referred to earlier. Therefore this type of distortion really does introduce any new phenomena from those mentioned previously, and we will not discuss it further.

With circumferential distortion, however, the asymmetry of the flow does introduce a new element into the fluid dynamic analysis of the compressor behavior. There has been a large amount of work on this topic. Much of it has been carried out treating the distortion as a small amplitude perturbation and using a linearized approach. Such an analysis can give very useful physical insight as well as quantitative information about the performance of the compressor in a circumferentially nonuniform flow. In particular it is able to show clearly the role of the various design parameters and operating conditions on distortion transfer, [10] to demonstrate the manner in which the upstream velocity field due to the distortion is substantially modified by the compressor, and to motivate the basic length rules for scaling the overall flow phenomena. Excellent examples of analyses of this type are the papers by Plourde and Stenning [35], Dunham [20], or Stenning [36], and these are recommended as an introduction to the field. (See also [37] for an extension of the basic ideas to compressible flow.)

From the point of view of *stability* prediction, however, the analysis of inlet distortion as a small perturbation can give no information as to the effect of inlet distortion on stability. This is because, in a linear system, the steady imposed distortion and the self-excited propagating perturbation (rotating stall) do not interact (i.e., the interaction is second order and thus not included in a linear analysis). We will therefore concentrate on those approaches which do attempt to assess the loss in stability due to inlet distortion. As in the uniform inlet flow situation, the approaches again range from correlations to more basic analyses.

The most empirical approach makes use of so-called "distortion indices" to describe the stability degradation to be expected from inlet distortion. The indices are based on the inlet total pressure distribution, and are a way to quantify the effects of a given total pressure pattern. For a given engine (or compressor) a correlation will be built up by testing with different types of inlet distortion to give the maximum distortion that can be tolerated at a given corrected speed. Once this is done the stability margin can be assessed in various other situations by converting the total pressure distortion in that case into an equivalent distortion index.

Much effort has been spent on these correlations for engines and they have been developed to take into account the fact that the distortions in general have a radial distribution to them as well as can contain a reasonable amount of unsteadiness. Descriptions of the basic methodology are given by Hercock and Williams [38] and by Collins [39], which also contains a list of other references on the subject. We will not discuss these correlations much further although there is one point which should be mentioned. There are situations in which the inlet total pressure distortion can be varying with time so that the instantaneous distortion factor undergoes very rapid changes. It is found in these situations that disturbances which have a time duration of less than *roughly* a rotor revolution have little effect on compressor stability [39]. Thus, for these higher frequency distortions, the unsteady response of the compressor appears to be beneficial as regards the onset of instability. We will discuss this important point further below.

At a somewhat more fundamental level of assessing compressor stability with inlet distortion are the so-called parallel comperessor model and its extensions. Descriptions of the basic ideas and assumption of this model are given by Reid [40], Stenning [36], and Mokelke [41]. The approach is to view a compressor operating in a circumferentially nonuniform flow as two compressors operating "in parallel" with each performing at a condition corresponding to the local value of mass flow. If the flow geometry downstream of the compressor is a constant area annulus, and the flow angle from the last stator is constant around the circumference, the two compressors can be taken to have the same exit *static* pressure. In addition, because there is very small opportunity

[10] The ratio of total pressure distortion at the exit to that at the inlet.

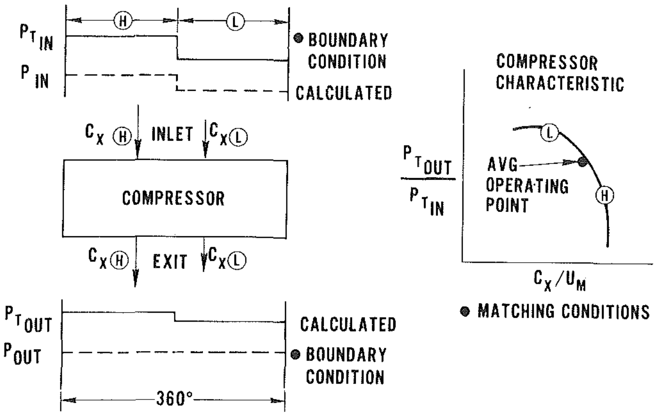

Fig. 10 Circumferential distortion analysis calculation procedure for 180 degree distortion

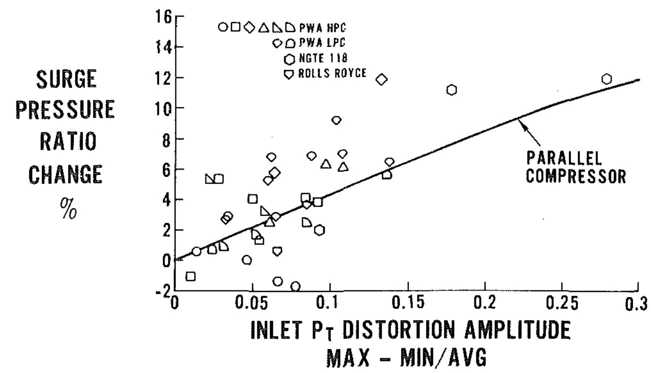

Fig. 11 Sensitivity to 180 degree circumferential distortion [45]

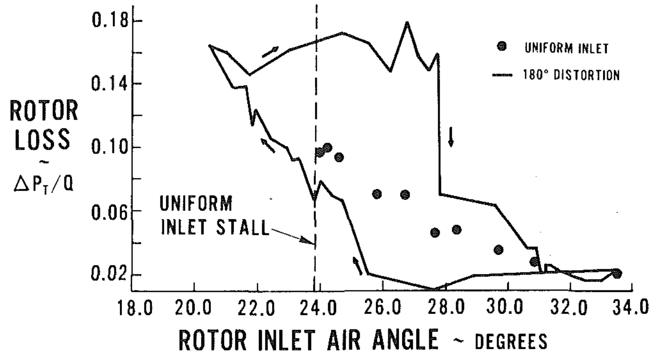

Fig. 12 Unsteady rotor response [45]

for circumferential cross flows within the compressor, the mass flow distribution at inlet and exit can be taken to be the same — i.e., there is no mixing of the flow in the two "segments" of the compressor. The overall mass flow, the two boundary conditions of specified local inlet total pressure distribution and constant exit static pressure, plus the idea that each segment is performing at some local point on the uniform flow characteristic, are enough to define the local mass flow distribution at the compressor inlet and thus the distorted flow performance. A schematic of this procedure is shown in Fig. 10 for a 180 degree square wave distortion in inlet total pressure. The model is not restricted to small amplitude nonuniformities, so effects due to the curvature of the compressor characteristic can be included. It should be noted that these models are also capable of treating inlet total temperature distortions as well as total pressure [41], [42].

In order to predict the stall point, however, some additional criterion must be invoked since the parallel compressor model is steady-state. The simplest is that the onset of rotating stall will occur when the flow in the low velocity region (point L in Fig. 10) is reduced to a level at which the machine will stall in a uniform flow. This concept, plus the basic parallel compressor ideas, leads one to the general design philosophy that steep speedlines are desired for distortion tolerant compressors [43], [44].

A comparison of the predictions of this model with test data is shown in Fig. 11, which presents the loss in stall pressure ratio as a function of the amplitude of the total pressure distortion, for 180 degree "square wave" distortions [45]. The solid line is the theoretical prediction given by the basic parallel compressor theory. It can be seen that although the model gives reasonable qualitative trends, on a quantitative basis the predictions leave considerable scope for improvement.

For a distortion of smaller extent this discrepancy is increased. For these distortions the theory predicts that there will be a greater loss in stall margin (compared to the uniform flow case) than is observed with the 180 degree distortion. (This is because the circumferential segment with the lowest axial velocity is assumed to be responsible for the stall, whereas the stall pressure ratio and flow are calculated as the average.) Experimental results are in substantial disagreement with this, however, since it is found that distortions of small circumferential extent have little effect on stall line.

The reason for this, and one of the major defects of the basic parallel compressor theory, appears to be associated with the assumption that the performance of the compressor can be described by the quasi-steady application of the local uniform flow performance curve. However, with a circumferential distortion, the rotor passes through a spatially nonuniform flow field and thus sees an unsteady flow. The response of the rotor can be considerably different than that predicted by a quasi-steady analysis, especially near the stall point. As one example of this, Fig. 12 shows the (relative) total pressure loss coefficient for a rotor operating in a 180 degree distortion [45]. The horizontal axis is inlet flow angle (measured from the tangential direction, so that decreasing β_1 means increasing angle of attack) and the vertical axis is the rotor total pressure loss coefficient. It can be seen that the loss coefficient exhibits an unsteady "loop" as a function of inlet angle as the rotor moves through the distortion. The values obtained in a uniform flow test of the rotor are also shown and it can be seen that there are substantial differences between the steady and unsteady performance.

A further very important point which is illustrated by this figure is that with distortion the rotor is operating transiently at a flow angle that is beyond the uniform flow stall point. That is, excursions to higher angles of attack (i.e., lower flow rates) than could be tolerated with a uniform flow can be tolerated locally (transiently) in the distorted case. This is in direct contrast to the hypothesis used in the basic parallel compressor model for defining the stall point, and the unsteady response appears to be an important feature to include in developing predictive methods for circumferential inlet distortion.

There are several different approaches to doing this. In the first, the idea of a critical angle of distortion has been developed [38], [40], [44]. This is essentially an angular extent of distortion below which "further loss of stall pressure ratio for a given distortion intensity is negligible" [40]. Thus, the procedure is to base the inlet distortion intensity on the average total pressure in a sector, of the extent of the critical angle, surrounding the lowest pressure. This constitutes an extension of the basic parallel compressor stall line hypothesis to account for unsteady response. However, it is necessary to evaluate the critical angle for each individual compressor, and quite different values have been reported in the literature [43]. In an alternative approach [45], a one-dimensional model of the unsteady rotor response is incorporated in a multisegment parallel compressor model (roughly thirty segments are used). This model can have some segments operating at flows below the uniform flow stall point.

Better prediction of stall than with the basic model can be achieved using such approaches. As an example, the use of one of these methods (the multisegment parallel compressor analysis) for prediction of the distorted flow stall line in a two spool turbofan engine is described by Braithwaite and Soeder [42]. It should be emphasized, however, that adequate definition of the uniform flow performance curve (including stall point) is a prerequisite for successful use of any of these methods.

The third class of studies of the problem of compressor stability in a distorted flow is the numerical solution of the time dependent nonlinear inviscid equations of motion. These describe the flow field from a specified initial state to a final "steady-state solution" similar to that described earlier with the uniform inlet flow. As with the uniform flow situation the analyses can either predict a steady flow or a large amplitude propagating disturbance, which is now modulated by the presence of the inlet total pressure distortion. They can therefore give an indication of the stability point of the distorted flow. So far these have been applied to single rows or stages and have not really been thoroughly assessed against experimental data. A good example of this approach is the work by Adamczyk [46], which has been extended to the compressible flow regime by Pandolfi and Colasurdo [47].

Again, one of the main limitations is the modeling of the dynamic blade row performance, especially at flows near and below the uniform flow stall point. There is not space in this review to discuss some of the various experimental and theoretical approaches to investigating unsteady blade row response. However, several useful review articles on unsteady flows have appeared recently and the interested reader is referred to these [48–50]. (In addition, one novel approach to obtaining data at flows below the stall flow is that described in [51].)

Compressor-Component Coupling. As stated previously, if one considers the behavior of compressors in asymmetric flow, the general conclusion that emerges is that for the type of flow nonuniformities which have the greatest effect on stability, namely those with a circumferential length scale of the order of the circumference of the compressor (i.e., a "one lobed" type of pattern with a strong first Fourier component), the compressor and the downstream components must be viewed as closely coupled. Hence the distorted flow performance of the compressor can be quite dependent on the downstream component.

An example of this is given in Fig. 13, which shows both predictions and experimental data from several research compressor experiments [23]. The data are from three different tests with an inlet circumferential total pressure distortion. In these tests the only thing that was altered was the exit section, one test being conducted with an annular exit nozzle downstream of the compressor, another with a constant area annulus exit section, and the third with an annular exit diffuser downstream of the compressor. The vertical axis shows the compressor distortion transfer, which is the ratio of the magnitude of the total pressure nonuniformity at exit to that at inlet, for the three different exit sections. It can be seen that the presence of different downstream ducts has a substantial effect on the compressor performance in distorted flow. This is quite contrary to the usual experience in axisymmetric flow where, for a given flow rate, the performance of an upstream component is only weakly affected by downstream conditions. Note that the results shown in the figure were obtained with downstream components that are all relatively passive fluid mechanical devices. With another

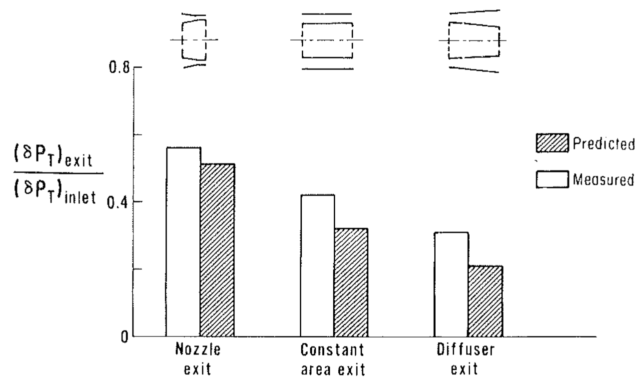

Fig. 13 Effect of downstream components on distortion transfer [23]

compressor downstream, as is usually the case in a modern multi-spool gas turbine engine, the effect of the coupling can be expected to be much stronger.

Coupled Compressor-Diffuser Flow Instability and Diffuser Instability. The above discussion of the coupling between a compressor and a downstream component furnishes a useful introduction to another aspect of the onset of instability in compression systems. Consider the compressor stability boundary for a nominally uniform inlet flow condition, i.e., the inception point of rotating stall. This occurs when conditions are such that a *circumferentially nonuniform* flow perturbation in the compressor annulus can grow. Based on the previous section, it might be expected that this point could be affected by components downstream of the compressor, since the downstream boundary condition on the flow perturbation is altered. The quantitative extent of changes in stability boundary is of course dependent not only on the nature of the downstream component, but on the circumferential length scale of the flow perturbation, since this also determines how closely the compressor and the downstream component are coupled [52]. For many situations of practical interest the predominant mode of instability occurs with a one-lobed type of disturbance so that the relevant axial distance within which there can be a strong interaction between components is on the order of the mean radius of the machine.[11]

This generally means that, in terms of the stability boundary, the compressor is *not* isolated from the influence of downstream components. More detailed analytic and experimental studies of this phenomenon bear out this general conclusion [53]. As noted before this has important application in analysis of two spool compression engines. It also emphasizes again that one cannot analyze the stability of a single row in a multistage compressor *by itself* as a guide to the compressor stability, but must include the other stages both upstream and downstream.

This type of interaction between the pumping element and the downstream components also can be strongly manifested in centrifugal compressors. In these situations the diffusers (vaned or vaneless) can be not only the cause of premature compressor stall, but can themselves exhibit rotating stall [54, 55]. In modern high pressure ratio centrifugal compressors, in fact, the initiation of system surge has been postulated to be linked to the instabilities in the diffuser [56].

A Digression on Unsteady Response. We have discussed the fact that the compressor blading responds in an unsteady manner, and that this departure from quasi-steady behavior appears to be quite significant near stall. However, there is a

[11] In other words, the coupling can only be neglected for axial separations that are significantly greater than this.

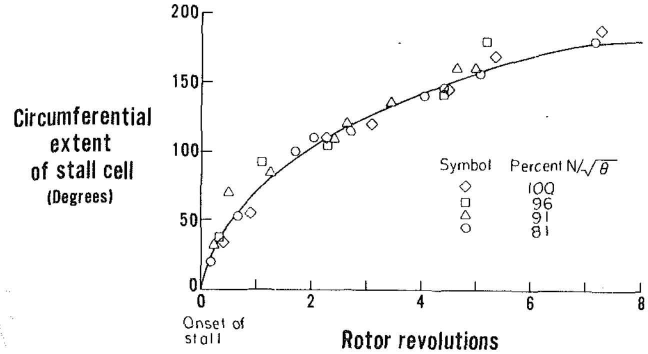

Fig. 14 Circumferential growth of stalled region (three stage compressor) [57]

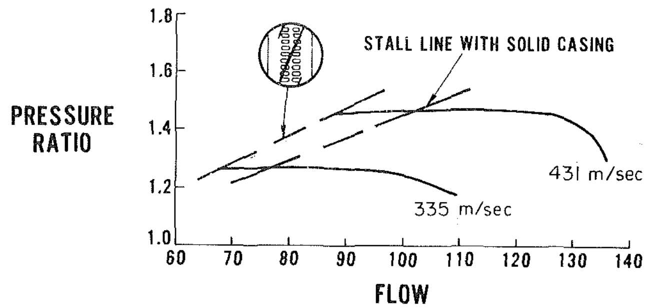

Fig. 16 Stall margin increase due to casing treatment [43]; flow in lbm/s

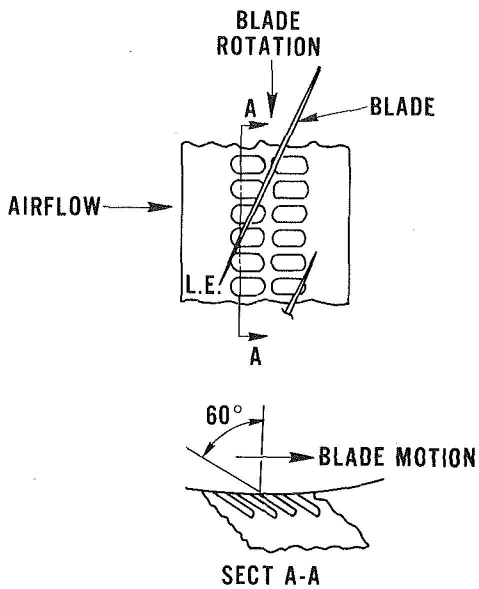

Fig. 15 Axial skewed groove casing treatment

larger scale aspect of the unsteadiness which is also associated with the onset of rotating stall.

The time scale that one would tend to associate with *blade* unsteady response is on the order of b/W where b is the blade chord and W is the relative velocity. However let us consider the time scale that characterizes the transformation of an axisymmetric flow to the severely nonuniform one of rotating stall. A representative time scale might now be the disturbance wavelength (e.g., the mean circumference of the machine) divided by the through flow velocity. This is the time scale used in Fig. 7 and is much longer than that based on blade unsteady response. A qualitative physical argument for this scaling can be made by noting that the change from unstalled to stalled flow involves the shedding of blade circulation of the same sign over a significant circumferential extent of the flow annulus. The flow will only approach a "fully developed" state when this shed vorticity has been convected downstream some distance on the order of the disturbance wavelength.

These rough considerations imply that for a given machine the stall cell growth time might scale with axial velocity (or since C_x/U is approximately constant at stall, with rotor speed) and circumference. This has been found to be the case in one of the few instances where data is available, as shown in Fig. 14. This response of the overall flow field is therefore likely to occur on a time scale an order of magnitude or more longer than that associated with the individual blades. The compressor performance (pressure rise, torque) during a stall transient may therefore differ substantially from the steady state performance. This facet of unsteady response, which may be quite important in the application of one-dimensional compression system models, for example, is another of the features of the stall process that are at present not well understood, and on which there is little data.

Stability Enhancement Using Rotor Casing Treatment

We have described some of the various types of instability, commented on the criteria for determining their onset, and discussed the adverse effects of inlet distortion on stall inception. However, another area which is of interest concerns techniques for enhancing the stability margin of a turbomachine. The most straightforward of these is to achieve the needed stability margin by matching the compression system below its peak efficiency point (in effect setting the match point so that the compressor blading has incidences and pressure rises far below the maximum). Although this would provide an increase in airfoil incidence range between the operating line and the stability limit, it would also lead to decreased efficiency on the (down-rated) operating line, which is generally unacceptable.

One solution to this problem is the use of so-called "rotor casing treatment" to improve the stability of compressors. This casing treatment consists of grooves or perforations over the tips of the rotors in an axial compressor or located on the shroud in a centrifugal machine. Numerous investigations of these types of configuration have been carried out under widely varying flow conditions (e.g., references [58-65]), and these have demonstrated that the range of usefulness of these casing configurations extends from compressor operation in basically incompressible flow (relative Mach numbers of roughly 0.15) to the supersonic flow regime (relative Mach numbers of 1.5). A schematic of one of the more successful of these casing configurations is shown in Fig. 15. These grooves are known as axial skewed grooves. A typical improvement in stall line brought about by use of these grooves is shown in Fig. 16 for a transonic axial fan. It is to be noted that far larger improvements have been seen and that casing treatment has also been used to inhibit instability in centrifugal compressors [66, 67].

Although the basic mechanism of operation of the casing treatment has not yet been fully elucidated, some important points have emerged. If we refer back to the description of the onset of rotating stall, it is apparent that for the axisymmetric flow to become unstable the blade passages must be operating at a condition such that they have the capacity to generate large blockages for small changes in inlet conditions. What is found, however, is that the level of blockage in the rotor passages can be greatly decreased due to the presence of these grooves. As an example of this, Fig. 17 shows data taken just downstream of an axial compressor rotor [68]. The measured quantity is the nondimensional relative frame total pressure loss across the rotor, i.e., the total pressure loss measured (by

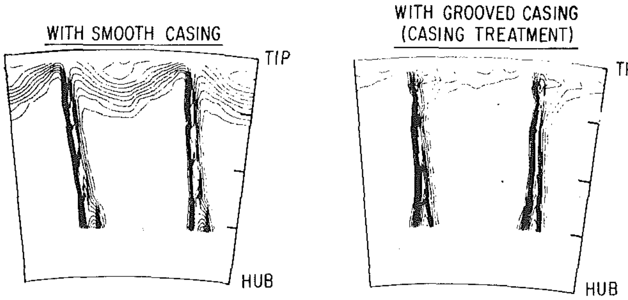

Fig. 17 Rotor exit total pressure loss contours near stall [68]

a probe rotating with the rotor) in a coordinate system fixed to the rotating blades. Data are shown for both solid wall and casing treatment, at a mass flow rate near the stall limit of the solid wall configuration. In this instance the onset of rotating stall appeared to be associated with a large blockage created near the end wall of the blade passage. The contours are increments of 0.05 in relative total pressure loss across the rotor divided by inlet dynamic pressure. It can be seen that with the casing treatment there is a much smaller region of low total pressure fluid and hence a greatly decreased blockage in the blade passage. One would therefore expect that the tendency towards rotating stall would also be greatly suppressed and the stable flow range of the machine extended with the grooved casing, which indeed was found to be the case.

These general ideas concerning the effect of the grooved casing on the flow blockage in the rotor passage can in fact be taken somewhat further. As described previously, the basic physical picture presented for the onset of rotating stall is one that is dependent on the level of flow blockage in the blade passages. In many instances, however, one can make a useful distinction between those situations in which the large blockages are associated with the endwall boundary layer and those in which the major part of the blockage is associated with the blade boundary layers away from the endwall. These two situations are referred to as wall stall and blade stall respectively.

Based on observations of widely varying effects of casing treatment on compressors exhibiting what appeared to be blade stall and those encountering wall stall, a series of experiments were carried out using two different rotor builds, one designed so that a blade stall was encountered, the other designed for a wall stall [68]. The rotors were tested both with and without casing treatment. It was found that the casing treatment markedly improved the stall range for the wall stall rotor (compared to the solid wall). In contrast, the blade stall rotor showed little change in stall point when the casing treatment was used. These experimental findings are consistent with the hypothesis that casing treatment is likely to be effective only in a situation in which a wall stall exists, and thus furnish one general guideline for its use.

The experimental results mentioned have shown some of the effects of the casing treatment on the overall flow field, but as stated, the mechanism of operation is still unknown. Different hypotheses have emerged but none is able to explain all the experimental results satisfactorily. As one example, several investigations have invoked the concept of a radial flow through the porous casing, and calculations carried out including this idea in the basic models for predicting rotating stall onset do show the possibility of a stabilizing influence of these radial flows [58], [65]. However, it should be noted that the analyses are based on the wavelength of the flow perturbations being much larger than the blade gap (i.e., treating the flow in the blade row on a passage averaged basis). While such radial flows can occur in situations where there is a plenum or external flow path, or where compressibility is important, experiments carried out using closed groove configurations at low speeds (where there can be no average radial flow into the casing over a sizeable fraction of the circumference) have also found substantial increases in stall margin. Thus while the "radial relief" idea may be important in some aspects of the problem it does not seem to be one of the most critical features.

In contrast to this passage average approach, it may be more useful to examine the flow in the grooves on a smaller scale, i.e., on a blade to blade basis. In particular consider the axial skewed groove configuration shown in Fig. 15. Note that there will in general be a higher static pressure over the rear part of the grooves than over the front part. One would therefore expect that fluid would be forced down into the groove at the rear and up as a jet at the front end. Such flows have been observed by Takata et al., who measured jet velocities that were comparable with free stream velocities [58]. In the frame of reference attached to the blade these jets have very high velocities (in a direction toward the pressure side of the blades). The flow in and out of the grooves is thus a potent device for the transfer of momentum between the moving wall and the low momentum flow in the boundary layer, thus increasing its total pressure (in the relative frame) and decreasing the blockage. In this connection it is relevant to note that data presented in [59] (from a compressor which was run with an axial skewed groove casing treatment) showed local blade surface pressures near the rotor tip that were higher than the relative total pressure. These overall considerations, however, have not yet been developed in a quantitative manner, and substantially more investigation is needed of the interaction between the flow in the grooves and that in the endwall region.

Although casing treatment is a potent remedy in increasing stall range, it should be emphasized that it is *not* a panacea. It has been found that those known treatments which have the most success in improving stall range generally have some penalty in efficiency associated with them. This situation gives a strong motivation to understand the basic fluid mechanics of operation of casing treatment in order to enable one to design optimum treatment configurations.

There are also situations in which casing treatment is unlikely to be successful. One has already been identified in the discussion of wall stall and blade stall, and a more obvious instance is when the stall occurs at the *hub*. In this situation, the flow shift towards the tip region, due to the casing treatment, can actually decrease the stall margin, since it will aggravate the hub stall condition. Finally, there can be mechanical and manufacturing difficulties associated with casing treatment usage in production engines. Designers have therefore been reluctant to adopt casing treatment as a standard item.

Behavior Subsequent to Instability Initiation

We have, until this point, been concerned primarily with the phenomena associated with the onset of instability. Another very important problem is associated with the behavior subsequent to the onset of instability, where one of two modes of system response, corresponding to surge or rotating stall, can be exhibited (see Fig. 3). For a system that goes directly into rotating stall, the transient behavior after the initial instability consists of a rapid decrease in pressure rise and flow, with eventual establishment of a new stable equilibrium point (in rotating stall) at a greatly reduced flow and pressure ratio. In surge, on the other hand, there is no new equilibrium point and both the mass flow and pressure rise undergo large amplitude oscillations.

For an aircraft gas turbine engine, a key requirement is recovery from stalled conditions once instability occurs. In this connection, therefore, it is important to note that even

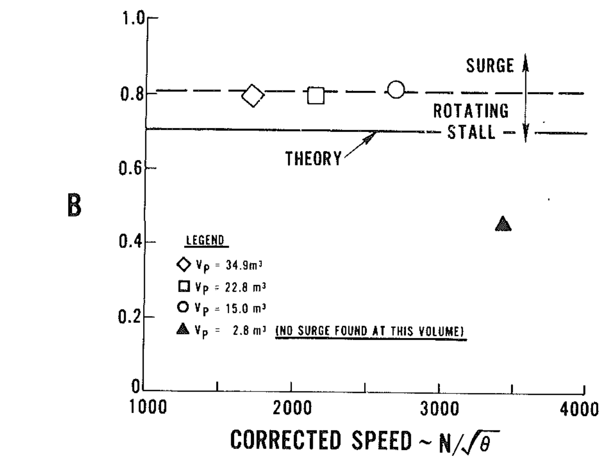

Fig. 18 Surge/rotating stall boundary for different plenum volumes [22]

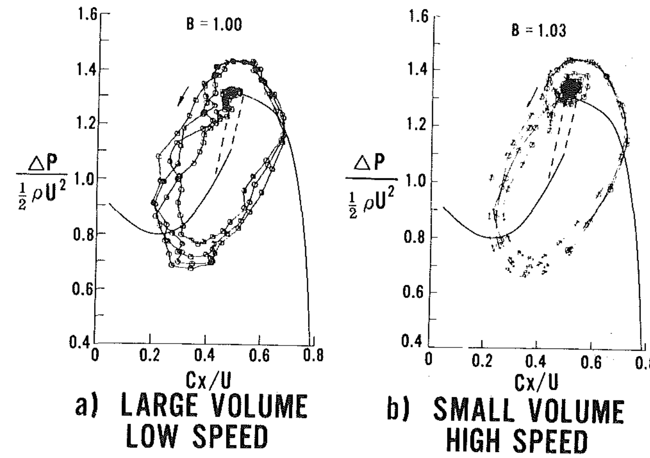

Fig. 19 Dependence of system response on B parameter [22]

though surge may be associated with severe transient stresses due to the large amplitude flow variations, the engine does operate in an unstalled condition over part of the surge cycle. Because of this, one is able to open bleed valves or make other changes that will have an effect on bringing the system out of surge. In contrast to this, rotating stall, and in particular the full span, large extent rotating stall which is characteristic of a multistage compressor, can be very difficult to recover from, as will be discussed in detail below. Therefore it is the surge mode that is much more favorable for recovery in systems such as aircraft gas turbine engines and is hence the more desirable one.

On the other hand, for an industrial centrifugal compressor or pump, it may be that operation in rotating stall (if one *must* encounter some sort of instability at some point in the operating range) can be tolerated, and that throttle movement is wide enough so that recovery is readily achieved. In this situation it may be that the often violent oscillations that occur during surge make this the less desirable of the two modes. Whatever the situation, however, it is apparent that an important question is whether a given pumping system will exhibit large amplitude oscillations of mass flow and pressure ratio (surge), or whether the system will operate in rotating stall where the annulus average mass flow and pressure ratio are essentially steady, but are greatly reduced from the pre-stall values.

To answer this, one must analyze the nonlinear system behavior. This was done by Greitzer [57] using the Helmholtz resonator type of compression system model introduced by Emmons (for the linear case) and pictured in Fig. 9. The analysis shows that for a given compression system i.e., specified compressor characteristic, plenum volume, compressor length, etc., there is an important nondimensional parameter on which the system response depends. This parameter is denoted as B:

$$B = \frac{U}{2\omega L_c}$$

where ω is the Helmholtz resonator frequency of the system, L_c is an "effective length" of the compressor duct, and U is the rotor speed. If we insert the definition of ω we can define B in terms of geometric and physical system parameters as

$$B = \frac{U}{2a}\sqrt{\frac{V_p}{A_c L_c}}$$

where a is the speed of sound, A_c is the compressor flow-through area and V_p is the plenum volume. For a given compressor characteristic there is a critical value of B which determines whether the mode of instability will be surge or rotating stall. Systems with B above the critical value (e.g., speeds above a critical value) will exhibit surge oscillations, while those having B lower than the critical (speeds below the critical value) will undergo an initial transient to the (annulus averaged) steady flow and pressure rise associated with operation in rotating stall.

Experimental evaluation of this concept is shown in Fig. 18, in which data is presented for a three stage compressor that was run with several different downstream volumes [22]. The horizontal axis is corrected speed and the vertical axis is the B parameter. The open symbols mark the experimental values of B at which the change-over from rotating stall to surge occurred for the different volumes tested. (The solid point indicates the highest value of B that could be obtained with the smallest volume used.) The regions in which one encounters surge (B above the critical value) or rotating stall (B below critical) as the mode of instability are also indicated. It can be seen that the prediction of a constant B as the boundary between surge and rotating stall is well borne out, although the value at which the change occurs is somewhat above that predicted by the theory. (It should be again emphasized that the value of 0.8 is for this particular compressor.)

Although it is necessary to carry out the calculations for any specific case, some motivation for this general result may be gained by the following qualitative physical arguments. Notice that we can write B as

$$B = \frac{(\rho U^2/2)A_c}{\rho U \omega L_c A_c}.$$

For a given compressor, the numerator, which is proportional to the magnitude of the pressure difference across the duct ($P_{\text{plenum}} - P_{\text{atmospheric}}$), represents the driving force for the acceleration of the fluid in the duct. If we consider oscillations that are essentially sinusoidal in character, the order of magnitude of the inertial forces that arise because of these local fluid accelerations will be given by the product of the fluid density, ρ, the frequency of the oscillations, ω, the amplitude of the axial velocity fluctuation and the volume of the fluid in the duct, $L_c A_c$. Hence if we assume that the axial velocity fluctuation is a specified fraction of the mean wheel speed the magnitude of these inertial forces will be proportional to the denominator. The ratio of the two forces (pressure to inertial) is therefore proportional to B. Thus as B is increased, for example by increasing the rotor speed, a fixed percent amplitude of the compressor axial velocity oscillations will result in inertial forces that are relatively smaller and smaller compared to the driving force due to the pressure differential. The capability to accelerate the fluid in the duct is thus increased as B increases. Hence as B becomes larger one would expect greater excursions in axial velocity and thus a general trend toward surge rather than rotating stall, and this is in accord with the experimental results.

The B parameter is useful not only in defining the boundary between surge and rotating stall but also as a guide to the scaling of the overall transient behavior of a compression system. In other words, for the same value of B, a compression system should exhibit the same transient behavior, regardless of whether this value has been obtained using a large volume and a low speed or a small volume and a high speed. Figure 19 shows a comparison of the measured transient response for two systems at approximately the same value of B, one with a large volume and low speed and one with a small volume and high speed, with the value of B high enough such that surge occurred [22]. The figures show the instantaneous annulus averaged axial flow parametr (C_x/U) versus nondimensional system pressure rise, i.e., the instantaneous system operating point. The steady-state compressor curves are also shown for reference. As can be seen the two surge cycles show extremely similar behavior, emphasizing the influence of B as the relevant nondimensional parameter for the phenomena under study. Although this figure does not show overall flow reversal, it should be emphasized that with high pressure ratio multistage compressors, periods of reversed flow during the surge cycle are to be expected.

We have said that the question of whether surge or rotating stall will occur is not only dependent on B but upon the compressor characteristic. A basic attempt to take into account one aspect of this, the number of stages (N) leads to the use of NB as the relevant parameter for scaling rather than B. The motivation for this is that for N identical stages and a given value of B the pressure rise will be approximately proportional to N, so that the ratio of pressure to inertia forces will increase as the number of stages. There is thus a strong trend toward lower values of B needed to encounter surge, rather than rotating stall, as the number of stages and overall pressure rise of the machine is increased.

Physical Mechanism for Compression System Oscillations. We have described the necessity for certain system parameters to be of a specified magnitude in order to encounter surge. One other aspect concerns the position of the compressor operating point in order to have surge cycles. In particular, in order for a surge cycle oscillation to occur, the compressor slope must be positive over some part of the surge cycle. The physical mechanism associated with this can be seen by considering a compression system undergoing oscillation about a mean operating point. Since dissipation is occurring due to the presence of the throttle, there must be energy put into the system to maintain this oscillation, and the only source available to do this is the compressor. Thus let us examine the mass flow and pressure rise perturbation through the compressor. These are shown in Fig. 20 which presents the *perturbations* in mass flow ($\dot{m}$) and pressure rise (ΔP) through the compressor, plotted versus time over a period of one cycle. The product of the two, $\delta\dot{m} \times \delta\Delta p$, which is proportional to the net *excess* (over the steady state value) of production of mechanical energy is also shown.

In the case of a positive slope it is found that favorable conditions for the energy addition occur since high mass flow rate and high rate of mechanical energy addition (in the form of pressure rise) go together. As shown in the figure the product of the two is positive definite over the whole cycle. Thus the net amount of mechanical energy that the compressor puts into the flow will be higher than if the system

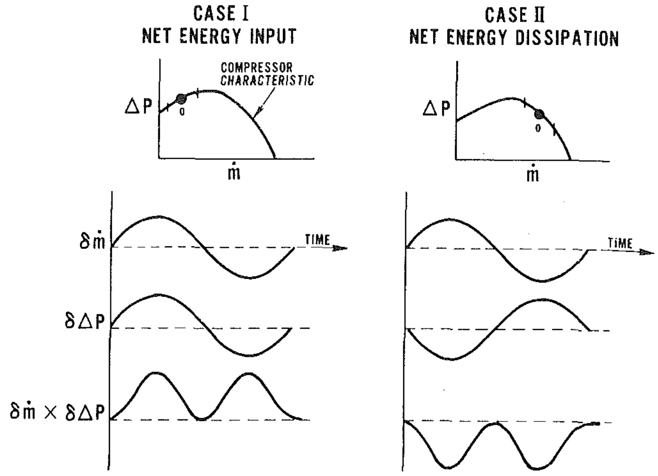

Fig. 20 Physical mechanism for surge oscillations

were in steady operation at the mean flow rate. (In a similar fashion the net dissipation due to the throttle will also be higher than if the system were in steady operation.) When the net energy input over a cycle balances the dissipation, a periodic oscillation can be maintained. In the case of an operating point on the negatively sloped region, as shown in the right-hand side of the figure, the compressor actually puts in less mechanical energy over a cycle than in steady operation, (since high mass flow is associated with low energy addition, and conversely) and no oscillations are possible. To summarize, (surge) oscillations are possible only when the mechanical energy input from the compressor is greater during the oscillatory flow than during a mean (steady) flow, and this can occur only if the characteristic is somewhere positively sloped so that high mass flow and high mechanical energy input per unit mass go together.

We have discussed some of the physical features of surge, and we now turn to examination of the other consequence of compressor instability – rotating stall. We will look at some of the overall features of this regime, then discuss some of the measurements on the structure of the flow field that have been carried out, and then examine a basic method for predicting compressor performance in rotating stall – particularly as regards the extent of the stall/unstall hysteresis loops.

General Features of Rotating Stall in Axial Compressors

Flow Regimes. It is useful first to describe the flow regimes encountered during compressor operation in rotating stall and to relate them to the changes in compressor performance that can be expected. Let us consider a hypothetical compressor, of, say, three stages, and examine the performance curves plotted in the form of ψ_{TS} versus ϕ, where ψ_{TS} = (exit static pressure – inlet total pressure)/ρU^2, and $\phi = C_x/U$. Two possibilities are shown in Fig. 21(a) and 21(b), which can be regarded as being representative of data from a number of actual compressor tests. Figure 21(a) shows a compressor whose performance curve is either continuous or has only a small discontinuity in pressure rise (a few percent) at the stall point, A. This behavior, where there is a very gradual drop in delivery pressure (or often no drop) at the inception of stall, is associated with the compressor operating with one or more stall cells that do not cover the total height of the annulus. This is known as part span stall, and an indication of a typical configuration is shown in the figure where we see two regions of severely retarded flow, i.e., two stall cells, at the tip. It is of course not always true that the cells appear at the tip; they can

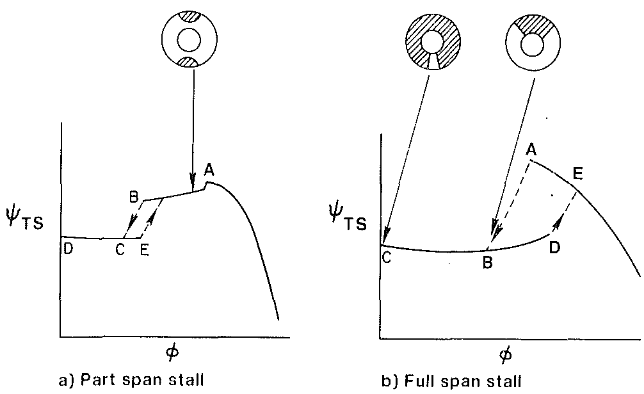

Fig. 21 **Compressor characteristics showing rotating stall regimes**

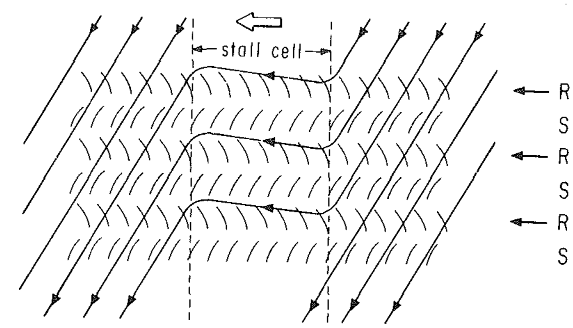

Fig. 22 **Sketch of stall cell structure (drawn in absolute reference frame)**

also be found at the hub, and the picture should just be regarded as giving one typical possibility.

Part span stalls are also often seen during low speed operation of multistage high pressure ratio compressors. Under these conditions, which can occur to the right of (i.e., below) the stall line shown in Fig. 1, there can be severe mismatching between the front and rear of the compressor, and the overall characteristic may thus still be quite negatively sloped.

As the throttle is closed from the stall point, and the mass flow through the compressor illustrated in Fig. 21(*a*) is further decreased, the performance curve can exhibit a large discontinuity where the pressure rise and mass flow jump to significantly reduced values; this occurs from point B to C on the figure. This jump is associated with a change in the type of stall. At point C there is one single cell, occupying a sizable fraction of the annulus and extending over the full annulus height. This regime is known as full span stall. Further throttling causes this cell to increase in size with the delivery pressure remaining relatively constant from point C to D. As the mass flow approaches zero, the stall cell can grow to fill the annulus so that the flow can become basically axisymmetric with the pressure rise often dropping off slightly. If we were to once again open the throttle, we would find that the mass flow at which the compressor left the *full span* stall regime, point E, was different from that at which it entered. However, this *hysteresis* is usually negligible for the onset and cessation of *part span* stall.

If we examine Fig. 21(*b*) we see a somewhat different picture. The large discontinuity in pressure rise and flow now occurs right at the stall limit (point A). The sharp drop in both these quantities as the operating point jumps from point A to point B in this figure is associated with the compressor going directly into single-cell, full span stall, as indicated schematically. Further throttling causes the stall cell area to grow, although it may not reach 100 percent of the annulus area even at zero mass flow through the compressor, with the pressure rise being relatively constant from point B to "shutoff" at point C. If one opens the throttle it is found that there is a substantial hysteresis between the onset and the cessation of stall, in that the throttle area has to be increased to a significantly larger value than that associated with stall onset, in order for the machine to unstall. It is in fact this large hysteresis which is responsible for the difficulty in recovering from a stalled condition.[12]

The two types of behavior at the stall limit point shown in Fig. 21(*a*) and 21(*b*) have also been termed progressive and abrupt stall respectively. This aptly describes the stall in terms of changes in the compressor output, since the former is associated with a relatively minor deterioration of the compressor performance from the unstalled state, but the latter can cause severe reductions in performance. On a fluid mechanical basis the occurrence of these different types of behavior is due to operation in different regimes of rotating stall. In order to predict the compressor performance subsequent to the onset of stall it is therefore important to be able to predict which type of rotating stall, part span or full span, will occur at the stall limit, since this is directly related to the occurrence and size of the hysteresis region.

Structure of the Stalled Flow Field. There have been many experimental investigations of the features of the flow fields associated with compressor performance in rotating stall. These have included flow visualization [69], studies of the three-dimensional velocity field [70, 71], as well as the use of instrumented blades to measure the time varying forces on the blades [72, 73].

Just as in the theoretical treatments of stall and stall inception, much of the detailed experimental work considers isolated rotors. However, it has become apparent that the constraints on the flow field in a multistage machine are quite different from those in an isolated rotor. For this reason we will concentrate here on one of the few investigations of rotating stall in a multistage environment, which was by Day and Cumpsty [21]. They used an ensemble averaging technique which was triggered on each passage of the stall cell so that they could average the results of many revolutions of the cell to provide detailed definition of stall cell properties. Their investigations were carried out on several different three stage compressors having quite different design values of axial velocity parameter, $(C_x/U)^*$, so the effect of this important parameter could be clearly seen.

Their paper, which addressed the full span stall problem, shows that in most cases the flow in the compressor can be divided into distinct areas of stalled and unstalled flow. To a reasonable approximation the flow in the unstalled area behaves as it would if there were no stall cell at all. In the stall cells the fluid velocities ahead of the rotors were near blade speed and in the direction of rotation, while behind the rotor the velocities were much lower. The axial velocities in the cells were small compared with either blade speed or the unstalled axial velocity; their precise magnitude and direction, however, varied from compressor to compressor. In addition it was emphasized that the full span stall cell extends *axially* through the compressor. The basic reason for this is that the axial spacing between blades is quite small compared to the width of the stall cell. Since a mass flow defect (which is what a stall cell is) in the front of the compressor can only be "filled in" by circumferential cross flows between the blade rows, the close spacing means that this defect will extend straight through the compressor (this is the same concept that is used in the modeling of circumferential distortion).

[12] This hysteresis between stall and unstall has, in the past, been held responsible for the existence of surge cycle oscillations. This is not correct and hysteresis in fact *inhibits* the occurrence of the surge cycles.

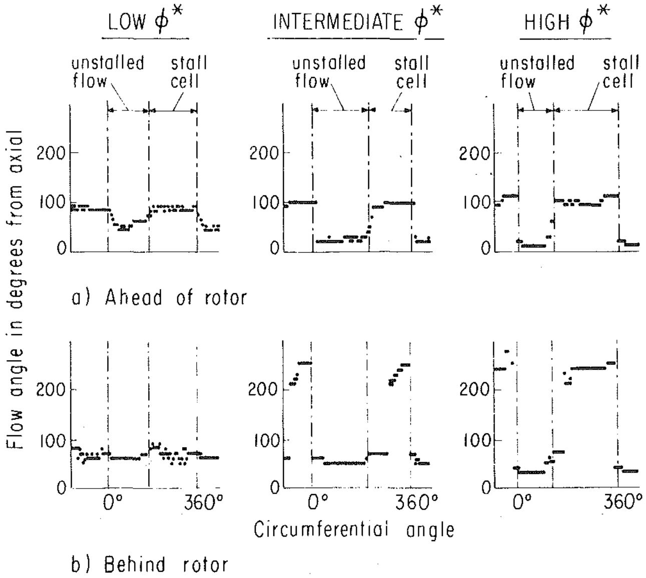

Fig. 23 Flow deflection measurements in three compressor builds [21]

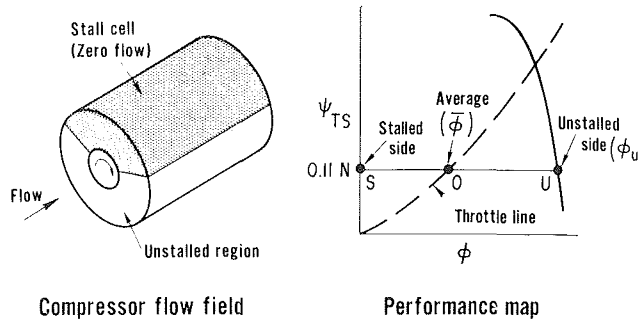

Fig. 24 Compressor operation in rotating stall

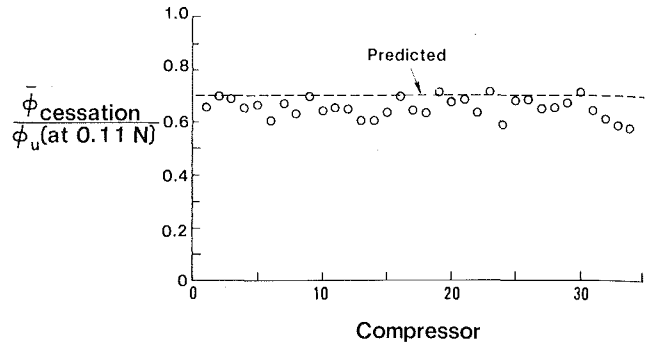

Fig. 25 Correlation for $\bar{\phi}$ at cessation of full span stall [27]; numbers on horizontal axis refer to individual compressors

Day and Cumpsty emphasize that the stall cell is very different from the wake of a bluff body, as many previous investigators have assumed. If this were the case, then in a coordinate system fixed to the cell the edges of the cell would coincide with the steady streamlines in the unstalled flow surrounding the wake. Such a description has been proposed by Rannie [74], for example, and has also been used as a model by Fabri for an isolated rotor [18]. The measurements, however, show clearly that were this to occur the cell would need to be helical, whereas it is essentially axial. The picture that emerges, therefore, of the instantaneous streamlines as viewed in an absolute coordinate system is shown in Fig. 22. There is, in fact, mass transport *across* the cell boundaries, so that fluid that was in the unstalled region (with a high axial velocity) is violently decelerated as it enters the cell, whereas particles which were in the cell are accelerated at the cell edges as they enter the unstalled flow. Therefore it should be emphasized that the "wake" models of stall cell flow cannot be applied to the single or multistage compressor situation.

Some idea of the magnitude of the changes in flow angle that can occur is shown in Fig. 23 which shows the circumferential distribution of flow angle (measured from axial) ahead of and behind the first rotor in a three stage compressor. The measurements were made at a radial location two thirds of the span from the hub. Data is shown from three builds — a low $(C_x/U)^*$ compressor, one with an intermediate value, and one with a high value. It can be seen that the stall cell flow is predominantly circumferential ahead of the rotor. Although not shown, this pattern is repeated stage-by-stage through the machine. In addition, because of the high values of swirl velocity, the centrifugal effects in the blade passages are very important. It is therefore suggested that analyses which do not model these effects may be more applicable to prediction of rotating stall in stationary rather than rotating blade rows.

Another important conclusion that was stressed in [21] is that "although the flow in a single stage build shows similarities with that in a multistage configuration of the same blading, the finer details of the flow are sufficiently different to preclude the extrapolation of single stage data to multistage configuration." The thesis by Day [24] discusses in detail these differences between the rotating stall behavior of isolated rotors (on which many previous investigations were focussed) and multistage compressors.

The emphasis in this discussion of rotating stall has been on the full span, large extent type, because this is the type which has the most severe effects on compressor performance, especially as far as the stall/unstall hysteresis. However, with regard to rotating stall in fans and single stage compressors (where part span stall is likely to be encountered) there have also been investigations of the part span stall regime [75], including one study of the stalled flow in a transonic compressor stage [76].

Prediction of Compressor Performance in Rotating Stall. The prediction of the compressor flow regimes in rotating stall is important for several reasons. One consideration can be that the speed and number of stall cells determine the forcing frequency for blade vibration and it is desirable to design the blades such that no resonance occurs. However, a very much more pressing consideration is that of stall *recovery,* i.e., the prediction of overall performance including the hysteresis loop.

In an approach to this problem Day, Greitzer, and Cumpsty [27] developed a correlation to predict stalled flow performance. The correlation is based on a heuristic fluid dynamic model of the compressor flow field in rotating stall. The compressor is divided into a stalled and unstalled zone analogous to the concept developed for inlet distortion. The stall cell is modeled by a zone of zero flow, and in the unstalled part of the flow the compressor is assumed to operate at a point on the unstalled performance curve. The situation is shown in Fig. 24 where we see the stalled "zone" of the compressor at S and the unstalled zone at the same pressure rise at U, with the mean operating point defined by the intersection of the horizontal line from S to U and the throttle curve at 0. This model is applied in conjunction with two experimental observations: 1) (following an observation originally due to McKenzie [77]) the nondimensional inlet total to exit static pressure rise per stage in rotating stall is constant *independent of the unstalled performance,* ($\psi_{TS} = 0.11N$ in stall, where N is the number of stages), 2) there is a critical value of stall cell blockage, λ, (i.e., extent of annulus covered by the cell) below which *full span* stall cannot exist,

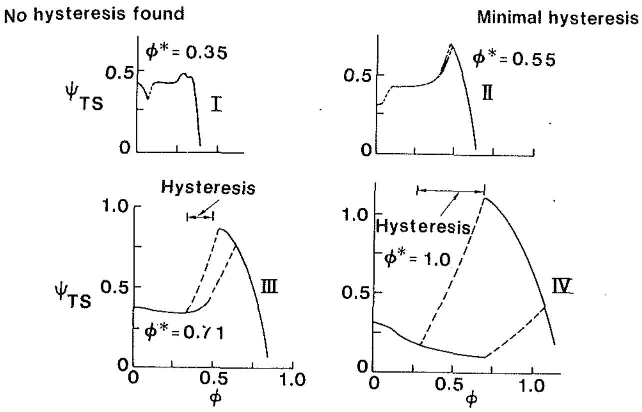

Fig. 26 Effect of ϕ^* (design C_x/U) on stall/unstall hysteresis (three stage compressors) [27]

(λ_{crit} = 0.3). Using these, one can make predictions about whether a given compressor will exhibit full span or part span stall as well as about the extent of the hysteresis loop.

To understand the central idea of the model, consider a compressor operating in the full span stall regime. To recover, we must open the throttle. Referring to Fig. 24, this would move point 0 to the right (points S and U remain fixed) as more of the annulus becomes unstalled and the blockage decreases. When the critical value of blockage is reached (at 0.3) full span stall can no longer exist and the compressor comes out of rotating stall. Since the blockage can be directly related to the ratio SO/SU, which is just $\bar{\phi}/\phi_{\text{unstalled}}$, by

$$\bar{\phi}/\phi_{\text{unstalled}} = 1 - \lambda,$$

we can obtain a formula for the value of $\bar{\phi}$ at stall cessation if we know the level of the horizontal line $\overline{SU}$, since this picks out $\phi_{\text{unstalled}}$ on the (known) unstalled part of the compressor characteristic. As stated, from experimental observations this can be placed at a level of $0.11N$ where N is the number of stages. Since the critical value of blockage at which the full span stall ceases is 0.3 we have, for stall cessation

$$\frac{\bar{\phi}_{\text{cessation}}}{\phi_{\text{unstalled at }0.11N}} = 0.7$$

This correlation is shown in Fig. 25. Data are presented from thirty-four different single and (mostly) multistage low speed compressors. It can be seen that the correlation furnishes a very useful method for predicting stall cessation.

Using this basic procedure, parametric studies can be carried out to determine the effect of different design parameters on stall performance. As shown in [27], two important parameters are $(C_x/U)^*$ (design value of axial velocity parameter) and number of stages. For a given design value of C_x/U the larger the number of stages the larger the size of the stall/unstall hysteresis. However, a more potent effect is the value of $(C_x/U)^*$; the higher the value for a given number of stages the larger the size of the stall/unstall hysteresis loop. Experimental evidence to support this idea is shown in Fig. 26 which presents data for four different three stage compressors with different values of $(C_x/U)^*$. At the lowest value of $(C_x/U)^*$ (curve I) no hysteresis could be found. As we examine curves II, III, and IV which have increasing values of $(C_x/U)^*$ we find an increase in the extent of hysteresis.

The procedure described in [27] also addresses the question of whether a given machine will exhibit part span stall or full span stall at the stall onset. Only one aspect of this will be pointed out here—namely that the same machine can develop either part span or full span stall and "the widely held view that part span and full span are the modes for low and high hub tip ratio machines is either wrong or else is a gross oversimplification" [21].

Research Needs and Suggestions for Future Work

We can now review those areas which the author thinks are fruitful to pursue. On the general topic of stall prediction in a uniform flow, there appears to be little more to be gained from purely two-dimensional linearized treatments, but three-dimensional aspects of stall inception are subjects on which work is needed. In particular, questions such as the impact of wall stall versus blade stall on the stall point, as well as stall onset in low hub/tip radius ratio compressors, have not been really resolved. Approaches that are based on treating the blade row or rows as a "black box," however, will still have to contend with the capability of present axisymmetric flow calculation procedures to predict the slopes of the uniform flow speedlines or loss characteristics near stall. Because of this it may well be that a numerical treatment of the flow in the blade passage region would be of considerable use here in clarifying how the blockage changes with small changes in incidence angle near the stall point.

For distorted flows there is also potential for research. To the present author, it seems that a disproportionate effort has gone into the relatively straightforward tasks of solving the inviscid flow equations outside of the blade row, while the simple models that are in use for the fluid dyamics *in* the blade row are those that were presented by Emmons [3] and Stenning [69] approximately twenty-five years ago. In fact, one still has to essentially sketch that part of the loss and turning characteristics associated with large angles of attack. Thus, the author feels that more effort should be spent concentrating on the more difficult problem of understanding the unsteady rotor response in the heavily loaded (near stall) regime. Probably both experimental and numerical investigations will be needed for this.

There is, in addition, a need to develop better stability criteria in the case of distorted inlet flows. As one example, it would be useful to attempt to formulate the linearized stability problem for a situation where the "mean flow" is a steady finite inlet distortion. Such an analysis might give some additional insight into the basic fluid mechanics of the onset of instability with the inlet distortion.

Although not discussed in the review, it should be noted that there are distortions that are characterized by large variations in inlet flow angles rather than by nonuniformities in total pressure. The inlet vortex (or ground vortex) is an important example of this type of distortion, which can have a significant effect on stability [78]. At present no theory exists to adequately describe this type of inlet nonuniformity.

On stability enhancement using casing treatment, there is considerable scope for useful work. As stated, the mechanism(s?) by which the treatment decreases the passage blockage is (are) still unclear and there is a need for both theoretical and experimental work on this topic.

There are also important areas which are associated with the behavior subsequent to the onset of the initial instability. More accurate models are needed to predict the local details of the surge phenomenon in multistage compressors. This is important from an aeroelastic standpoint as well as from purely aerodynamic grounds.

Prediction of the features of rotating stall in multistage compressors is also needed. We can discuss first the overall performance characteristic. The data reported in [27] and the subsequent discussion by Harman [79] seem to bear out the hypothesis that the inlet total to exit static pressure rise in rotating stall is roughly constant per stage independent of unstalled performance, but there is no understanding of why this occurs. Further, even though the pressure rise is independent of the unstalled performance, the torque does depend strongly on details of stall behavior related to compressor design parameters, and this is not understood. Investigation of these "global" properties should be pursued since they directly affect the hysteresis phenomenon and hence the recoverability of the compressor. A further facet of this topic is that for aircraft engines which have two or more spools, it is extremely important to understand the interactions between the high and low pressure compressors and their effect on stalled flow performance.

The central features of the stall cell structure also are not well modeled at present. Work is being carried out on this topic, but present models are still rudimentary, compared to the experimentally measured details of the flow. Here again an important facet seems to be the complex three-dimensional, separated unsteady flow in the blade passages. The blade row models in use for the calculation procedures are essentially the same as those used for the inlet distortion problem, however they are now being extrapolated even further (to the negative flow regime). Again, the author feels that more attention should be paid to the blade row characteristics, particularly in view of Takata's comment that the nonlinearities in the equations of motion do not seem to have an important role in determining the wave shape, disturbance amplitude, etc., but rather that these aspects are believed to be determined chiefly through the nonlinear effects due to the blade row characteristic [25].

Work also is needed in the area of overall compression system modeling, particularly for high pressure ratio multistage compressors. Systems which include two streams and two spools (i.e., turbofans) are an important application of this work. In this context, it should be noted that some of the transients associated with surge in modern gas turbine compressors can be on the same time scale as rotor rotation, or particle flow-through time. Under such conditions the compressor response can be far from a quasi-steady one. This is true for transients both from unstalled to negative flow as well as from negative flow to unstalled. Thus (again) if one wants to do significantly better than present models, the effort should be in trying to understand the overall unsteady performance of the blading.

As a summation, it is interesting to refer to a survey paper on aircraft engines written by Hawthorne over twenty years ago [80]. He stated: "Stall propagation is undoubtedly a phenomenon of the greatest importance to the engine aerodyamics and is one in which our understanding is still in the preliminary stage." It should be apparent from this review that the author agrees completely with the first part of this statement. As to the latter part, although much has been achieved, it should also be apparent that understanding, if not quite preliminary, is still far from complete.

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References

1 Iura, T., and Rannie, W. D., "Experimental Investigations of Propagating Stall in Axial-Flow Compressors," ASME *Transactions*, Apr. 1954, pp. 463–471.

2 Mazzawy, R. S., "Surge Induced Structural Loads in Gas Turbines," ASME Paper 79-GT-91, 1979.

3 Emmons, H. W., Pearson, C. E., and Grant, H. P., "Compressor Surge and Stall Propagation," ASME *Transactions*, Vol. 27, Apr. 1955, pp. 455–469.

4 Taylor, E. S., "Evolution of the Jet Engine," *Astronautics and Aeronautics*, Vol. 8, 1970, pp. 64–72.

5 Horlock, J. H., *Axial Flow Compressors*, Chapter 3, "Two Dimensional Cascades, Experimental Work," Krieger Publishers, 1973.

6 Leiblein, S., "Experimental Flow in Two Dimensional Cascades," Chapter VI in *Aerodynamic Design of Axial Flow Compressors*, NASA SP-36, 1965.

7 Leiblein, S., "Loss and Stall Analysis of Compressor Cascades," ASME *Journal of Basic Engineering*, Vol. 81, Sept. 1959, pp. 387–400.

8 Rodgers, C., "A Diffusion Factor Correlation for Centrifugal Impeller Stalling," ASME *Journal of Engineering for Power*, Vol. 100, Oct. 1978, pp. 592–603.

9 Smith, L. H., Jr., "Casing Boundary Layers in Multistage Axial Flow Compressors," in *Flow Research on Blading*, edited by L. S. Dzung, Elsevier Publishing Co., 1970.

10 Fligg, J. A., "Tests of a Low Speed Three-Stage Axial Flow Compressor at Aspect Ratios of One, Two and Four," AIAA Paper 66-613, AIAA Second Propulsion Joint Specialist Conference, 1966.

11 Smith, L. H., "The Effect of Tip Clearance on the Peak Pressure Rise of Axial Flow Fans and Compressors," ASME Symposium on Stall, 1958.

12 McNair, R. E., "Tip Clearance Effects on Stalling Pressure Rise in Axial Flow Compressors," ASME Symposium on Compressor Stall, Surge, and System Response, 1960.

13 Howell, A. R., and Calvert, W. J., "A New Stage Stacking Technique for Axial-Flow Compressor Performance Prediction," ASME *Journal of Engineering for Power*, Vol. 100, Oct. 1978, pp. 698–703.

14 Mikolajczak, A. A., "Introduction" to *Unsteady Flows in Jet Engines*, Project Squid/AFOSR/UARL Workshop at United Aircraft Research Laboratories, ed. by F. O. Carta, 1974.

15 Nenni, J. P., and Ludwig, G. R., "A Theory to Predict the Inception of Rotating Stall in Axial Flow Compressors," AIAA Paper 74-528, AIAA 7th Fluid and Plasma Dynamics Conference, 1974.

16 Fabri, J., "Growth of a Perturbation in an Axial Flow Compressor," ASME Paper 78-GT-30, 1978.

17 Emmons, H. W., Kronauer, R. E., and Rockett, J. A., "A Survey of Stall Propagation — Experiment and Theory," ASME *Journal of Basic Engineering*, Vol. 81, Sept. 1959, pp. 409–416.

18 Fabri, J., "Rotating Stall in Axial Flow Compressors," *Internal Aerodynamics (Turbomachinery)*, Institution of Mechanical Engineers, 1967.

19 Stenning, A. H., "Rotating Stall and Surge," Chapter 15 in *Fluid Dynamics of Turbomachinery*, ASME Lecture Course, Iowa State University, 1973.

20 Dunham, J., "Non Axisymmetric Flows in Axial Flow Compressors," Mechanical Engineering Sciences Monograph No. 3, Institution of Mechanical Engineers, 1963.

21 Day, I. J., and Cumpsty, N. A., "The Measurement and Interpretation of Flow within Rotating Stall Cells in Axial Compressors," *Journal of Mechanical Engineering Sciences*, Vol. 20, 1978, pp. 101–114.

22 Greitzer, E. M., "Surge and Rotating Stall in Axial Flow Compressors. Part II: Experimental Results and Comparisons with Theory," ASME *Journal of Engineering for Power*, Vol. 98, Apr. 1978, pp. 199–217.

23 Greitzer, E. M., Mazzawy, R. S., and Fulkerson, D. A., "Flow Field Coupling Between Compression System Components in Asymmetric Flow," ASME *Journal of Engineering for Power*, Vol. 100, Jan. 1978, pp. 66–72.

24 Day, I. J., "Axial Compressor Stall," Ph.D. thesis, Cambridge University Engineering Department, 1976.

25 Takata, H., and Nagano, S., "Nonlinear Analysis of Rotating Stall," ASME Paper 72-GT-3, 1972.

26 Orner, N., "Rotating Stall in Axial Flow Compressors," Von Karman Institute Lecture Series, "Unsteady Flow in Turbomachines," 1979, Brussels, Belgium.

27 Day, I. J., Greitzer, E. M., and Cumpsty, N. A., "Prediction of Compressor Performance in Rotating Stall," ASME *Journal of Engineering for Power*, Vol. 100, Jan. 1978, pp. 1–14.

28 Montgomery, S. R., and Braun, J. J., "Investigation of Rotating Stall in a Single Stage Axial Compressor," NACA Technical Note TN3823, 1957.

29 Hill, P. G., and Peterson, C. R., *Mechanics and Thermodynamics of Propulsion*, Addison-Wesley Publishers, 1965.

30 Tesch, W. A., Steenken, W. G., "J85 Clean Inlet Flow and Parallel Compressor Models," NASA Contractor Report CR-134978, Mar. 1976.

31 Corbett, A. G., and Elder, R. L., "Stability of an Axial Flow Compressor with Steady Inlet Conditions," *Journal of Mechanical Engineering Science*, Vol. 16, No. 6, 1975.

32 Corbett, A. G., and Elder, R. L., "Mathematical Modeling of Compressor Stability in Steady and Unsteady Flow Conditions," in *Unsteady Phenomena in Turbomachinery*, AGARD CP-177, 1976.

33 Jansen, W., and Swarden, M. C., "Compressor Sensitivity to Transient and Distorted Transient Flows," AIAA paper 71-670, 1971.

34 Willoh, R. G., and Seldner, K., "Multistage Compressor Simulation Applied to the Prediction of Axial Flow Instabilities," NASA TMX-1880, 1960.

35 Plourde, G. A., and Stenning, A. A., "Attenuation of Circumferential Inlet Distortion in Multistage Axial Compressors," *Journal of Aircraft*, Vol. 5, No. 3, 1968.

36 Stenning, A. A., "Inlet Distortion Effects in Axial Compressors," Chapter 14 in *Fluid Dynamics of Turbomachinery*, ASME Lecture Course, Iowa State University, 1973.

37 Mokelke, A., "The Prediction of Steady, Circumferential Pressure and Temperature Distortions in Multistage Axial Flow Compressors," ASME Paper 79-GT-184.

38 Hercock, R. G., and Williams, D. D., "Aerodynamic Response," in *Distortion Induced Engine Instability*, AGARD Lecture Series, No. 72, 1974.

39 Collins, T. P., "Engine Stability," Chapter 21, in *The Aerothermodynamics of Aircraft Gas Turbine Engines*, edited by G. Oates, AFAPL-TR-78-52, 1978.

40 Reid, C., "The Response of Axial Flow Compressors to Intake Flow Distortion," ASME Paper 69-GT-29, 1969 ASME Gas Turbine Conference.

41 Mokelke, H., "Prediction Methods," in *Distortion Induced Engine Instability*, AGARD Lecture Series, No. 72, 1974.

42 Braithwaite, W. M., and Soeder, R. H., "Combined Pressure and Temperature Distortion Effects on Internal Flow of a Turbofan Engineer," Paper presented at AIAA/SAE, 15th Joint Propulsion Specialists Conference, 1979.

43 Mikolajczak, A. A., and Pfeffer, A. M., "Methods to Increase Engine Stability and Tolerance to Distortion," in *Distortion Induced Engine Instability*, AGARD Lecture Series, No. 72, 1974.

44 Williams, D. D., and Yost, J. O., "Some Aspects of Inlet-Engine Flow Compatability," *Aeronautical Journal*, Sept. 1973, pp. 483-492.

45 Mazzawy, R. S., "Multiple Segment Parallel Compressor Model," ASME *Journal of Engineering for Power*, Vol. 99, Apr. 1977, pp. 288-296.

46 Adamczyk, J. J., "Unsteady Fluid Dynamic Response of an Isolated Rotor with Distorted Inflow," AIAA Paper 74-49, 1974.

47 Pandolfi, M., and Colasurdo, "Numerical Investigations on the Generation and Development of Rotating Stalls," ASME Paper 78-WA/GT-5.

48 McCroskey, W. J., "Some Current Research in Unsteady Fluid Dynamics," ASME Journal of Fluids Engineering, Vol. 99, Mar. 1977, pp. 8-38.

49 Platzer, M. F., "Unsteady Flows in Turbomachines — A Review of Current Developments," from *Unsteady Aerodyamics*, AGARD CP-227, 1977.

50 Telionis, D. P., "Review – Unsteady Boundary Layers, Separated and Attached," ASME Journal of Fluids Engineering, Vol. 101, Mar. 1979, pp. 29-43.

51 Nagano, S., Machida, Y., and Takata, H., "Dynamic Performance of Stalled Blade Rows," Japan Soc. of Mech. Eng. Paper #JSME 11, presented at Tokyo Joint Int. Gas Turbine Conf., Tokyo, Japan, Oct. 1971.

52 Greitzer, E. M., and Griswold, H. R., "Compressor Diffuser Interaction with Circumferential Flow Distortion," *Journal Mechanical Engineering Science*, Vol. 18, Feb. 1976, pp. 25-43.

53 Greitzer, E. M., "Coupled Compressor-Diffuser Flow Instability," *J. Aircraft*, Vol. 14, Mar. 1977, pp. 233-238.

54 Jansen, W., "Rotating Stall in Radial Vaneless Diffusers," *ASME Journal of Basic Engineering*, Vol. 86, Dec. 1964, pp. 750-758.

55 Abdelhamid, A. N., and Bertrand, J., "Distinctions between Two Types of Self-Excited Gas Oscillations in Radial Vaneless Diffusers," ASME Paper 79-GT-58, 1979.

56 Toyama, K., Runstadler, P. W., and Dean, R. C., "An Experimental Study of Surge in Centrifugal Compressors," ASME Journal of Fluids Engineering, Vol. 99, Mar. 1977, pp. 115-129.

57 Greitzer, E. M., "Surge and Rotating Stall in Axial Flow Compressors. Part I: Theoretical Compression System Model," ASME *Journal of Engineering for Power*, Vol. 98, Apr. 1976, pp. 190-198.

58 Takata, H., and Tsukuda, Y., "Study on the Mechanism of Stall Margin Improvement of Casing Treatment," ASME Paper 75-GT-13, ASME Gas Turbine Conference, Mar. 1975.

59 Prince, D. C., Jr., Wisler, D. C., and Hilvers, D. E., "Study of Casing Treatment Stall Margin Improvement Phenomena," NASA CR-134552, Mar. 1974.

60 Osborn, W. M., Lewis, G. W., Jr., and Heidelberg, L. J., "Effect of Several Porous Casing Treatments on Stall Limit and on Overall Performance of an Axial-Flow Compressor Rotor," NASA TN D-6537, Nov. 1971.

61 Moore, R. D., Kovich, G., and Blade, R. J., "Effect of Casing Treatment on Overall and Blade-Element Performance of a Compressor Rotor," NASA TN D-6538, Nov. 1971.

62 Fabri, J., and Reboux, J., "Effect of Outer Casing Treatment on Stall Margin of a Supersonic Rotating Cascade," ASME Paper 75-GT-95, ASME Gas Turbine Conference, Mar. 1975.

63 Tesch, W. A., "Evaluation of Range and Distortion Tolerance for High Mach Number Transonic Fan Stages, Task IV Stage Data and Performance Report for Casing Treatment Investigations, Volume I," NASA CR-72862, May 1971.

64 Bailey, E. E., "Effect of Grooved Casing Treatment on the Flow Range Capability of a Single-Stage Axial Flow Compressor," NASA TM X-2459, Jan. 1972.

65 Horlock, J. H., and Lakhwani, C. M., "Propagating Stall in Compressors with Porous Walls," ASME Paper 75-GT-59, 1975.

66 Amann, C. A., Nordensen, G. E., and Skellenger, G. D., "Casing Modification for Increasing the Surge Margin of a Centrifugal Compressor in an Automotive Turbine Engine," ASME *Journal of Engineering for Power*, Vol. 97, July 1975, pp. 329-336.

67 Jansen, W., Presentation at Panel on Centrifugal Compressor Efficiency Improvement, ASME Gas Turbine Conference, 1979.

68 Greitzer, E. M., Nikkanen, J. P., Haddad, D. E., Mazzawy, R. S., and Joslyn, H. D., "A Fundamental Criterion for the Application of Rotor Casing Treatment," ASME Journal of Fluids Engineering, Vol. 101, June 1979, pp. 237-244.

69 Stenning, A. H., and Kriebel, "Stall Propagation in a Cascade of Airfoils," *Trans. ASME*, Vol. 80, May 1958, pp. 777-790.

70 Dunham, J., "Observations of Stall Cells in a Single Stage Compressor," Aeronautical Research Council, C. P. 589, 1961.

71 Tanaka, S., and Murata, S., "On the Partial Flow Rate Performance of Axial Flow Compressor and Rotating Stall, First and Second Reports," *Bulletin of the J.S.M.E.*, Vol. 18, No. 117, Mar. 1975.

72 Sexton, M. R., O'Brien, W. F., and Moses, H. L., "An On-Rotor Investigation of Rotating Stall in an Axial Flow Compressor," in *Unsteady Phenomena in Turbomachinery*, AGARD CP-177, 1976.

73 Cossar, B. F. J., Moffat, W. C., and Peacock, R. E., "Compressor Rotating Stall in Uniform and Non-Uniform Flow," unpublished report, Cranfield Inst. of Technology, 1979.

74 Rannie, W. D., "The Axial Compressor Stage," Part F in *Aerodynamics of Turbines and Compressors*, edited by W. R. Hawthorne, Princeton University Press, 1964.

75 Breugelmans, "Flow Measurements in a Rotating Stall cell," from Von Karman Institute Lecture Series, "Unsteady Flow in Turbomachines," Brussels, Belgium, 1979.

76 Bartlett, F., and Greitzer, E. M., "Stalled Flow Performance of a Single Stage Transonic Compressor," M.I.T. Gas Turbine and Plasma Dynamics Laboratory Report 144, Dec. 1978.

77 McKenzie, A., Private Communication.

78 Motycka, D. L., "Ground Vortex - Limit to Engine/Reverser Operation," ASME *Journal of Engineering for Power*, Vol. 98, Apr. 1976, pp. 258-264.

79 Harman, J. W., "Discussion" (on the paper "Prediction of Compressor Performance in Rotating Stall,") ASME *Journal of Engineering for Power*, Vol. 100, Jan. 1978, pp. 13-14.

80 Hawthorne, W. R., "Some Aerodynamic Problems of Aircraft Engines," *Journal of Aeronautical Sciences*, Vol. 24, Oct. 1957, pp. 713-730.