Stall, Surge, and 75 Years of Research

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Work on rotating stall and its related disturbances have been in progress since the Second World War. During this period, certain "hot topics" have come to the fore—mostly in response to pressing problems associated with new engine designs. This paper will take a semihistorical look at some of these fields of study (stall, surge, active control, rotating instabilities, etc.) and will examine the ideas which underpin each topic. Good progress can be reported, but the paper will not be an unrestricted celebration of our successes because, after 75 years of research, we are still unable to predict the stalling behavior of a new compressor or to contribute much to the design of a more stall-resistant machine. Looking forward from where we are today, it is clear that future developments will come from CFD in the form of better performance predictions, better flow modeling, and improved interpretation of experimental results. It is also clear that future experimental work will be most effective when focussed on real compressors with real problems—such as stage matching, large tip clearances, eccentricity, and service life degradation. Today's topics of interest are mostly associated with compressible effects and so further research will require more high-speed testing. [DOI: 10.1115/1.4031473]

1 Introduction

Rotating stall and surge represent the breakdown of orderly flow through a compressor. In the past, these flow disturbances have variously been described as: "the most unfortunate occurrence to befall an aero-engine compressor"; "ranking among the most serious problem areas in turbomachinery aerodynamics"; "a violent phenomenon constraining the operating range of an axial compressor"; and "something to be avoided at all costs." These statements are all true, especially as the occurrence of stall or surge in a flying engine will lead to costly repairs and a loss of revenue. In keeping with the seriousness of the problem, much effort has been expended over the past 75 years in trying to understand and avoid these disturbances.

In our research, we have produced useful models of stall propagation, learned about stall cell structure, studied the install performance of compressors, and gained an understanding of stall/unstall hysteresis. We have also created analytic models which set out the limits of compressor stability, explained the divide between stall and surge, predicted (and subsequently measured) modal activity prior to stall, achieved active control of stall and surge, and used modern computational fluid dynamics (CFD) to highlight the role of leading edge separation and shed vorticity in the stall inception process. This is an impressive list of achievements, but at the end of the day, it must be said that all our efforts have only produced knowledge of a descriptive type. We can explain what happens when a compressor stalls or surges, but we have little to offer in the way of rigorous rules for designing a more stable compressor.

It is perhaps a bit unfair to suggest that we have not made much progress in our work to date because stall and surge are inherently difficult problems. There are many features of the stalling process, which are similar in all compressors and, as such, can be recognized and explained by our past efforts. There will, however, always be unexpected variations between one compressor and the next—particularly when a new design is attempted. This similarity and dissimilarity surrounding stall and surge mean that we can usually explain what happened after an event has occurred, but we can seldom predict what will happen before we start. Thus, there will always be more qualitative than quantitative aspects to stall research.

Against the idea that stall is such an intractable problem, we should note that there are thousands of flights takingoff and landing every day and yet very few stall events are reported. From this, it is obvious that the engine companies have the problem under control, but who should take the credit for this remarkable achievement? Should it be the researchers who have written thousands of papers on the subject of stall and surge, should it be the aerodynamicists who have designed well-matched compressors, the control engineers who keep the engine operating on the right side of the surge line, or should it be the structural engineers who produce round casing and tight clearances? Sadly, the stall workers are at the bottom of the list. This may seem harsh, but one of the objectives of this review is to take a cold hard look at what we have achieved over the years and then to consider what we might do in the future which will be more useful.

2 Background to Rotating Stall

At the outset, it should be mentioned that this paper will focus on axial compressors of the jet engine type. These compressors have received the greater part of our attention in the past and are very important from a commercial point of view. Centrifugal compressors are also economically important, but they often exhibit features which put them in a class of their own—for example when stall occurs in the exit diffuser or when the delivery pressure remains high during stall. Ventilation fans and other axial units with small numbers of blades and low solidities also behave in ways specific to their type and cannot be considered as representative of aero-engine compressors.

To introduce the topic of rotating stall, it will be useful to summarize the dominant features which characterize the stalling process of an axial compressor. (To aid the discussion, Fig. 1 shows a fixed-speed performance characteristic of a typical axial compressor. The inset sketch depicts an annulus occupied by a large full-span stall cell.)

(1) In a compressor of similar stage loading and flow rate to an aero-engine ("delta H upon U squared" of 0.4 and flow coefficient of 0.6), stall is always accompanied by a sudden loss of pressure rise—from C to D in Fig. 1.

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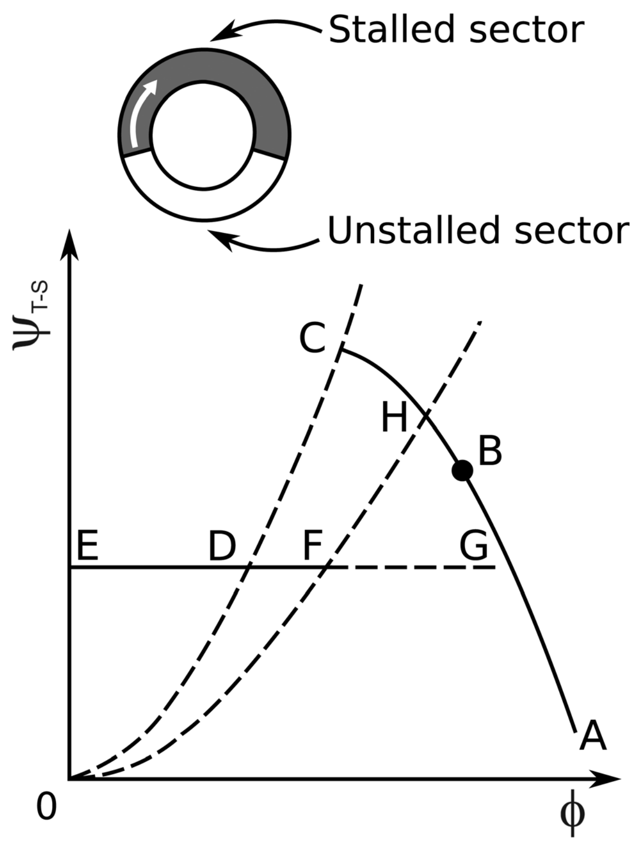

Fig. 1 Typical fixed-speed performance characteristic for an axial flow compressor

(2) The flow rate at which stall occurs, more-or-less, coincides with the peak of the total-to-static pressure rise characteristic, i.e., point C. (Note that the "stall margin" of a compressor is represented by the operating range between the design point, B, and the stall point, C.)

(3) The discharge throttle setting at which a compressor will unstall is always greater than that at which it first stalled. This gives rise to a hysteresis loop: C, D, F, and H in Fig. 1. The size of this loop increases with the design pressure rise and flow rate of the compressor.

(4) The flow in a stalled compressor will always divide into clearly defined sectors of stalled and unstalled flow.

(5) A stall cell will always rotate in the same direction as the rotor blades, but at a slower speed.

(6) Highly loaded blades of low aspect ratio (short blades) usually give rise to a single full-span stall cell. Long blades usually produce multiple part-span cells near the casing.

(7) When stall occurs, the flow breakdown process nearly always begins in the vicinity of the rotor tips.

(8) Stall inception may be initiated by long length-scale disturbances (modes; measured in terms of circumferential length) or by short length-scale disturbances (spikes; measured in terms of blade pitches). In between these two "pure" forms of stall inception there is a whole range of flow breakdown patterns which are neither of one type nor the other.

(9) Part-span cells rotate faster than full-span ones and, in general, the smaller the disturbance the quicker it will rotate.

(10) Prestall disturbances, i.e., disturbances occurring at flow rates greater than the stall point, are sometimes observed in the form of radial vortices or oscillations of the tip clearance flow. These disturbances are usually referred to as rotating instabilities and are most likely to be observed when tip clearances are large.

As a rule, the statements at the top of this list are always true. These are the rules dealing with the behavior of the compressor after stalling has occurred. As we move down the list into the realms of stall inception, we find flow features which occur with less certainty and finally, at the end of the list, we have the disputed area of prestall activity.

It is important to note that the uncertainty surrounding the stalling behavior of a new compressor disappears once the compressor has been stalled for the first time. A compressor which is stalled repeatedly will stall at the same flow coefficient each time, the number and speed of the cells will be the same each time, and the stall inception patterns will only differ in minor detail. We thus end up in the situation of not knowing what a newly designed compressor will do when it stalls for the first time, but once it has been stalled we know precisely what to expect after that. This is an example of hindsight being an exact science.

If, as has been suggested, there is so much uncertainty about the stalling behavior of a new compressor, what do the engine companies do to ensure the stall-free operation of their latest designs? Except under exceptional circumstances, the companies tend to develop new compressors starting from previous successful designs. They also draw on years of experience and proprietary rules of thumb. Occasionally, a radically new compressor is designed and then the risks of unexpected stalling behavior are greatly increased; witness the problems experienced with the V2500 and PW6000 compressors.

In the above discussion, the topic of surge has not been mentioned. This has been done partly in the interest of simplicity, but also because surge will be discussed separately in a later section. In addition, there is the philosophical point to be considered as to whether or not surge is a disturbance in its own right or just a consequence of rotating stall.

3 Seventy-Five Years of Research in Ten Episodes

In this section, a semihistorical approach will be used to introduce ten topics of research, which form the greater part of our collective knowledge of stall and surge. The intention is not to list all the work that has ever been done on a particular subject, but to focus on the birth of ideas and their development.

In all fields of research, progress is seldom made in sequential steps, but rather in a cloud of thoughts emanating from the minds of many people all concentrating on the same problem. One only has to think of Frank Whittle who, for a short period, was given all the credit for inventing the jet engine, but, as time has passed and more information has come to light, we have been forced to concede that work on the same idea was going on at the same time in Russia, Germany, and Switzerland. The idea of a jet engine was "in the air" so to speak.

In taking a historical approach, it is often interesting to discover that an idea we have traditionally attributed to one researcher was actually in existence many years before—though not always expressed in easily accessible terms. In this context, we are reminded of the scientist who was asked if he was the first to discover some or other phenomenon and he replied; "No, but I hope I am the last." By this he meant to say that now that he had explained the phenomenon properly it would not be "discovered" again. In the review which follows, many cases will emerge where an idea was in existence long before a definitive paper on the subject was written.

4 Early Work and Small Perturbation Theories

One of the earliest references to rotating stall comes from Prandtl et al. [1], who reported recirculating disturbances in a water pump. Shortly after this, Grünagel [2] published a paper on "circumferential pump pulsations." In this work, potential flow analysis was used to show that there are two possible stable flow conditions in a pump or compressor; uniform flow and circumferentially distorted flow (rotating stall). Other early work on the subject was directed at surge in piston-engine turbochargers; Brooke [3], but it was not until work on the jet engine began that rotating stall received urgent attention. Frank Whittle is cited by Stenning [11] as being the first to observe propagating stall in the inducer vanes of a centrifugal compressor. He is said to have held a firework at the entry to his compressor and seen the smoke move in a rotating pattern.

From the literature, it is clear that the late 1940s and early 1950s were a period of great advancement in compressor performance and efficiency. At that time, all the engine companies were suffering with problems of "pulsating flows and other forms of unsteadiness." As a result, Bullock and Finger [4] undertook detailed measurements of surge in a multistage compressor. They used early hot-wire techniques and home-invented pressure transducers (a light beam reflected from a pressure sensitive diaphragm) to record the unsteady flow in the compressor. Although no distinction was made between stall and surge, some interesting observations were reported. For example, the onset of instability was linked to the slope of the characteristic, stall/unstall hysteresis was observed, surge suppression by flow recirculation was demonstrated, and the influence of ducting volumes on surge behavior was reported: "a large external volume gave low-frequency high-amplitude pulsations; whereas a small volume resulted in high-frequency, low-amplitude pulsations." This last point sounds very familiar today, but it was reported 35 years before Greitzer's definitive work on surge.

At about the same time, Foley [5] summarized the proceedings of an interesting round table discussion on the topic of compressor surge. The meeting was attended by engineers from Pratt and Whitney, General Electric, Westinghouse, NACA, Alison, and Wright Aeronautical. This meeting dealt with many of the problems we still face today: the effects of installation ducting on the severity of surge; the extra complexity brought about by the need for variable guide vanes; the weight of interstage bleed valves; the effects of rotational speed on stage matching; the need for dual spool compressors, and the undesirable effects of deterioration, dirt accumulation, and production tolerances! The report also mentions a request made to control engineers for a device "which could anticipate the onset of surge and automatically reduce the fuel flow." At the meeting, it was agreed that there was no immediate hope for such a device.

Foley's report included an interesting illustration of stalled flow in a single-stage compressor; see Fig. 2. The clarity of this picture is remarkable because it was produced at a time when rotating stall and surge were only just being recognized as disturbances having different characteristics.

In early documents, the word surge was used to describe any disturbance which made a compressor "noisy." At the beginning of the 1950s, however, the first academic papers were published and from this point onward a distinction was made between stall cells propagating in the tangential direction and flow pulsating in the axial direction. The reason for mentioning the word "academic" here is because it was only when laboratory compressors, i.e., low-speed rigs of limited length, came into existence

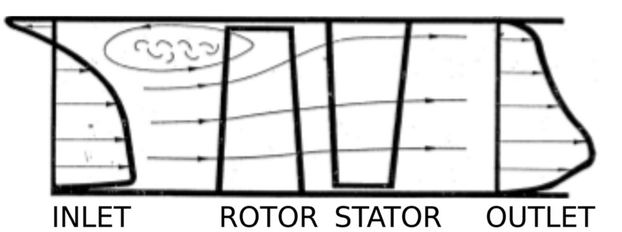

Fig. 2 Picture of part-span stall from Ref. [5]

that rotating stall and surge were identified as separate disturbances.

Around 1953, five important papers were presented at various conferences in the U.S.—all dealing with stability issues associated with stall and surge. At the time, blade failures caused by stall-induced fatigue were a problem and so the work had three clear objectives: to define a criterion for stability, to predict stall cell number, and to calculate stall cell speed.

Of the five papers, two were predominantly experimental while the others concentrated on modeling the stall propagation process. On the experimental side, Iura and Rannie [6] stressed the need for high-response instrumentation if propagating stall was to be detected other than by sound. They also noted that "as the average flow angle of a blade row approaches stalling incidence, the preferred flow pattern is one with groups of blades severely stalled alternating with groups of unstalled blades, rather than uniform stalling of all blades." (This is the same conclusion reached in 1932 in connection with centrifugal water pumps.) They also distinguished between rotating stall and surge on the basis of the average flow rate through the compressor; the average being steady for rotating stall and fluctuating for surge. From their measurements, they also reported fast moving multiple part-span cells and slower moving single full-span cells. From today's standpoint, this is familiar stuff, but someone had to be the first to make these observations.

The other experimental paper from 1953 was by Huppert and Benser [7], who concentrated on the problem of rotating stall in multistage compressors. They also distinguished between multi-cell part-span stall and single-cell full-span stall. A large part of their paper was devoted to the problem of stall margin reduction when an engine is accelerated. The front-to-rear shift in stage loading as the compressor speed increases was discussed for the first time.

The other three papers published around 1953 were by Sears [8], Emmons et al. [9], and Marble [10]. (It appears that Emmons' ideas were in circulation as early as 1951, and this may explain why the theoretical workings in all these papers are so similar.)

Emmons et al. [9] began by commenting on the confusion surrounding the behavior of compressors at low-flow rates: "Here, violent over-all instabilities identified with the generic term "surge" most often have been reported, typical symptoms being audible thumping and honking at inlet and exit. severe mechanical vibration, and oscillating pressures throughout the machine." In their theoretical work, Emmons et al. modeled a two-dimensional cascade of blades as a series of parallel channels, each with a variable exit area controlled by separation-induced blockage on the suction side of the blade. The blockage was linked to changes in incidence angle which were, in turn, related to variations in the incoming flow. The resulting linear theory was restricted to small perturbations and provided a criterion for flow stability, but could not predict cell speed or number. In their work on surge, Emmons et al. derived a formula for surging frequency and showed that instability will only occur when the slope of the "work input characteristic" is zero—the forerunner of the findings of many subsequent theories.

Besides the impetus which Emmons' work gave to decades of analytical modeling, his paper also provided the now famous sketch used to explain stall cell propagation. Emmons's idea is reproduced in Fig. 3. This sketch is still used today in every class where stall propagation is discussed.

After 1953, modeling continued with ever greater levels of sophistication. Stenning and Kriebel [11], Fabri and Siestrunck [12], and Takata and Nagano [13] all produced nonlinear finite disturbance models and Dunham [14] reiterated that the limit of stable operation occurs when the total-to-static pressure rise characteristic is at its peak, i.e., where the slope is zero.

Looked at from the original aims of predicting stall cell speeds and numbers, these small perturbation theories were not really successful. A perspective on the situation was given much later by

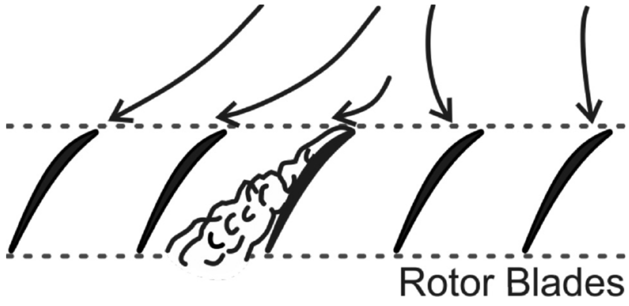

Fig. 3 Pictures explaining stall propagation based on early work by Emmons et al. [9]

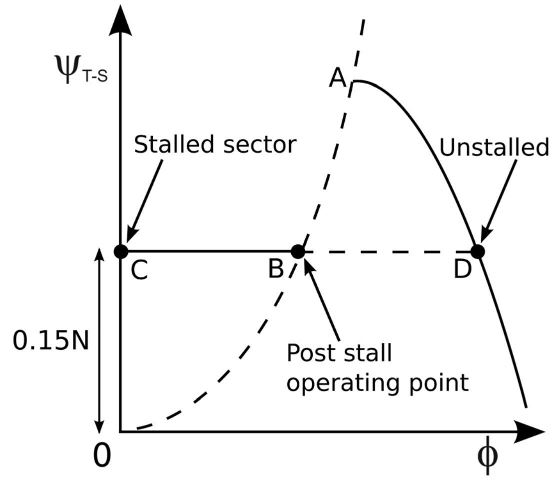

Fig. 4 Parallel compressor model as proposed by McKenzie [17]. Note the pressure rise in stall is 0.15 times the number of stages, and the portion of the annulus occupied by stalled flow is (D – B)/(D – C).

Stenning [15], who stated that "fully developed rotating stall is a highly developed nonlinear phenomenon and there is, in fact, no reason to expect that small perturbation theory will be successful in predicting cell speed after the cells have grown to their limiting size."

5 Install Performance and Stall Cell Details

Through the 1960s, the need to predict stall cell speed and number declined with the falling rate of stall-induced blade failures. Instead, a new difficulty arose know as the "hung stall problem." This was a problem experienced by pilots who were unlucky enough to have a stalled engine. The usual approach in this situation was to cut the fuel supply, enter a steep dive, and then hope the engine would restart under the ram effect. Experimental evidence suggested that the problem was linked to the size of the hysteresis loop and that this problem would get worse as compressor pressure ratios increased. A need thus arose for a better understanding of the performance of a compressor once it has stalled; what pressure rise could be expected, what type of stall cell would form, and how big would the hysteresis loop be? These were the motivating questions of the day.

5.1 Install Performance. The stalled performance of a compressor, especially when multiple stages are involved, needs a more robust treatment than is possible with single-blade row theories. The problem was thus approached by studying complete stages and even the performance of the compressor as a whole. Smith and Fletcher [16] were the first to adopt this approach. They viewed the annulus as being made up of two compressors working in parallel, one stalled and the other unstalled. (In this arrangement, both compressors are assumed to discharge to the same static pressure in the exit duct.)

Smith and Fletcher did not have access to accurate measurements and so came to the conclusion that all stall cells were of a fixed size, i.e., equal to half the annulus. This assumption meant that in order to change the overall flow rate through the compressor, as would happen if the exit throttle were adjusted, the flow through the stall cell would have to change. This idea was soon proved wrong by Fabri and Siestrunck [12] and Dunham [14] who both made the progressive step of allowing the stall cell size to change in line with the overall compressor flow rate.

McKenzie [17] later proposed a simple model in which there is no net flow in the stalled sector of the annulus and full flow through the unstalled sector, i.e., the stalled and unstalled sectors operate separately at points C and D in Fig. 4. (Note, in Fig. 4, the size of the stalled sector of the annulus is given by (D – B)/(D – C).) In McKenzie's model, the stalled branch of the characteristic (C to D) is assumed to be a horizontal line. This step ensures that the two sectors of the annulus will have the same static pressure in the exit duct. Furthermore, the position of this line on the pressure rise axis (total-to-static) is fixed at an empirically determined level of $0.15N$, where N is the number of stages. This level, which represents the performance of the compressor when stalled, was observed to be more-or-less independent of compressor design and only dependent on the number of stages.

McKenzie's ideas were later extended by Day et al. [18] who used experimental information to enhance the predictive capabilities of the model. The principal addition was the observation that part-span stall cells seldom occupy more than 30% of the annulus and full-span cells seldom occupy less than 30%. Thus, depending on the shape of the unstalled characteristic, these rules make it possible to estimate which type of stall cell is likely to form when the compressor first stalls. Additional rules also make it possible to estimate the extent of the stall/unstall hysteresis loop. (The details of the parallel compressor model given here are an abbreviation of those originally reported.)

The importance of stage design in determining the behavior of the compressor when it stalls is also explained by this model. The unstalled pressure rise characteristics of two different compressor designs are presented in Fig. 5. (For simplicity, both compressors are shown as stalling at the same throttle setting.) The characteristic on the left (Chic 1) represents a low-pressure rise compressor, while the one on the right is for a higher pressure rise machine with a higher design flow rate. Because of the difference in peak pressure rise between the two compressors, the size of the pressure drop which occurs at stall is very different; B to C for characteristic 1 and A to C for characteristic 2. The differences in the peak pressure levels of the two characteristics also has implications for the type of stall cell which will form when the compressor stalls—as shown in the lower part of Fig. 5.

5.2 Stall Cell Details. Before detailed measurements were possible, the structure of the flow inside a stall cell was portrayed in different ways. There are pictures of stall cells as groups of individually separated blades [19], as wakes downstream of bluff bodies [12], as parallel lines of vortices [11], and as pairs of contrarotating vortices [20]. We now know that all these models contain elements of truth, but none fit the flow patterns observed in more recent experiments.

A clearer picture of the flow inside a stall cell was provided by Day and Cumpsty [21] who used fast response instrumentation and a computerized logging system (new at the time) to obtain detailed measurements in a multistage compressor. Their findings can be summarized as follows:

(1) Velocity measurements confirm the basic assumptions of the parallel compressor model. The average axial velocity in the stalled sector of the annulus is near zero and in the

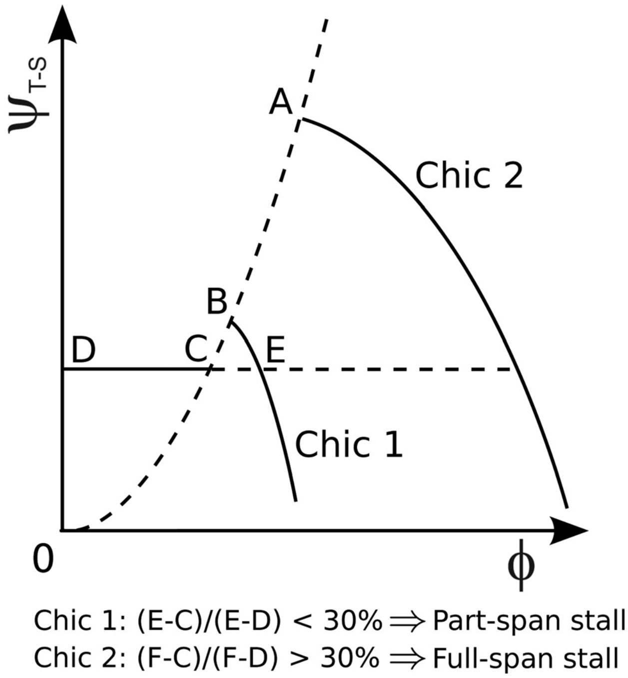

Fig. 5 Pressure rise characteristics of two compressors of different designs

unstalled sector is above the design value. The measurements also confirm that the model gives a good estimate of stall cell size and thus of stall cell type.

(2) As was sometimes observed in the early literature, the flow inside a large cell is very energetic. Upstream of a rotor row, the tangential velocity will exceed wheel speed.

(3) In a multistage compressor, the stall cell extends axially through the blade rows with flow being transported from one side of the cell to the other by the rotor blades—as shown in Fig. 6; taken from Ref. [22]. The old idea of a stall cell as a "dead wake" spiralling around the annulus is thus not appropriate.

(4) Strong centrifugal forces accompany the high tangential velocities in the stall cell and this makes the flow in the cell strongly three-dimensional. It also locally raises the pressure on the casing, making tip rubs more likely on the opposite side of the compressor.

Although these measurements tell us something about stall cell structure, we still cannot say anything about what controls stall cell speed. A recent study at the Whittle Laboratory looked at the effect of five variables on the stall cell speed in a single-stage compressor. The results are summarized in Fig. 7 where it can be seen that rotational speed, blade profile, and tip clearance have little effect on cell speed. Rotor stagger has some effect, but moving the inlet guide vane (IGV) forward by two chord lengths had the greatest effect of all. These measurements do not explain the physics governing cell speed, but they do show that some geometric features have more influence than others—and should not be neglected in any CFD studies.

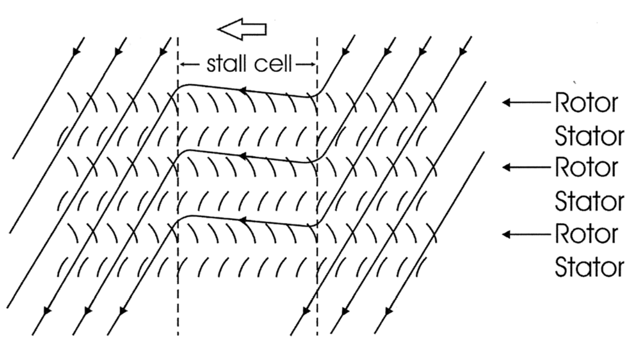

Fig. 6 Sketch of a stall cell extending radially through a multistage compressor; drawn in absolute frame. The trailing edge of the cell is on the right-hand side.

6 Surge

In elementary terms, stall is a disturbance of the flow in the tangential direction, while surge is a disturbance in the axial direction. During stalled operation, the average flow rate through the compressor is steady, but during surge, the flow rate will pulse—sometimes so violently that reversed flow is induced. Both these forms of disturbance occur at the same instability point on the performance map, but it is the lengths and volumes of the entire compression system which determine which form of instability will be dominant. As a rule of thumb, a compressor attached to a short exit duct (as in a laboratory situation) will stall, whereas a compressor attached to any form of storage volume (a combustion chamber for example) will surge.

In the 1940s and early 1950s, surge studies started out in the same way as stall studies with a variety of words used to describe a phenomenon which was not clearly understood. The confusion is illustrated in the work of Emmons et al. [9], who had to admit that "the phenomenon previously (in the same paper) called surge in the single-stage Harvard axial rig was not true surge, but consisted of a single-cell stall propagation." Another similarity with the early work on rotating stall was the frequent use of audible signals to infer what was happening in the compressor. Pearson and Bowmer [23], when studying surge, reported that "sometimes this phenomenon is gradual, taking the form of a sort of burbling, but more generally it takes the form of a sudden "bang" associated with a violent shake of the whole foundation of the bed on which the compressor is mounted."

There is another, more subtle, factor which hindered the early understanding of stall and surge. This is to do with the behavior of the compressor when it stalls. Many centrifugal compressors and low-pressure rise axial machines do not exhibit a discontinuous pressure rise characteristics at the stability limit. On the other hand, axial compressors of medium and high loading experience a large pressure drop when stall occurs—see the example in Fig. 8 where the characteristics for an axial and a centrifugal compressor are compared and where, for the axial machine, the operating

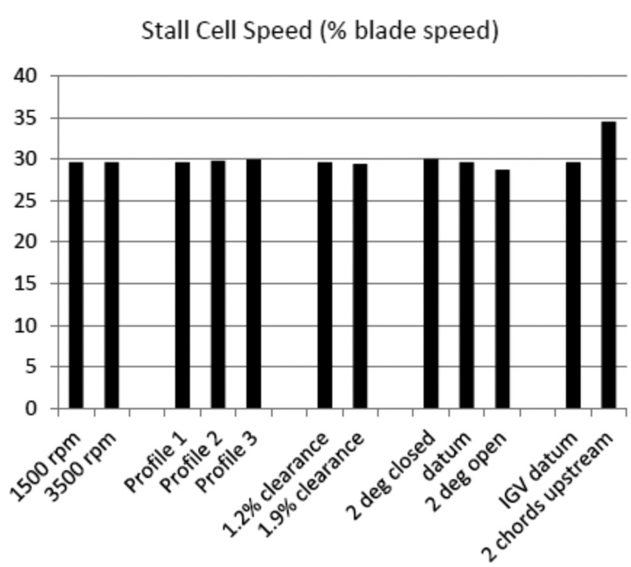

Fig. 7 The effect of various compressor details on stall cell speed

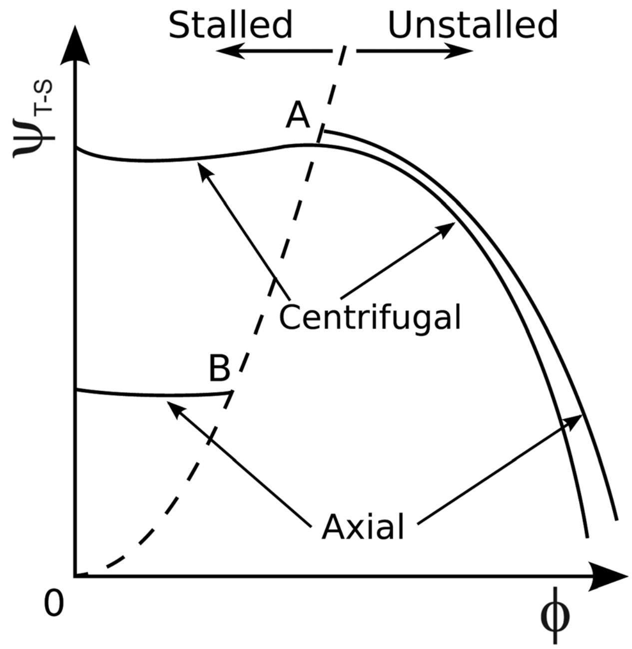

Fig. 8 Centrifugal and axial compressor characteristics illustrating the difference in pressure level after stall has occurred

pressure drops by 50% when stall occurs (A to B). In a situation like this it is not surprising that the severity of a surge event will be different in the two types of compressor. Emmons et al. [9] said that surge in their centrifugal compressor, which had an almost continuous pressure rise characteristic, was "so mild as to be undetectable without the aid of instrumentation." This difference in stalling behavior between continuous and discontinuous characteristics, i.e., between small and large pressure drops at the point of stall, has not always been appreciated.

In a centrifugal compressor having a characteristic like that in Fig. 8, the surging process is likely to produce a flow oscillation which is limited in amplitude and will be of simple harmonic form, i.e., the mass flow/pressure rise trajectory on the performance map will be circular. The axial compressor, with its dramatic drop in pressure rise at the point of stall, is likely to produce a performance trajectory which is more rectangular in shape and may be extensive enough to include reversed flow. Surge in this case represents a relaxation oscillation.

Early surge models come from Refs. [7,9,24]. All used a lumped parameter model to represent the components of the compression system, i.e., the compressor, the ducting, the plenum, and the exit throttle. The pressure-rise versus mass-flow trajectories derived by Moritz were very realistic, but it was Greitzer [25,26] who produced the definitive work on surge. He also used a lumped parameter model in which four equations were used to represent the dynamic behavior of the system: one equation to model the oscillation of the flow in the compressor duct (incompressible); one to keep track of the mass of air in the plenum (compressible); one to deal with the flow in the exit duct; and one to represent the pressure drop through the discharge throttle. A time delay was also introduced to simulate the unsteady response of the compressor. From the model, Greitzer developed the now famous B Parameter; a nondimensional grouping of system variables which indicates whether a particular compressor configuration will stall or surge.

The details of the B parameter are set out below along with a descriptive interpretation of the parameter as a ratio of pressure and inertial forces

$$B = \frac{U}{2a}\sqrt{\frac{V}{AL}} \quad \Rightarrow \quad \frac{\text{Pressure force}}{\text{Inertial force}}$$

The ratio shows that if pressure forces are dominant, the flow in the compressor duct will undergo velocity oscillations (surge), but if inertia forces are largest, the compressor will only experience rotating stall. Thus, what had been qualitatively known since the 1940s about the effects of lengths and volumes on surge behavior was now convincingly stated in a useful mathematical expression.

Greitzer's experimental work demonstrated that, depending on the compression system details, there will a critical value of B (B_{crit}) above which surge will occur and below which the compressor will stall. Since these ideas were first published, there have been many cases in which the critical value of B has been assumed to be a universal number, i.e., equal to 0.8 as it was for Greitzer's compressor. This belief comes from inattentive reading of the paper. The critical value of B is not a fixed number but depends on the design of the compressor. This point is illustrated in Fig. 9 from Ref. [27], where the pressure rise characteristics of the compressors used by Greitzer and Day are shown. The characteristics are very different and so are the values of B_{crit} for the two machines: 0.8 for the low-pressure rise machine and 0.4 for the other. The numerical value of B_{crit} is thus clearly compressor specific.

The reason why B_{crit} changes from installation to installation is because the only quantity representing compressor performance in the definition of B is "U," the blade speed; delivery pressure is not taken into account. This problem was discussed by Day [27] where an attempt was made to normalize the shape of the compressor characteristic to bring about a universal value of B_{crit}. The outcome of this exercise was partly successful, but some secondary performance parameters were still not included.

Some experimental work was done by Day [27], taking a closer look at the behavior of a compressor when it surges. Two additional points are relevant here:

(1) In all cases of surge in axial compressors, each surge cycle is initiated by a brief period of rotating stall. The sudden loss of pressure rise which occurs when stalling begins reduces the compressor's ability to support the pressure in the plenum and so the flow in the compressor duct is force to decelerate or even reverse. The point here is that, in an engine-type compressor, surge is not only preceded by stall but also caused by it.

(2) Helmholtz-type pressure fluctuations occur in all test facilities. In the compressor rig used by Day [27], the oscillations were weak and the amplitude remained fixed as the stability boundary was approached. It was found that the

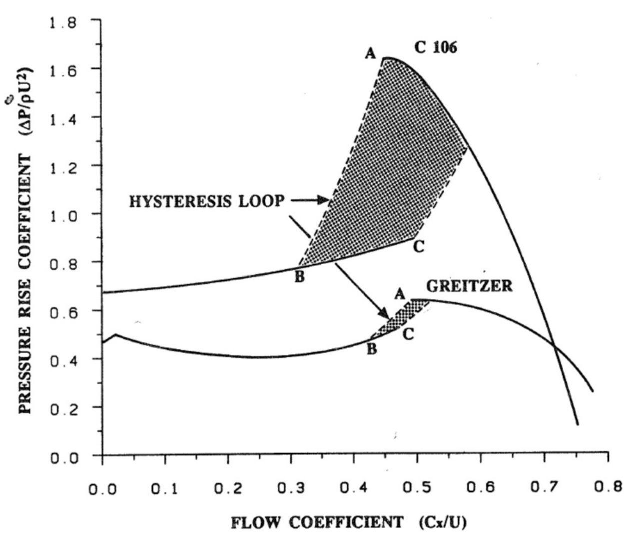

Fig. 9 Comparison of the performance of the C106 and the Greitzer compressors—from Ref. [27]. $B_{\text{crit}} = 0.4$ for the C106 compressor and 0.8 for the Greitzer compressor.

surge onset process was not related, in phase or frequency, to the Helmholtz oscillation. What this tells us is that the Helmholtz and surging oscillations are independent of each other.

The details of surge in high-speed compressors have not been considered here. This can only be done once the inherent differences between low-speed and high-speed machines have been discussed—as will be done in Sec. 9.

7 Analytical Studies

The 1980s was a period of unusual activity on the theoretical front. The decade began with some fairly simple models of compressor stability and ended with impressive analytical descriptions of stall, surge, and inlet distortion.

The first paper of note was by Cumpsty and Greitzer [28] and, although this is not strictly an analytical paper, it is regarded as the origin of ideas which proved useful in later works. The paper starts with the experimental observation that, in a multistage machine, a full-span stall cell will extend axially through the blade rows. To sustain this axial profile, flow must pass from the unstalled sector of the annulus into the stalled sector through the trailing edge of the stall cell, as shown in Fig. 6. The flow crossing the cell boundary experiences a sudden change in axial momentum and thus a pressure difference is generated in the blade passages—positive for rotors and negative for stators. By linking the rate of change of momentum at the cell boundary to the speed at which the cell rotates and by making the row by row pressure changes add up to the overall pressure rise across the compressor, Cumpsty and Greitzer obtained an expression for stall cell speed. Although several simplifying assumptions and empirical constants were necessary to arrive at this prediction, good agreement with experimental results was achieved. This paper highlights the importance of interblade inertial effects which will become a key feature of subsequent studies.

A few years later, a major work was published by Moore [29–31] which dealt with the linear and nonlinear requirements for the existence of circumferential and axial flow disturbances. The work was presented in three parts covering small perturbations, finite disturbances, and large limit cycle oscillations. Here, the idea of an underlying axisymmetric pressure rise characteristic was introduced for the first time. This idealized characteristic represents the performance of the compressor when unsteady effects are eliminated. In the modeling, the unsteady behavior of the compressor is based around the shape of this characteristic through the inclusion of inertial effects. Moore's work produced realistic predictions of stall cell speed and explained the factors influencing hysteresis. The analysis also sets out the conditions under which small amplitude flow disturbances (modes) can exist prior to stall.

After the introduction of the idea of an underlying axisymmetric characteristic, Koff and Greitzer [32] considered how the shape of this characteristic might be determined. Experiments in this direction are not easy and therefore the authors decided "to obtain an axisymmetric characteristic by inertially correcting experimental surge data." This approach led to an ambiguous result with two noncoincident characteristics. It was then shown that, by fitting a cubic curve between the peak of the unstalled characteristic and the lowest point of the reversed flow curve, useful analytical results could be obtained. This cubic fit is shown in Fig. 10 and has been used in most subsequent analyses.

Following on from Moore's analysis, Moore and Greitzer [33] and Greitzer and Moore [34] presented a unified view of stall and surge in multistage compressors. The analysis makes use of the axisymmetric characteristic of Koff and Greitzer [32] and allows for compression system transients by coupling a two-dimensional representation of the unsteady flow in the compressor with a lumped parameter model of the overall system. The governing equations include the Greitzer B parameter as a variable and captures states of pure rotating stall, pure surge, and combinations of both. The examples given to demonstrate the capabilities of the analysis agree very well with the experimental measurements.

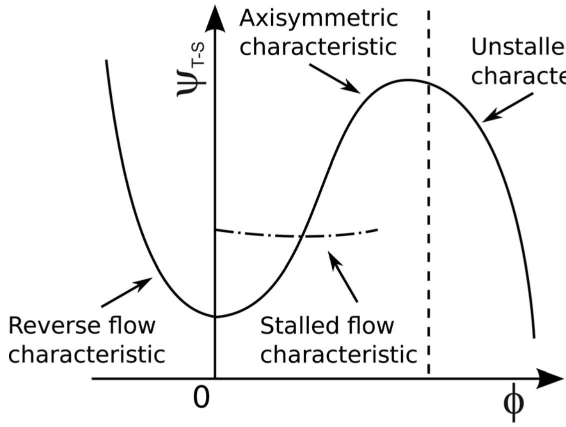

Fig. 10 Axisymmetric characteristic with cubic center section

Later, a paper by Hynes and Greitzer [35] focussed on the effects of inlet distortion on the stability of multistage compressors. The work builds on that of Ref. [29–31] using a similar axisymmetric characteristic, but with a further correction for the unsteady response of the compressor due to inlet distortion. Linear equations were set up for the inlet and outlet flow fields, with nonlinear treatments applied only to the compressor and the overall system. The main objective of the work was to assess the loss of stability margin brought about by various levels of inlet distortion.

The last paper in this series was by Longley and Hynes [36], which dealt with the stabilizing effects which stages can have on their neighbors. The paper provides experimental support for the idea of an underlying axisymmetric characteristic. A three-stage compressor was used in which the two downstream stages were restaggered so as to stall at a lower flow coefficient than the first stage. The result of this is that the stable operating range of the first stage is extended smoothly over the peak of the characteristic by the stabilizing effect of the downstream stages. The results of the experiment are shown in Fig. 11.

(From a historical point of view, it is interesting that the influence which stages can have on each other was first reported by Foley [5] in 1951. He observed that "subsequent stages helped to maintain a more uniform flow through the first stage, even when stalled.")

The analytical work of the 1980s, supported by experimental results, produced models which successfully dealt with clean and

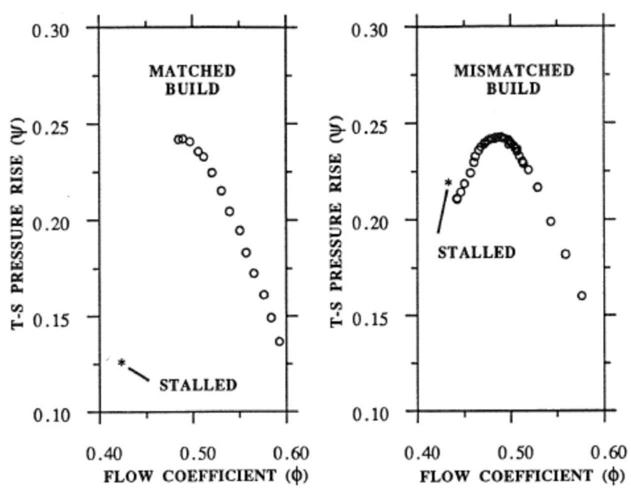

Fig. 11 First-stage characteristics showing extended range of stable operating when support is provided by mismatched rear stages (Longley and Hynes [36])

distorted flow and which were capable of mapping compressor behavior during stall and surge. In addition, the analytical work pointed to the possible existence of small amplitude circumferential perturbations (modes) which may exist in the compressor just before rotating stall begins.

8 Active Control and Stall Inception

After all the theoretical work of the 1980s, a new and infinitely more practical field of study opened up with the introduction of the idea that stall and surge might be amenable to active control. This change in direction came about almost overnight with the publication of a paper by Epstein et al. [37]. From this one paper, a whole new industry sprang up bringing together experimental and theoretical work and drawing control engineering into the sphere of aerodynamics.

The background to this sudden upsurge of activity lies in the theoretical works of Moore, Greitzer, and Hynes in which velocity perturbations were predicted to precede the onset of rotating stall. Being initially of low amplitude, these perturbations might be amenable to active control, in which case their development could be suppressed. This would give rise to improvements in performance and operating range.

The paper by Epstein et al. [37] strikes an enthusiastic note and reads like an advertisement for a new technique which will solve all our stability problems. Figure 12 gives a good impression of the upbeat nature of the paper. In this figure, active control moves the operating point up the characteristic to a higher pressure level and at the same time moves the stall point left to a more favorable position. In the long run, the idea of extending the operating range of a compressor by applying active control proved possible, but the optimistic increase in delivery pressure shown in Fig. 12 never materialized.

Although the idea of active control was predicated on the existence of low amplitude, long length-scale circumferential disturbances (modes), these disturbances had, as yet, not been observed in practice. Then, in 1988, a method of proving their existence was suggested by Longley and Greitzer [38] who proposed using six hot-wires equally spaced around the annulus to check for prestall velocity perturbations. The method was implemented by McDougall et al. [39], whose experiments clearly demonstrated the existence of modes. (An example of this type of wavelike prestall activity is shown in Fig. 13.) These experiments were important because, for the first time in the history of stall and surge, a purely analytical prediction was proved correct in practice.

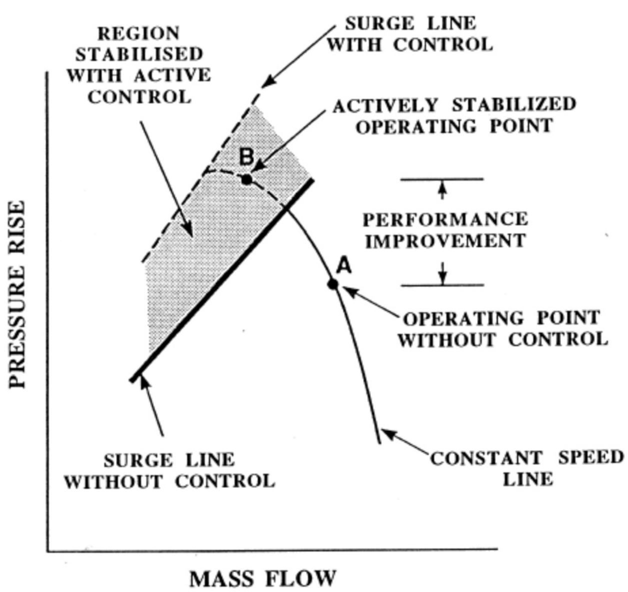

Fig. 12 Schematic illustration of the effects of active control on compressor performance (Epstein et al. [37])

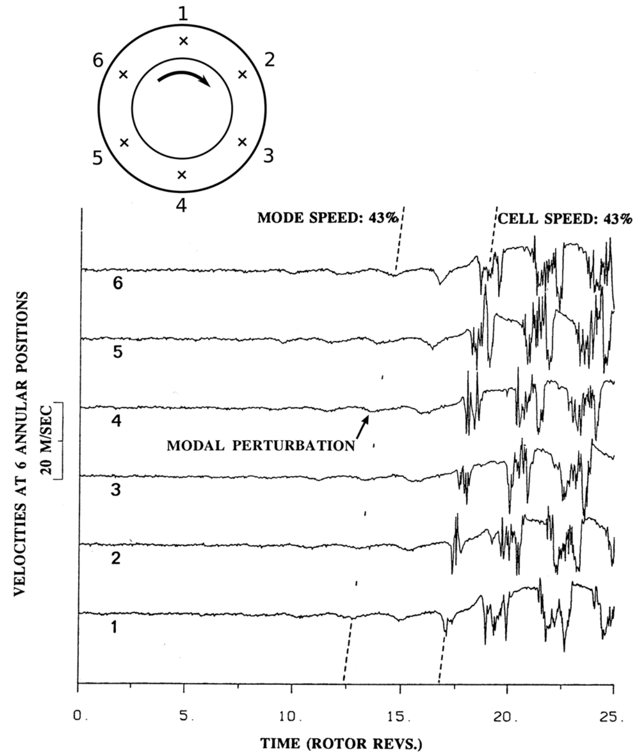

Fig. 13 Hot-wire measurements around circumference of compressor showing modal activity before stall. (Model oscillations can be seen between revolutions 7 and 17 [44].)

With the existence of modes firmly established, and with their amplitude growing exponentially with time in the early stages of development, the way was clear for experimental work on active control. Some early experiments were carried out on centrifugal compressors by Pinsley et al. [40] and Gysling et al. [41], but the most important work of the time was done by Paduano et al. [42] at MIT on a single-stage axial compressor. This group, using fast response, variable stagger, inlet guide vanes (12 in all, each individually controlled), achieved a useful increase in stall margin by controlling the first, second, and third circumferential modes.

In UK, a parallel active control project was in progress also using a low-speed axial compressor. In this case, Day [43] used a circumferential array of fast-acting air jets to achieve control of the first circumferential mode. The control system was not as sophisticated as that employed by Paduano et al., but still a reasonable stall margin improvement was achieved. At this time, and just when active control seemed to be on a sound footing, experiments in Cambridge pointed to an alternative stall inception mechanism which did not involve modal oscillations.

Day [44] observed small disturbances in the rotor tip region, which were not of the long length-scale modal type. These disturbances appeared without warning, were small in circumferential extent, localized to the blade tip region, and developed directly into stall cells. These disturbances, now known as "spikes," were found to rotate at about 70% of rotor speed—which is much higher than for a modal disturbance which usually rotates at less than 50% speed. An example of this type of spike stall inception is given in Fig. 14.

In 1991, we thus had two types of stall inception mechanism: one of long length-scale and progressive growth (modes), and the other of short extent and abrupt formation (spikes). The discovery of spikes was not welcomed at the time because it was not clear how these new disturbances would fit into the active control program. As it turned out, spikes were demonstrated to be susceptible to corrective control and this could be achieved using the same hardware as was used to control modes.

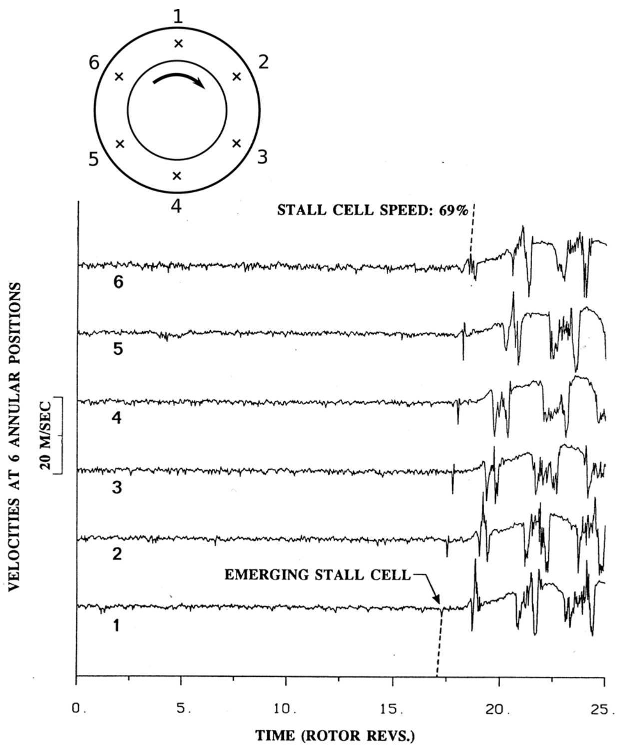

Fig. 14 Hot-wire measurements around circumference of compressor showing spike-type stall inception. A small disturbance appears without warning, at revolution 17 [44].

(A brief historical note is necessary at this point, because the spike-type disturbances reported by Day [44] were actually first observed by Jackson [45]. The spikes in his case were not clearly linked with cell formation and their importance was overlooked.)

At about this time, Camp and Day [46] took a further look at stall inception. They were interested in why some compressors stall at the peak of the total-to-static pressure rise characteristic (as they should do according to linear theory) while others stall before the peak is reached. It was found that modes only appear when the slope of the characteristic is zero, while spikes often occur before this point is reached. It was found that modes will only develop if localized overloading does not cause spikes to appear first. Modes are thus a feature of the flow in the whole compressor, whereas spikes are a localized phenomena starting in an individual blade row.

Work on active control continued for many years in various centers around the world. Control algorithms were improved [47], passive control using aeromechanical feedback was demonstrated [48] and a small company was started in the U.S. to investigate the market potential for active control. At NASA, active control was applied to stage 35 (transonic) using pulsed air jets [49] and in England, Ffowcs Williams and Graham [50] implemented active control in an automotive gas turbine. Rolls-Royce also backed a project in which a small jet-engine (Viper) was modified to demonstrate active control [51].

In spite of reported successes, work on active control died out in the UK after the Viper engine experiments. Freeman [52] made the observation that "When the engine is new, you don't need control, but when the engine gets old, you then expect all this equipment you have been carrying around for years to suddenly start working perfectly." Work in the U.S. continued for a few years and then also drew to a close.

9 Low Speed Versus High Speed

Most of the analytical work described earlier was restricted to 2D incompressible flow and most of the experimental work was done in low Mach number (incompressible) machines. In this section, attention will be focussed on turbomachines where compressible effects are important and where stage matching makes stall prediction a moving target.

Day and Freeman [53] conducted a study of stall inception in an eight-stage Rolls-Royce Viper engine. In this compressor, part-span stall was observed at low speeds, full-span stall at medium speeds, and surge at top speed. In the midspeed range, the stall inception pattern was similar to that observed in low-speed studies with spike-type disturbances leading directly to full-span stall. In the upper midspeed range, the surging process was also found to be similar to that observed in low-speed testing. At full speed, however, something different happened—rotating stall still preceded surge, but reversed flow took hold so quickly that the stall inception details were hard to unravel. No sign of modal activity was observed during this series of tests.

Later, as part of a large European funded project, four multistage high-speed compressors were instrumented and tested. The objective was to determine the generic features of stall in high-speed machines and to examine the diversity of stalling processes with which an active control system might be expected to cope. The work was reported by Day et al. [54] showing that three of the four compressors exhibited spike-type stall at low speeds, modes in the midspeed range and spikes or abrupt stalling at full speed. A list of stalling disturbances observed in the compressors included: front end start-up stall, multicell part-span stall (with frequencies over ten times shaft rotation), full-span stall, spikes, modes, and abrupt surging at full speed. The main point in the paper was that an active control system in a high-speed multistage machine would have to be very sophisticated, and effective throughout the length of the compressor, if it were to cope with all possible stall inception patterns.

The eight-stage Viper compressor was included in the list of high-speed compressors mentioned above. Modal activity was not detected when this compressor was first tested by Day and Freeman [53], but in the later European tests, modes were observed. The reason for this is because, in the Viper compressor, modal activity only occurred in a narrow speed range between 85% and 87%. The modes were thus overlooked in the first series of tests. For modes to exist, all the stages need to be well matched so that none succumb to spike stall before modes have had a chance to develop. In a compressor of many stages, it is thus easy to see that the range of speeds over which all the stages are well matched could be very narrow. (In one of the shorter compressors (three stages), the range over which modes were observed was much wider from 61% to 80% speed.)

At about the same time as the European project, a paper on stall in high-speed compressors was published by Tryfonidis et al. [55] from MIT. This work also dealt with the behavior of high-speed compressors, nine in all, but here the focus of attention was on the flow before rotating stall begins. In a sense, this was a stall warning paper which highlighted the existence of modal activity hundreds of revolutions before stall. The point of interest here, however, is that, in all nine compressors, surge was reported as being preceded by rotating stall and that "rotating stall, and not surge, is the performance-limiting instability in axial compressors."

In the mid 1990s, a lot of work was done on compressor stability in high-speed machines. In an extension to the Moore and Greitzer analysis, Bonnaure [56], and later Hendricks et al. [57], introduced the effects of compressibility into the Moore and Greitzer analysis and found a new type of modal disturbance having circumferential and axial structure. The analysis suggested that the conventional Moore and Greitzer modes would predominate in compressors of medium speed, but the new compressible modes would be present at transonic speeds and above. (Tryfonidis et al. [55] also observed modes of the compressible type in some of the high-speed compressors they tested). In further work, Weigl et al. [49] used a control strategy based on the concept of compressible modes to stabilize the NASA 35 transonic compressor. Later still, Spakovzsky et al. [58] used compressible stability theory to estimate the stability margin of deteriorated engines. Engineering success was achieved here in that the available surge margin could be measured without actually surging the machine.

The abrupt nature of surge onset in high-speed compressors, as mentioned in connection with the Viper tests, was considered years earlier by Mazzawy [59]. He studied the structural loads experienced by compressor blades due to reversed flow during surge. In this paper, a surge event is described as, "a stoppage and reversal of flow accomplished through an internally generated shock wave of high strength, the shock wave being initiated by some form of flow separation (stall) somewhere towards the rear of the compressor." It was also stated that the asymmetric start of the shock process gives rise to an uneven distribution of the reversing flow when it reaches the front of the compressor. This can cause high bearing loads and casing rubs. For many years now, Mazzawy's ideas have been used by mechanical designers to determine the inter-row spacing necessary to avoid blade clashing during surge.

In this section, new flow features have been introduced, which are not observed in low Mach number machines. More work is necessary, and although high-speed testing is expensive, this is where our need is greatest.

10 Stall Warning

A system which can warn of impending stall has been sought for many years. The idea of such a system attracts interest because a solution is always temptingly close and the rewards will be high.

Engine control systems are designed to keep the operating point of the compressor well away from the surge line, especially during transient manoeuvres. Modern full authority digital electronic control systems execute this task with ease, but the old hydromechanical systems often got confused. When the compressor stalled, the old control systems would mistakenly correct the loss of pressure rise by injecting more fuel—with disastrous consequences for the turbine.

Ludwig and Nenni [60] were among the first to design a fast-response control system which would alter vane settings or open bleed valves when stall occurred. In their system, pressure signals were measured at various axial and circumferential locations along the length of the compressor. These signals were collectively used to sense any unusual activity in the compressor and then to activate the control system. Their ideas were tested on a J-85 engine and shown to be very effective in the midspeed range. At the time, critics of the system suggested that the speed of response of the equipment would be insufficient for high-speed operation where hysteresis would make recovery more difficult.

A decade or so later, a stall warning study was carried out by Garnier et al. [61] using experimental data. Measurements from three compressors were analyzed using the Moore and Greitzer stability model as a foundation. Prestall modal disturbances were identified hundreds of revolutions before stall and the growth rate of these disturbances was shown to agree with analytical predictions. These findings meant that the early detection of modal activity could be used for stall warning or for stability margin assessment during engine pass-off tests. In addition, it was shown that the modal disturbances grew into fully developed stall cells, "without apparent sharp changes in either phase or amplitude." From this, it was concluded that modal waves and rotating stall are two stages of the same phenomenon. (This idea is still debated today because some see modes as promoting stall inception rather than being part of it.)

Later, when short length-scale disturbances were routinely observed, Bright et al. [62] used data from high-speed compressors to produce a spike detection system. A statistical scheme was used and spike movement around the annulus in the NASA 37 stage was successfully monitored. The technique was also used to demonstrate stall warning based on measurements from a single sensor. Stall warning times of about 20 rotor revolutions were reported.

The stall warning work mentioned thus far has involved the detection of disturbances which are directly involved in the stall inception process. An alternative approach is to monitor the changing behavior of individual blade-passing signals as the compressor is driven toward stall. Gallus and Hoenen [63] used hot-wire measurements to show that the repetitiveness of a blade-passing signal will deteriorate as the compressor approaches the stability limit. Inoue et al. [64] made the same observation using over-tip pressure measurements saying that "a rotating stall precursor can be detected by observing the collapse of the periodicity in the pressure fluctuation." Based on this idea, a patent application was submitted by Inoue for a stall warning device. The same was done a few years later by Hoenen and Gallus [65] who studied changes in blade-passing signature and related these to the proximity of the surge line. Their system was successfully tested on an LM5000 gas turbine.

A group at the Georgia Tech also made a study of blade-passing signals in relation to stall proximity. Dhingra et al. [66] monitored the "nonrepeatability" of blade pressure signals and proposed a "correlation measure" to keep track of the changes. This correlation measure was shown to drop as the stall limit was approached and so could be used for stall warning. It was also proposed that the method could be used to monitor the degradation of tip clearance as the compressor ages. Christensen et al. [67] adapted Dhingra's correlation measure and developed a real time engine control system. Successful tests were carried out on an aircraft engine and in 2010 a patent application was submitted by GE.

Young et al. [68] also examined the changes in blade-passing signature as the point of stall onset was approached. In this work, attention was not focussed on the level of irregularity, but on the geometrical features of the compressor which control that irregularity. Tip clearance and eccentricity were specifically considered. It was found that when clearances were tight, and the shaft and casing were concentric, the rise in blade-passing irregularity just before stall was insignificant. With larger tip clearances (3% chord), however, the ramp-up in irregularity began earlier and increased faster. The situation became even more complicated when the casing was positioned eccentrically, relative to the rotor. In this case, the rise in irregularity only occurred in that part of the annulus where the tip clearance was largest. (These trends are illustrated in Fig. 15.) It was therefore concluded that a stall warning system based on blade-passing irregularity is unlikely to be of any use in an aero-engine compressor where concentricity and tip clearance change continually during the flight cycle.

11 Rotating Instabilities

This section covers the contentious and ill-defined topic of "rotating instabilities." Even the name has been the subject of lively discussion.

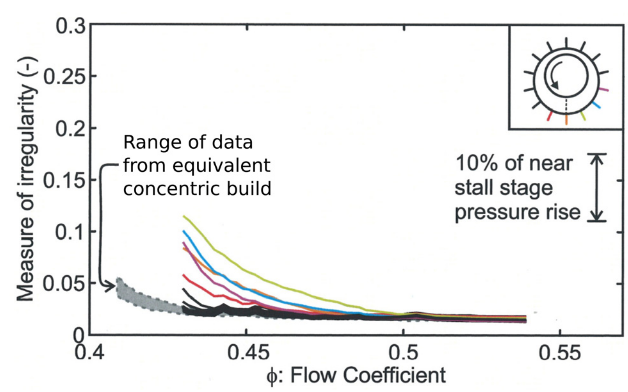

Fig. 15 Plot showing rise in blade-passing irregularity as flow coefficient is reduced—for eccentric compressor (Young et al. [68])

A collection of descriptive phrases taken from a number of papers will help to introduce the subject. Rotating instabilities are prestall disturbances, unrelated to modes or spikes, which occur near the tips of rotor blades with large clearances. These perturbations are small in circumferential extent (one or two blade pitches wide), rotate at about half rotor speed, in the same direction, have a high circumferential count (up to half the number of blades), and are always changing in intensity, wave number, and frequency. A typical example of the frequency spectrum obtained from a compressor operating with rotating instabilities is shown in Fig. 16. The hump at about one-third of the blade-passing frequency is typical of this type of disturbance, and the breadth of the hump is indicative of its time varying nature. (The words rotating instabilities are meant to describe disturbances which rotate, but are unsteady in their behavior.)

Rotating instabilities were first noticed by Mathioudakis and Breugelmans [69] who, when studying multicell stall, observed "the simultaneous existence of disturbances of different wavelengths." Later, Inoue et al. [64] noted that near stall, when tip clearances were large (greater than 3% chord), the flow at the rotor leading edge became very unsteady. From their pressure measurements, they suggested that this behavior was caused by "propagation of somewhat coherent disturbance at a different speed from rotor revolution." From this a new type of disturbance was proposed, which is neither mode nor spike and which appears long before the stability limit is reached. Kameier and Neise [70] also found that, when clearances were large, vortex separation in the tip region was unsteady and lead to a broad band hump in the frequency spectrum.

Inoue et al. [71] then published a paper dealing with the physical structure of small stall cells. This paper does not really fit into this section on rotating instabilities, but it does contain new ideas about prestall disturbances. Experiments were conducted on a low-pressure rise compressor in which small stall cells were observed to the left of the peak of the characteristic. It was argued that these cells were similar to the short wavelength disturbances observed during spike stall inception and thus it was useful to study these cells which are less ephemeral than those involved in the inception process. Casing pressure measurements revealed intense areas of low pressure just upstream of the rotor blades. These gave rise to the idea of a disturbance which looks like a trumpet-shaped vortex with the open end attached to the casing and the other end attached to the suction side of the blade—as shown in Fig. 17. Propagation of this "mini-cell" occurs when the casing end of the vortex moves toward the pressure side of the neighboring blade and there triggers a fresh vortex. (Note these disturbances are of subpassage dimensions.)

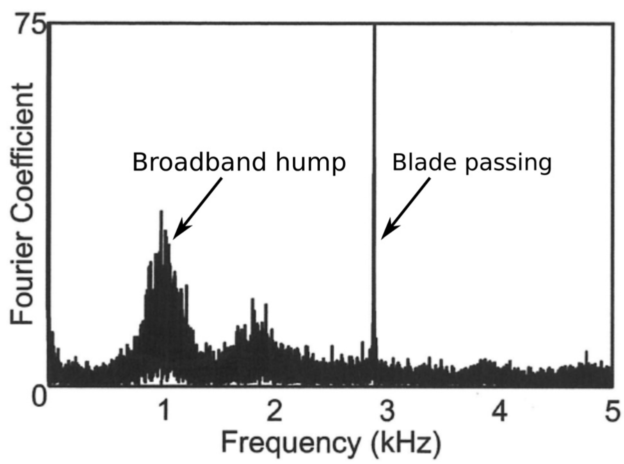

Fig. 16 Frequency spectrum from a compressor operating in a condition where rotating instability is present [68]

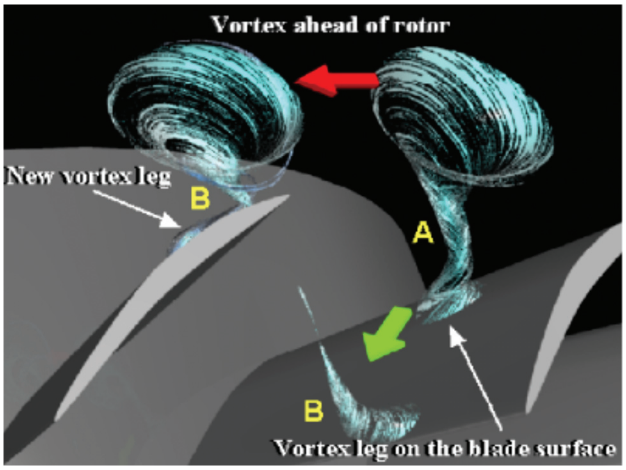

Fig. 17 Illustration of tornado-type stall propagation. (From Ref. [71].)

Mailach et al. [72] published a paper which, for the first time, used the words Rotating Instabilities in the title. Experiments were conducted in the large four-stage Dresden compressor where the tip clearance of the third stage was increased to 3.00% chord. In this stage, fluctuations of the over-tip clearance vortices were observed well before stall. No pressure measurements were available upstream of the rotor row and so the low-pressure spots reported by Inoue, if present, were not detected. In postpublication discussion, it was suggested that the disturbances observed were actually due to part-span stall. This idea was rejected on the grounds that part-span stall would give rise to a spike in the frequency spectrum, not a hump.

In work of a similar nature, März et al. [73] studied the unsteady behavior of near-stall flow in a single-stage compressor. Unsteadiness attributed to rotating instabilities was observed when the rotor clearance was raised beyond 2.8%. Numerical studies accompanying the experiments revealed a low-pressure "spot" moving from the suction side of one blade to the pressure side of the next. (This behavior is similar to that described by Inoue, but in this case the low-pressure region was within the blade passage and not ahead of it.) The calculations suggested that the low-pressure spot was the footprint of a radial vortex. The authors suggested that the erratic movement of this vortex from one passage to the next is the mechanism underpinning rotating instabilities.

More recently, interblade vortex structures have also been observed by Young et al. [68], again at large clearances and at operating conditions well before stall onset. In this case, intense patches of low pressure were recorded in the forward part of some blade passages. The sporadic formation of these regions is in line with earlier observations of intermittent behavior. A new finding from this work was that when the compressor was operated in an eccentric configuration, the low-pressure regions (blue in Fig. 18) did not propagate all the way round the annulus, but only existed in the sector of the annulus where the tip clearance was largest.

With the repeated mention of radial vortices, it is historically interesting to note that Stenning and Kriebel [11] detected interblade vortices almost 60 years ago. The picture in Fig. 19 has gone virtually unnoticed all these years.

The number of publications on the subject of rotating instabilities has increased recently, but the focus of attention appears to be drifting away from turbomachinery toward the realms of acoustics. Although there is still a lot of controversy surrounding this subject, there is enough experimental evidence from tests with large clearances to suggest that a new form of prestall instability may well exist. One thing is clear, this subject is unlikely to go away if the trend toward smaller engine cores and larger tip clearances continues.

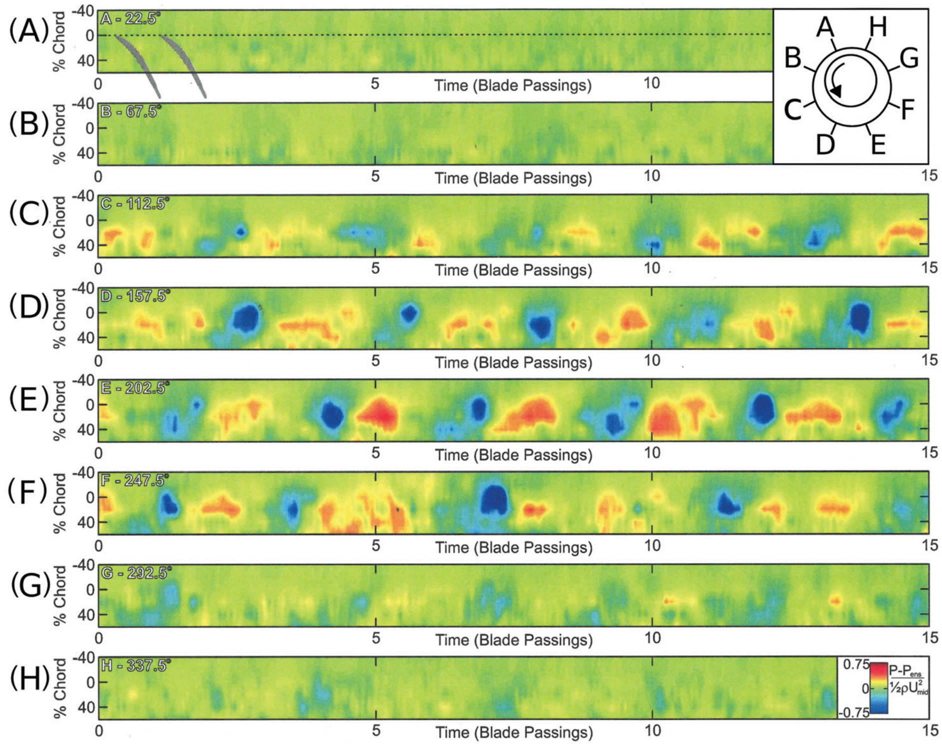

Fig. 18 Plot showing areas of low-static pressure (dark spots) in an eccentric compressor. Disturbances appear in large clearance region only. From Ref. [68]. (NB the above datasets were not recorded simultaneously, so disturbance tracking is not possible in this figure.)

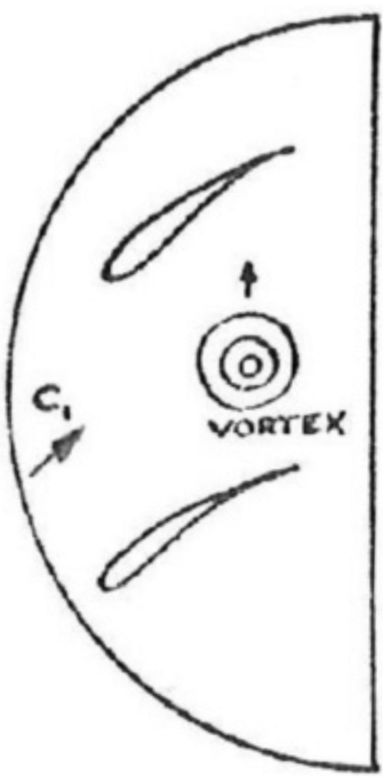

Fig. 19 Sketch from Ref. [11] showing an interpassage radial vortex during stall propagation

12 Casing Treatment and Flow Recirculation

Casing treatment is a means of extending the surge margin of a compressor by flow recirculation—either locally over the tips of the blades or in larger loops spanning a stage or even the whole compressor. The topic has received steady interest since the 1970s; driven mostly by the hope of finding an engine applicable system.

The earliest reference to a recirculatory stall delaying system comes from a patent by Wilde [74], in which it was claimed that stability could be improved if air was bled from the rear of a multistage compressor and reinjected at the front. Later in the 1970s, NASA and GE were involved in an extensive study of localized (over-tip) casing treatments. This work started when it was fortuitously discovered that holes in the compressor casing, used for other purposes, were actually beneficial to the stability of the compressor. Since this time, casings treatment designs have settled into two distinct groups, axial slots and circumferential grooves.

Most casing treatment designs produce useful stall margin improvements, but nearly all lead to a loss of efficiency. This loss is particularly troublesome at the compressor design point where air will be circulating through the casing treatment even though, at the time, it is not needed for stall margin improvement. Fujita and Takata [75] produced a summary of test results in which they showed that, with casing treatment, the efficiency of a compressor (measured at the design point) decreases as the effectiveness of the treatment increases. In other words, the bigger the gain in stall margin, the bigger the loss of efficiency. This is not to say that efficiency improvements when using casing treatment are never reported, but where they are, it is usually found that the design was inferior in the first place. In general, efficiency measurements are seldom quoted in the literature, with more than half the papers ignoring the subject altogether.

After the initial work at NASA, which was summarized by Hathaway [76], attention shifted from trying to perfect casing treatment to trying to understand how it works. Takata and

Tsukuda [77] conducted experiments showing that axial skewed slots, positioned squarely over the rotor tips, increased through-flow momentum near the casing. Smith and Cumpsty [78] also studied over-tip slots and concluded that the removal of high loss fluid from the casing was the primary reason for the effectiveness of the treatment. Crook et al. [79] undertook a numerical study of skewed slots and showed that the slots reduced flow blockage at rotor exit. All these explanations were overtaken, however, when Waterman [80], and then Seitz [81], moved the slots forward. Seitz showed that when axial skewed slots were moved upstream so that they only covered the first 25% of the blade chord, the efficiency penalty was reduced and the amount of flow recirculated through the slots was reduced tenfold. (Engine tests of this new configuration were undertaken, but were curtailed because of the difficulty of keeping the perforated abradable lining attached to the casing.)

In the literature, two papers (published 30 years apart) stand out because they seek to answer the same question, i.e., Will casing treatment be effective for all compressor designs? Greitzer et al. [82] compared the effectiveness of casing treatment when applied to a "wall-stalling" compressor (separation coming from the tip leakage flow) and a "blade-stalling" compressor (separation beginning on the aerofoil). It was found that the casing treatment worked well when flow separation occurred on the casing, but did not work at all when the separation occurred on the blades. Thirty years later, after modes and spikes had been discovered, Houghton and Day [83] applied casing treatment to compressors of each stalling type; spikes representing stall near the casing and modes representing flow separation on the blades. In this case, the casing treatment worked best when applied to the "spike" compressor (casing-stall) and least effectively when applied to the "modal" compressor (blade-stall). This is the same result as was obtained 30 years earlier, even though the method of implementation was different. Greitzer reduced the solidity of his rotor to force a change from wall-stall to blade-stall, Houghton and Day manipulated the IGV and rotor stagger settings to achieve a change from spikes to modes.

As mentioned previously, the idea of improving stall margin by recirculating air from downstream to upstream has been around since the patent of Wilde in 1951. In recent years, we have seen examples of over-stage recirculation (Strazisar et al. [84]) and over-tip recirculation (Guinet et al. [85]). The over-tip recirculation system uses a limited number of recirculation loops (less than the number of blades) with the loops tailored to extract flow from one location and reinject it at another. An example is given in Fig. 20. (Note that the reduced number of recirculation loops means that less of the compressor circumference is perforated by holes.) With this arrangement, there is again a problem with recirculation taking place all the time regardless of whether or not it is needed. A way around this was demonstrated by Weichert et al. [86], where the air extraction point was located above the rotor tips at such a position that recirculation only began when the casing pressure field shifted forward during periods of high loading.

In spite of the drawbacks mentioned, casing treatment provides a welcome service in situations where additional stall margin is required—for example, in some military engines where flight manoeuvres place extra demands on compressor stability and occasionally in civil compressors during preliminary development.

13 CFD

Computers were initially used to provide numerical solutions of theoretical equations. In the 1970s, however, Euler codes began to appear and, later on, when viscous terms were introduced, all aspects of turbomachinery benefited from improved modeling. In terms of early stall studies, He [87] simulated the growth of both short and long length-scales disturbances—using a mesh of just 47,000 nodes! At about the same time, Hendricks et al. [88] produced a nonlinear compressible model (part CFD) which simulated modal activity, stall inception, and rotating stall and surge. Their calculations were in agreement with measurements from a high-speed compressor and clearly showed rotating stall preceding surge.

Further CFD work was carried out by Saxer-Felici et al. [89] using unsteady solutions of the Euler equations producing detailed contours of the flow in a full-span stall cell. Also included were realistic simulations of cell size and speed of rotation. Recently, Choi et al. [90] have simulated the stalling and unstalling processes associated with hysteresis in a low hub–tip ratio fan. Their depiction of the stalling behavior of the fan agrees with common experimental experience. Most recently, Dodds and Vahdati [91,92] have used CFD to interpret experimental measurements from an engine compressor. They were able to show that the compressor was operating with two multilobed disturbances rotating independently in the first and second stages of the machine.

If we focus on stall inception, an interesting story unfolds. In a partial casing treatment experiment carried out by Cumpsty [93], spillage of the tip clearance flow forward of the rotor face was found to be responsible for a build up of blockage prior to stall. This idea received support from Hoying et al. [94], who presented a CFD simulation showing that stall begins when the tip clearance vortex moves forward of the leading edge of the neighboring blade. This forward movement of the leakage vortex was said to be the origin of a spike-type stall cell.

Inoue et al. [71] then published a paper drawing attention to the importance of radial vorticity in stall cell formation. They suggested that the low-pressure region seen by Hoying et al. ahead of their rotor was not the tip clearance vortex moving upstream, but a region of low pressure associated with a radial vortex anchored on the casing. Vo et al. [95] subsequently produced a fuller picture of spike inception based on multipassage CFD. In this case, it was suggested that two specific conditions, both linked to the tip clearance flow, were necessary for the formation of a spike-type disturbance. The first condition stipulated that the interface between the tip clearance flow and the oncoming flow should be parallel to the leading edge plane. The second condition required the simultaneous occurrence of back-flow from a neighboring passage—as shown in Fig. 21; taken from Vo [96]. This process of forward and rearward spillage spans two blade passages and is thus roughly the same size as a spike when it is first detected.

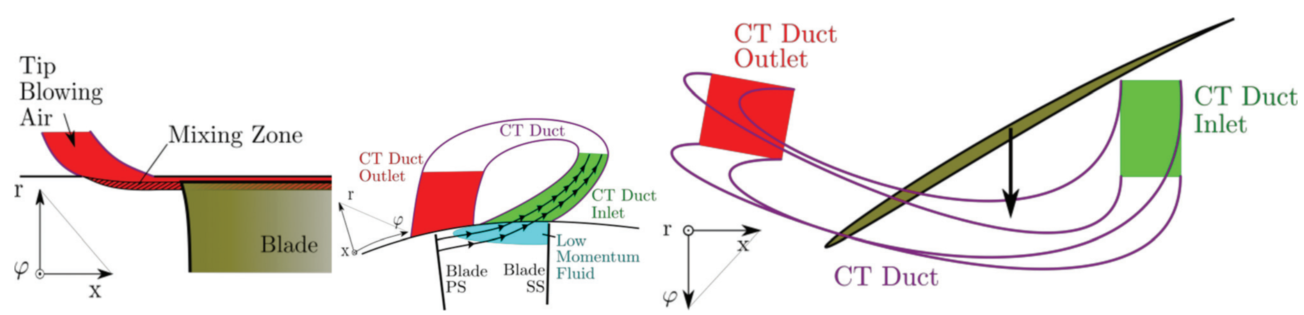

Fig. 20 Schematic of over-tip recirculation loop (from Ref. [85])

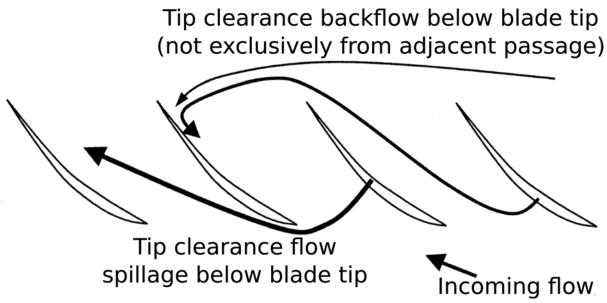

Fig. 21 Sketch of the two conditions required for spike-type stall inception; forward and rearward spillage

The next part of the story comes from the CFD work of Pullan et al. [97], who simulated spike-type stall inception in a tip critical rotor. The calculations suggest that flow separation begins near the blade tip due to high local incidence. The separation gives rise to shed vorticity from the leading edge and this develops into a trumpet-shaped vortex stretching between the casing and the suction side of the blade, as sketched in Fig. 22. Propagation ensues when the casing end of the vortex moves toward the pressure side of the neighboring blade and there triggers a new vortex. This stalling mechanism mirrors the tornado-type vortices proposed by Inoue and explains the inter blade vortices seen by Stenning and Kriebel in 1957. More interestingly, however, the work of Pullan et al. suggests that tip clearance flow is not a necessary requirement for spike-type stall. Their CFD simulations show that the formation of the radial vortex is caused by excessive incidence and this can take place without any tip clearance at all!

At present, we do not have a description of stall cell formation which is acceptable to all. Pullan et al. point to the importance of radial vorticity. Yamada et al. [98] reached the same conclusion, but added that forward and rearward spillage are real effects occurring after the formation of a radial vortex. More recently, Ju and Ning [99] have simulated stall in a high-speed compressor and find forward spillage in the tip region, but no sign of radial vorticity. With this background, the question must be asked if we are right to expect a unified model of stall inception, or are there an infinite number of routes to stall?

14 The Past and the Future

In the early days of the jet engine, it soon became clear that stall and surge were going to make compressor development difficult. Spurred on by the need to overcome these problems, research progress was impressive. It was soon possible to identify stall cells as the cause blade failures, to understand the differences between stall and surge, and to divide stall into part-span and full-span—the former being less destructive than the latter. Then, by studying how a compressor behaves when it stalls, it has been possible to delay stall inception using active control, improve operating range using casing treatment, and to develop quick acting surge arresting techniques. On a more erudite level, we have produced analytical models of stall and surge, discovered two types of stall inception mechanisms, explained stall/unstall hysteresis, and improved our understanding of stage matching. These achievements are impressive.

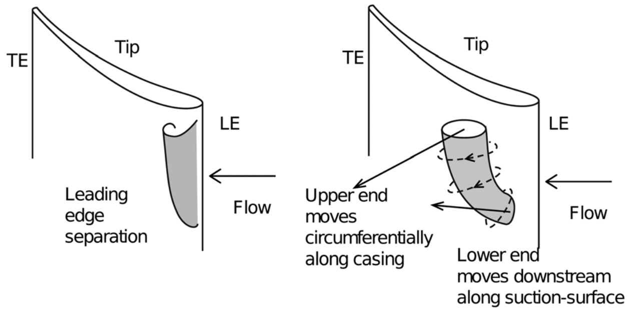

Fig. 22 Schematic showing how leading edge separation becomes a vortical disturbance. From Ref. [97].

If we consider, however, that few of our achievements have made the step from qualitative understanding to improved engine designs, we should perhaps ask if a change in direction is needed. What should we be doing to make our research more useful? The answer to this is to consider the problems which are currently holding back the development of new engines. The following is a list of topics which might be considered:

(1) CFD has been of great benefit to modern compressor design, and the calculations have proved reliable near the design point. The inability to predict performance gradients near the peak of the characteristic is, however, a hindrance—particularly as it is these gradients which define the stability limit. Work on improving CFD predictions for flow conditions near the surge line is ongoing, but it is mostly being done by computational specialists without much input from the stall fraternity.
(2) One of the things which hinders CFD predictions of offdesign performance is the ineffectiveness of turbulence models at conditions near stall. Here, wakes are thicker and regions of separated flow are large. The turbulence models in current use are inadequate under these conditions and more research is needed. In addition, simple CFD methods generally average out properties between blade rows and, with the highly nonuniform flow which develops near stall, this averaging is probably misleading. Solving this problem will require detailed measurements and improved computer codes.
(3) The recent trend toward engines with smaller cores and larger gap-to-span ratios gives rise to a need for more work on clearance and eccentricity. The loss of stability, pressure rise, and efficiency which accompany a shift to larger tip gaps needs to be quantified. So do the effects of casing eccentricity. Questions of casing stiffness versus loss of stall margin also need to be investigated.
(4) There is an opportunity to develop new blade profiles based on alternative diffusion profiles. This is an old topic, but new ideas can now be studied more easily using modern CFD. Can new blades be designed to improve efficiency and stability?
(5) Due to manufacturing variations, blade sets are never perfectly uniform and the effects of small geometric deviations on the stability margin need to be investigated. Are these minor deviations significant enough to warrant inclusion in the CFD modeling? (Recent work suggests that small variations in the leading edge geometry are important in controlling boundary layer development on the blade surfaces.)
(6) A study is needed of the factors which control the loss of stall margin in older compressors. Foremost among these factors are blade deterioration (due to erosion and foreign object damage), leakage flows (due to worn seals) and, most importantly, tip clearance and eccentricity (due to rubbing). These all have a detrimental effect on fuel burn and thus need assessment and economic evaluation.
(7) In multistage compressors, stage matching is a problem which becomes more severe as pressure ratios increase. More work is needed on the scheduling of variable vanes and bleed valves—particularly when operating at part speed. Are current scheduling practices optimal or are improvements possible?

The above list contains some items which can only be studied using high-speed test rigs. Cost has always been a hindrance, but it is an unavoidable fact that the research questions which now need answering are not the same as those of 75 years ago.

Low-speed testing is not going to provide information about compressible effects, shock waves, or stage matching. A need thus exists for more investment in high-speed facilities which, to be relevant, need to be multistaged and of high-pressure ratio. This is an expensive request and cannot be answered by academic budgets alone. Perhaps the only way forward is for better cooperation between institutions and for more sharing of costs and results.

Another important parameter in the design of a test rig is pitch/chord ratio. In an aero-engine, the average pitch/chord ratio lies between 0.8 and 1.0 and so it is a matter of concern when stall inception studies are carried out in compressors with 16 or 20 blades and pitch/chord ratios of 1.2 and more. Using low-blade counts to reduce costs, or power consumption, seldom produces useful results.

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Nomenclature

A = area of compressor duct
H = total enthalpy
L = length of compressor duct
U = blade speed (midspan) or tip
V = volume of plenum chamber
$\phi = \dfrac{V_x}{U_{\text{mid}}}$ = flow coefficient
$\Psi_{T-S} = \dfrac{P_{\text{exit}} - P_{\text{o,in}}}{\frac{1}{2}\rho U_{\text{mid}}^2}$ = total-to-static pressure rise coefficient

Appendix: Experimental Shortcomings of the Past

In the experimental arena, there are pitfalls which can be avoided simply by learning from our past mistakes. The following is a list of some of the measuring and reporting practices which have not been satisfactory. By listing these shortcomings it is hoped to improve experimental practices and to equip us to read stall papers with more discernment.

In low-speed work, the pressure rise characteristics are often not clearly labeled as being of the "total-to-static" or the "total-to-total" type. Total-to-static measurements are by far the most useful when dealing with compressor stability because the slope of the curve is useful in indicating when the compressor might stall. The slope of a total-to-total, or static-to-static, characteristic is less useful in this regard.

When mapping the performance of a compressor, measurements recorded at discrete throttle settings are not of much use in determining the precise shape of the characteristic at the instant when stable operation ceases. The continuous recording of pressures and flow rates as the compressor is driven into stall provides useful "last minute" information.

The use of test rigs built for convenience rather than realism seldom leads to useful results. An example of this might be the use of a stationary blade row, rather than a moving one, just because it makes data collection easier. The effects of relative casing movement and centrifugal forces are neglected in such stationary blade row. The other disadvantage is that, after doing the convenient experiment, the problem remains of proving that the results are relevant.

A hidden problem often exists where lower than normal levels of stage loading are used in a research compressor. A stage with "delta H upon U squared" of 0.2 and flow coefficient of 0.3 (i.e., in the lower left-hand corner of the Smith chart) is not only unrepresentative of normal engine conditions (0.4 and 0.6, respectively) but also the stalling behavior of the compressor will be different. A low-pressure rise machine will experience little or no pressure drop when stall occurs and this will influence stall cell details (part-span/full-span) and the size of the hysteresis loop.

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