A Study of Spike and Modal Stall Phenomena in a Low-Speed Axial Compressor

T. R. Camp
Rolls-Royce plc,
Derby, United Kingdom

I. J. Day
Whittle Laboratory,
University of Cambridge,
Cambridge, United Kingdom

This paper presents a study of stall inception mechanisms in a low-speed axial compressor. Previous work has identified two common flow breakdown sequences, the first associated with a short length-scale disturbance known as a "spike," and the second with a longer length-scale disturbance known as a "modal oscillation." In this paper the physical differences between these two mechanisms are illustrated with detailed measurements. Experimental results are also presented that relate the occurrence of the two stalling mechanisms to the operating conditions of the compressor. It is shown that the stability criteria for the two disturbances are different: Long length-scale disturbances are related to a two-dimensional instability of the whole compression system, while short length-scale disturbances indicate a three-dimensional breakdown of the flow-field associated with high rotor incidence angles. Based on the experimental measurements, a simple model is proposed that explains the type of stall inception pattern observed in a particular compressor. Measurements from a single-stage low-speed compressor and from a multistage high-speed compressor are presented in support of the model.

Introduction and Background

The phenomena of stall and surge in compression systems have been studied almost continuously since the early development of the gas turbine engine. The objective of the work has always been the same, namely to extend the stable operating range of the compressor, but from time to time the immediate motivation for the work has changed. Over the past 20 years the driving problems have included hung stall, inlet distortion, casing treatments and, most recently, the application of active stall control. Active control was first proposed by Epstein et al. in 1986, and since then a significant amount of further research has been done, in both Europe and America. Most of this work can be divided into three categories: theoretical studies of control techniques (Simon et al., 1993; Feulner et al., 1996), the implementation of active control (Paduano et al., 1993; Day, 1993a; Haynes et al., 1994) and detailed studies of stall inception (Garnier et al., 1991; Day, 1993b; Tryfonidis et al., 1995; Hendricks et al., 1993). The work presented in this paper falls into the category of stall inception studies and builds on previous experimental work by Day (1993b).

Experimental work by McDougall et al. (1990) and Day (1993b) confirmed the existence of two different stall inception patterns in low-speed compressors. The first stalling pattern initiates with a short length-scale disturbance, which appears suddenly and develops directly into rotating stall. This type of disturbance, known generally as a "spike" because of its spikelike appearance in the velocity traces, is created by the localized stalling of a particular blade row. Figure 1 illustrates this type of stalling pattern, showing a sharp spike appearing on the velocity traces in an otherwise steady flow field. When the spike first emerges, it is small in circumferential extent and thus propagates quickly around the annulus, usually between 60 and 80 percent of rotor speed. (It is now well established that the fewer blade passages a stall cell occupies the faster it will rotate; Day, 1996). As the spike begins to propagate, it rapidly increases in size and its speed of rotation reduces. A high initial speed of rotation is characteristic of this type of disturbance.

The second type of stall inception pattern, which was predicted theoretically by Moore and Greitzer (1986) before being observed by McDougall et al. (1990), involves the gradual build-up of a long length-scale perturbation, which appears prior to the formation of a finite stall cell. The term "modal oscillation" is used to describe this phenomenon, a first-order mode having a wavelength equal to the circumference of the compressor, and a second-order mode having a wavelength of half the circumference, etc. An example of this phenomenon is given in Fig. 2, which shows a gentle undulation in the velocity traces prior to a gradual transition into rotating stall. In most cases a modal oscillation affects the flow throughout the length of the compressor, although it has recently been found that stage mismatching can limit the disturbance to a particular axial region. The rotational frequency of modal oscillations in low-speed compressors is usually less than 50 percent of rotor speed. (Modes of higher proportional frequencies may occur in high-speed machines, as found by Hendricks et al., 1993.)

The modal oscillation shown in Fig. 2 grows smoothly into a fully developed stall cell. Detailed measurements show that in such cases the low-velocity trough in the modal pattern initiates flow breakdown over a wide sector of the annulus. This results in a broad stall cell, which, because of its size, rotates comparatively slowly, in this case at a speed similar to the fully developed cell. The formation of a broad, slow-moving cell is thought to be associated with flow separation near the hub. In other situations the velocity trough in the modal wave will trigger flow separation in a localized region near the tip of one particular blade row. In such cases the transition from modal oscillation to rotating stall occurs via a spike disturbance, which initially propagates at a speed higher than the mode or the final

Contributed by the International Gas Turbine Institute and presented at the 42nd International Gas Turbine and Aeroengine Congress and Exhibition, Orlando, Florida, June 2–5, 1997. Manuscript received at ASME Headquarters February 1997. Paper No. 97-GT-526. Associate Technical Editor: H. A. Kidd.

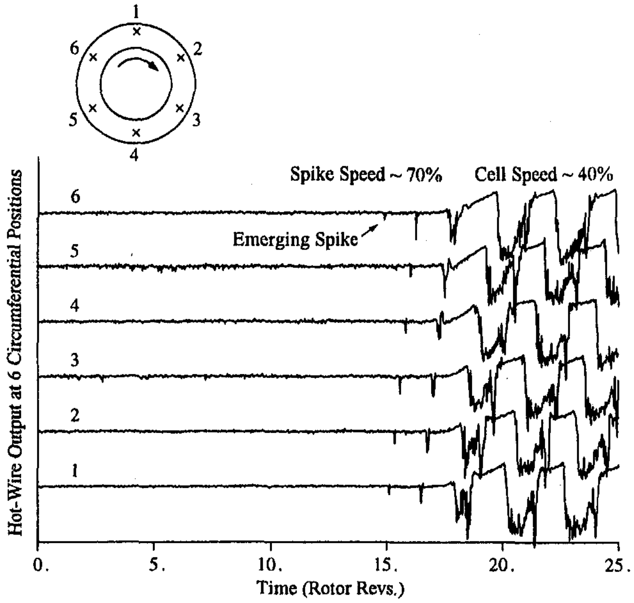

Fig. 1 Example of spike-type stall inception

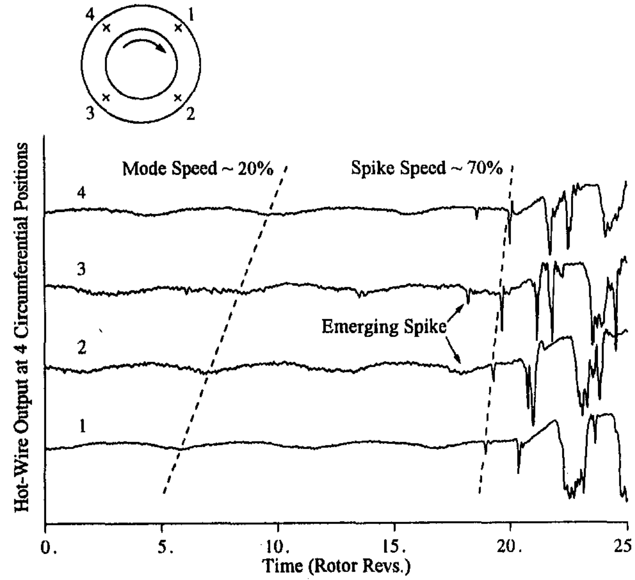

Fig. 3 Example of a spike-type stall cell originating in the trough of a modal velocity perturbation

stall cell. An example of a spike disturbance being initiated by a modal oscillation is given in Fig. 3. A mode can therefore be viewed as an oscillation of the flow-field, which promotes flow breakdown, either at the hub or at the casing, but which is distinct from the actual flow separation, which develops into rotating stall. This concept will be developed further in this paper.

As discussed above, stall in axial compressors is sometimes preceded by modal oscillations, while in other cases it initiates suddenly in the form of a spike. The criteria determining which of the two stalling sequences will occur in a particular compressor have been a topic of much discussion. It was shown by Day (1993b) that either type can occur in the same compressor if the tip clearance is changed. It has since been found that axial stage matching has a more pronounced effect on the stalling pattern than tip clearance. In this paper experiments are reported that identify the operating conditions that promote each of the stalling sequences. Based on these results, a model is proposed that sets out the stability criteria for the two phenomena. Before this is done, however, it is necessary to present some additional information on the physical properties of spikes and modes.

Physical Features of Spikes

When monitoring stall inception patterns using a circumferential array of probes, a spike can be identified by the narrow width of the disturbance (just one or two blade passages) and the high speed at which it rotates. The spike only retains this narrow structure and high rotational speed for a short time, seldom more than half a revolution around the circumference, before it increases in size and its speed decreases. Because the initial flow breakdown process is localized in both circumferential and axial extent, the identification of spikes is made easier if the probes are located near the particular blade row in which the spikes initiate. In many cases spikes remain unidentified, or

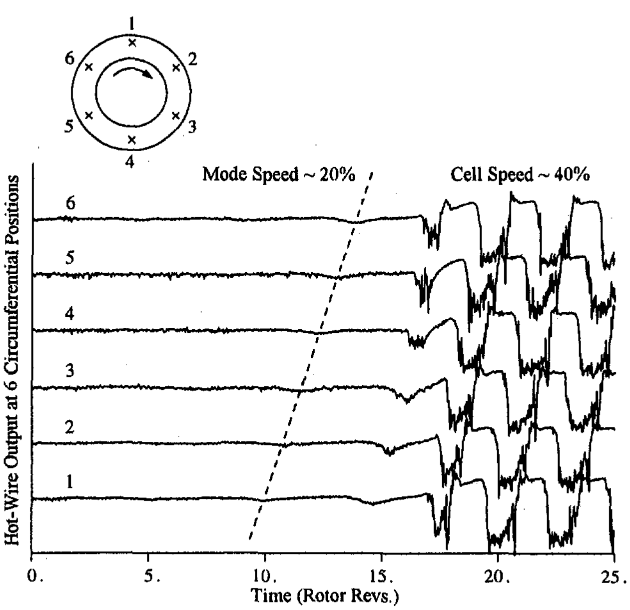

Fig. 2 Example of a modal perturbation preceding stall

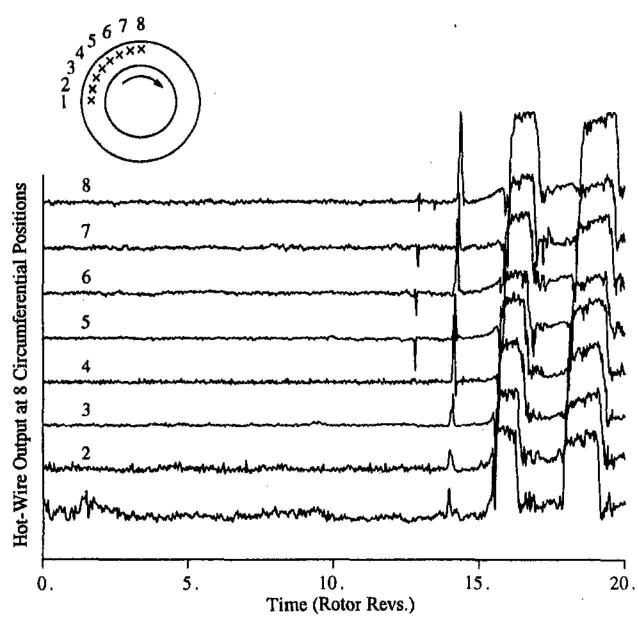

Fig. 4 Tangential velocity measurements showing the abrupt appearance of a spike

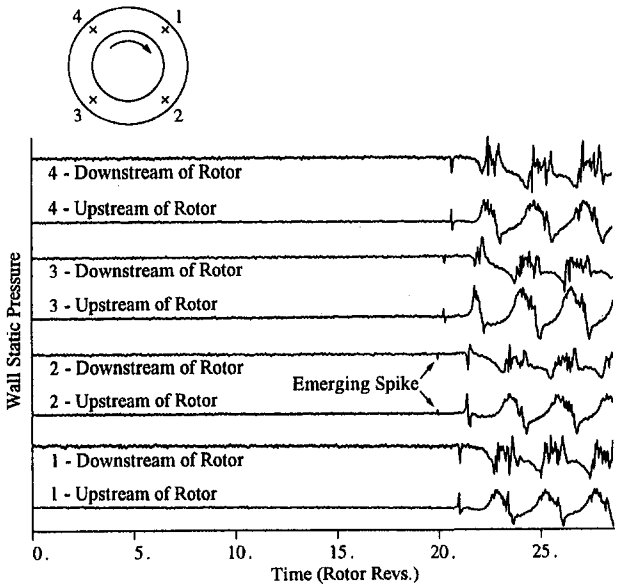

Fig. 5 Wall static measurements showing potential effect of stall cell blockage

are misclassified as modes, because they have originated at a location far removed from the measuring probes and therefore their signature is already broad and slow moving by the time the disturbance is detected.

Experimental work has been performed on the Cambridge four-stage compressor to examine the flow breakdown process associated with spikes. Details of the compressor are given in Appendix 1. Figure 4 shows the tangential velocities measured by an array of eight hot-wire probes positioned a quarter of a chord upstream of the first rotor and separated circumferentially by two blade pitches. The traces show the abrupt nature of the spike formation, the disturbance requiring only two or three blade pitches to become a significant disturbance. The measurements also show that the tangential velocity in the vicinity of the spike, as measured just upstream of the rotor face, is low when the disturbance first appears. From other measurements, such as those in Fig. 1, it is seen that the axial velocity is low in the same region. With low tangential and axial velocities upstream of the disturbance, the flow breakdown region has the appearance of a blockage in the flow-field, with an accompanying upstream stagnation region.

The picture of the spike as a localized blockage originating in one particular blade row is supported by the wall static pressure measurements shown in Fig. 5. These traces were recorded by pressure transducers positioned in axial pairs upstream and downstream of the same rotor row. When the spike first appears near the second pair of probes, the static pressure rises ahead of the disturbance (as it would in a stagnation region) while behind the rotor the pressure falls due to the localized loss of work input. The spike thus grows out of a disturbance that is initially confined to one or two blade passages in one particular blade row. Hot-wire measurements show that the initial flow separation is limited to the tip region of the rotor blades. This is illustrated by the measurements in Fig. 6 for which the probes were positioned alternately near the hub and near the casing. The traces show a drop in axial velocity in the stagnation region ahead of the spike near the rotor tip, and a corresponding increase in velocity near the hub where the flow is diverted under the blockage. The disturbance therefore originates near the blade tips and initially occupies only part of the span.

It has been suggested that spike formation may be related to the instability of the over-tip leakage flow. This point, and the precise details of the flow breakdown in the blade passage, have been examined computationally by Hoying (1996). Over-tip leakage flows also occur in cantilevered stator rows, but no sign of spike stall inception has yet been detected in these blade rows, possibly because of the high reaction of most of the blading in use.

In the Cambridge four-stage compressor, spikes are usually observed in the first rotor row. Experiments on other low-speed compressors having repeating blade geometry in each stage, also show that the first rotor is the most susceptible to spike initiation. It is thought that this occurs because the deviation angle of the flow leaving the inlet guide vanes is approximately constant as stall is approached, whereas the deviation angles from the downstream stator rows increase. Near the point of stall, the first rotor thus operates at a higher incidence than the downstream rotors and therefore it is this row that first succumbs to flow separation. However, spike-type stalling is not confined to the first stage, and can be induced in any of the four stages of the Cambridge compressor by simply restaggering the stator rows. The formation of spikes in the rear stages of high-speed compressors has also been observed, as reported by Day et al. (1998).

Characteristics of Modal Oscillations

In contrast with the sharply defined origins of spike-type disturbances, modal oscillations have no specific origin in time or space and the instant at which they are first detected is largely determined by the sensitivity of the measuring equipment. In most cases modal oscillations appear many revolutions before stall and intensify as the flow rate is reduced. The amplitude of the oscillation usually remains small and seldom exceeds 2 or 3 percent of the free-stream velocity before flow breakdown occurs and stall propagation begins. (However, under certain stage matching conditions, velocity fluctuations of more than 20 percent have been observed.)

In multistage low-speed machines, modal oscillations are usually of equal intensity from the inlet to the exit of the compressor, although variations in stage loading can restrict the perturbation to a particular axial region. An example of this is given in Fig. 7 for the case when a close-coupled screen was positioned upstream of the compressor to simulate a stable group of stages. As shown, modal activity was confined to the front of the compressor in this case. The results in this figure are particularly interesting because they show a spike initiating

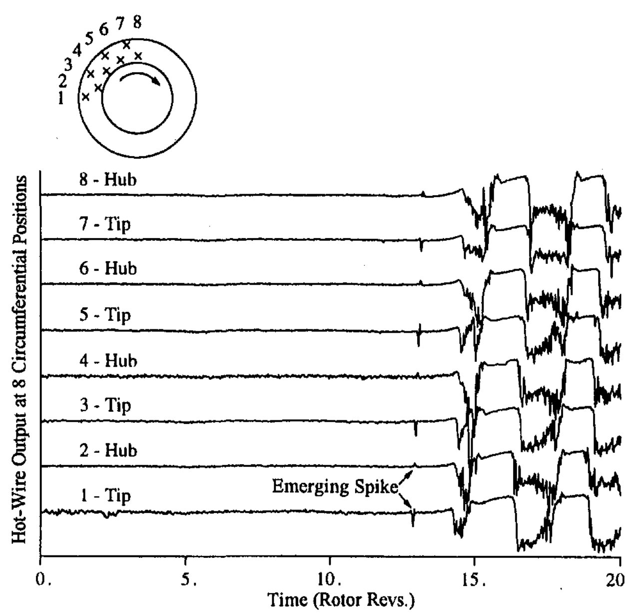

Fig. 6 Hot-wire measurements showing increase in hub velocity owing to spike blockage at the casing

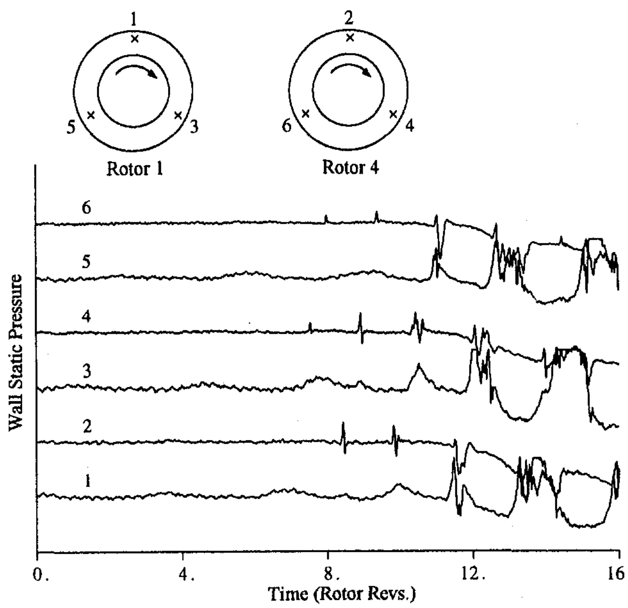

Fig. 7 Static pressure measurements showing modal activity at rotor 1 and spike-type stall initiating in rotor 4

stall from the rear of the compressor independently of the modal activity at the front.

While a spike can be regarded as an embryonic stall cell, which develops continuously into rotating stall, a modal oscillation is not in itself an early form of stall cell. Instead, a mode is an oscillation of the flow-field, which appears close to the peak of the total-to-static pressure rise characteristic, as discussed by Moore and Greitzer (1986). At the peak of the characteristic, some or all of the blade rows are operating close to their stalling limits and hence any modal induced velocity deficit may be sufficient to initiate flow separation, either in the form of a spike, as in Fig. 3, or as a broader stall cell, as shown in Fig. 2.

The observation that modal oscillations occur at the peak of the pressure rise characteristic is illustrated by the example in Fig. 8 where, owing to the presence of a downstream screen, the characteristic has a well-defined peak. In this case, sustained modal activity appears only at the peak of the characteristic and is not present at other flow rates where the slope is non-zero. The lower part of Fig. 8 shows the output from a hot-wire ahead of rotor 1 at three different flow coefficients. At high flow rates the flow field is steady, but at the peak of the characteristic a clear modal perturbation is present, as shown in the center trace. At lower flow coefficients the velocity fluctuation disappears and the flow field is again axisymmetric. Stalling was eventually initiated by a spike at the front of the compressor.

In this discussion we have tried to show that modal perturbations are a particular form of flow instability that occurs when the compressor operating point approaches the peak of the characteristic. A modal perturbation can theoretically develop smoothly into a large nonlinear disturbance, but in most cases its development is interrupted by the appearance of a stall cell. The reason for distinguishing modal activity from the actual flow breakdown process will become clear in the following section where it is shown that stalling occurs when a critical value of blade incidence is exceeded. The velocity fluctuations associated with modes increase the local blade incidence in certain regions of the annulus and therefore the intensity of modal activity required to initiate stall depends on how near or far any of the blade rows in the compressor are from critical incidence. In some situations critical incidence may be exceeded somewhere in the compressor before the peak of the characteristic is reached, in which case stalling, usually of the spike type, will occur before modes have developed.

Axial Matching Experiments

So far it has been established that modes and spikes are different aspects of the stall inception process and that both phenomena can be observed in the same compressor. To investigate the effects of characteristic slope and local blade incidence on the stall inception pattern, a series of experiments was conducted on the Cambridge compressor, making use of its variable stagger IGV and stator rows. Stage pressure rise characteristics and the stalling patterns of the compressor were measured for a large number of settings of the stators and IGVs. The results of two specific experiments are presented below, showing the effect of changing the relative loadings of stage 1 and the group of stages 2–4. This division of the compressor between stage 1 and the remaining downstream stages was made for two reasons. First, it has been observed that the first rotor row in a low-speed multistage compressor of repeating stage design is the most susceptible to flow separation. Second, because the IGV deviation angle is small and the boundary layer at IGV exit is relatively thin, the rotor 1 incidence angle can be controlled more accurately than the incidence onto any of the downstream blade rows.

To measure the stall inception events in these experiments, circumferential arrays of six hot-wire probes were used, following the procedure described by Day (1993b). From the measurements, the presence of modal oscillations prior to stall was determined, either by eye from the shapes of the velocity traces or, in cases which were unclear, by using the spatial Fourier analysis procedure first demonstrated by McDougall et al. (1990).

When measuring the compressor characteristics, it was found that by using the traditional method of recording the flow rate and pressure rise at relatively few throttle positions, the shape

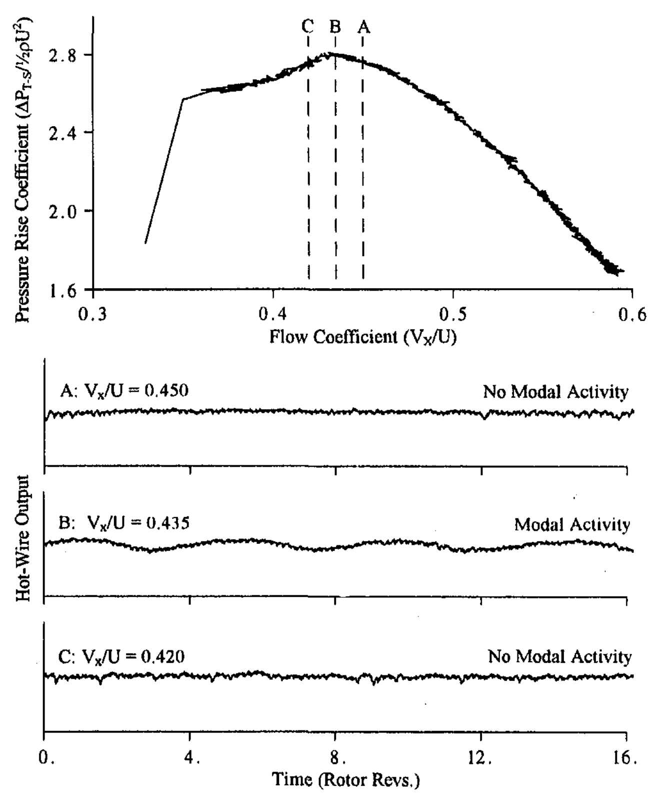

Fig. 8 An example of modal activity that does not lead directly to stall (downstream screen fitted)

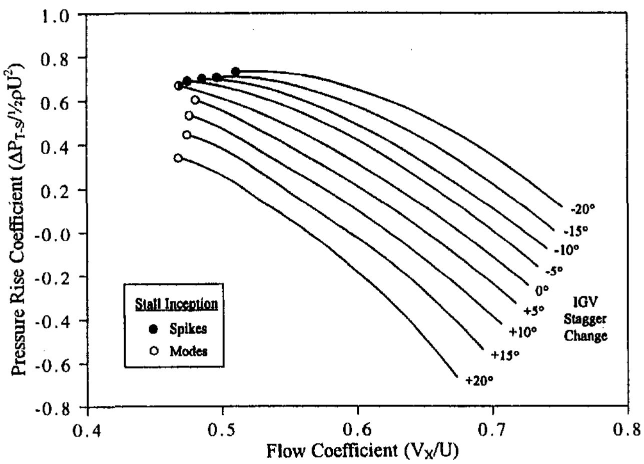

Fig. 9 Stage 1 characteristics as a function of IGV stagger angle

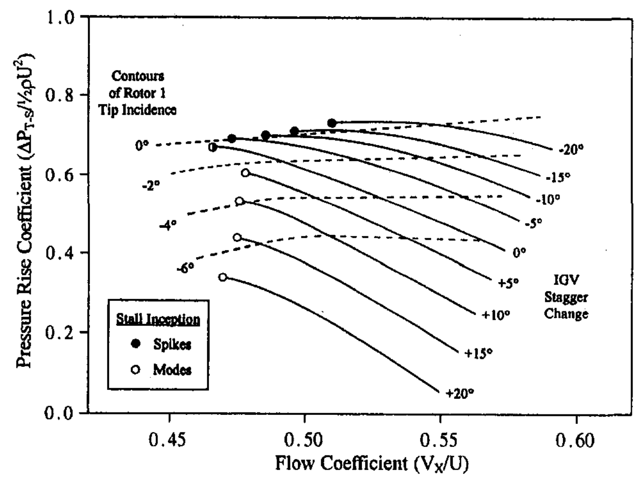

Fig. 11 Enlargement of Fig. 9 showing contours of rotor 1 tip incidence

of the characteristic in the region of the stall point was not sufficiently well defined. To overcome this limitation, an "on-line" data acquisition system was used, which recorded the pressure rise and flow rate of the compressor at every revolution of the rotor. While sampling continuously, the motor-driven throttle valve was closed slowly until the compressor stalled. Using this method, it was possible to record continuous characteristics right up to the instant of stall.

In the first experiment the stalling patterns and stage characteristics were measured for a range of IGV stagger angles while the downstream stators were held fixed at their design settings. The stagger of the IGVs was varied from −20 to +20 deg in steps of 5 deg, relative to their design position. Figure 9 shows the pressure rise characteristics of stage 1 for each stagger setting of the IGVs. The operating points at which the compressor stalled are shown in this figure by circular symbols, an open symbol indicating that modal activity was present at stall onset and a closed symbol indicating that spikes appeared in an axisymmetric flow-field. Figure 10 shows the complementary characteristics of the group of stages 2, 3, and 4, plotted to a larger scale than Fig. 9 in order to distinguish the individual stall points. As expected, the stage 1 characteristics are displaced relative to each other because the inlet flow angle to this stage changed as the IGVs were restaggered. In contrast, the characteristics of stages 2–4 follow a nearly unique curve because stators 2–4 and the inlet flow angle to this group of stages (principally determined by the stagger of stator 1) were held fixed. Figures 9 and 10 show that for the design geometry (IGV stagger of 0 deg) modal oscillations were observed in some cases but not others. When the IGV stagger angles were lower than design (increased rotor 1 incidence) modes did not appear and spikes initiated at rotor 1 in an otherwise steady flow-field. Conversely, for IGV stagger angles greater than design (decreased rotor 1 incidence) stall was always preceded by a clear first-order modal perturbation.

Figure 11 shows enlarged regions of the stage 1 characteristics from Fig. 9, on which contours of rotor 1 tip incidence have been plotted. The method used to estimate rotor tip incidence for this purpose is described in Appendix 2. From these measurements we see that when the compressor stalled via a spike in a clean, non-modal flow-field, the rotor 1 incidence angles at stall were approximately constant. However, when stall was preceded by a modal oscillation, the average rotor 1 incidence angle at the stall point was no longer fixed but covered a range of more negative values. The decision to plot rotor tip incidence in this figure was influenced by the observation that it is at the tip of the rotor blades where stall cells initiate.

Figure 12 shows regions near stall of the total-to-static pressure rise characteristics of the whole compressor as a function of IGV stagger angle. This figure shows that modes appeared prior to stall when the slope of the overall characteristic was zero. When modal oscillations were not present and stall cells grew via spikes from an axisymmetric flow-field, this occurred before the peak of the characteristic was reached, on the negatively sloped region of the characteristic.

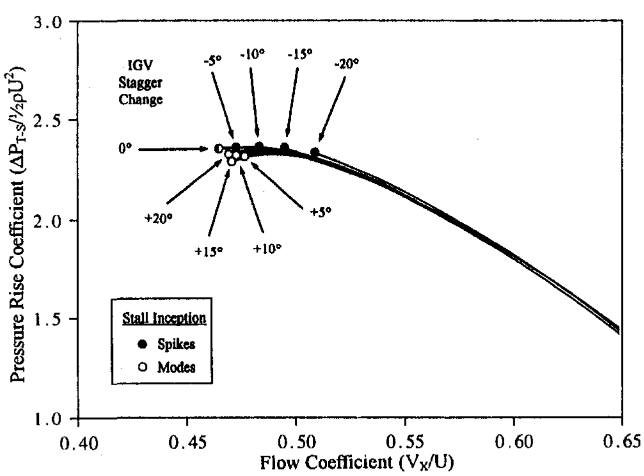

Fig. 10 Stage 2–4 characteristics as a function of IGV stagger angle

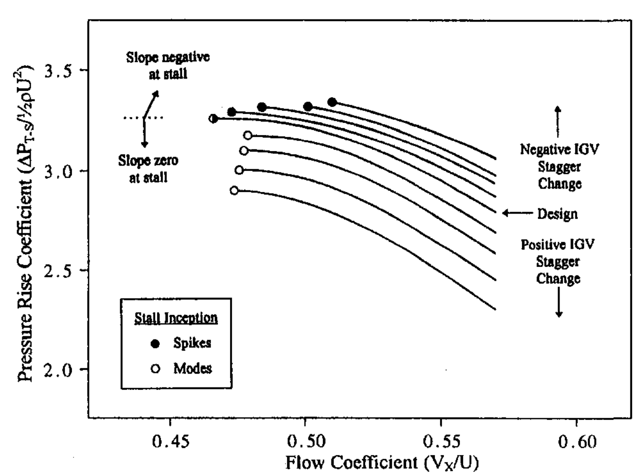

Fig. 12 Stages 1–4 characteristics as a function of IGV stagger angle

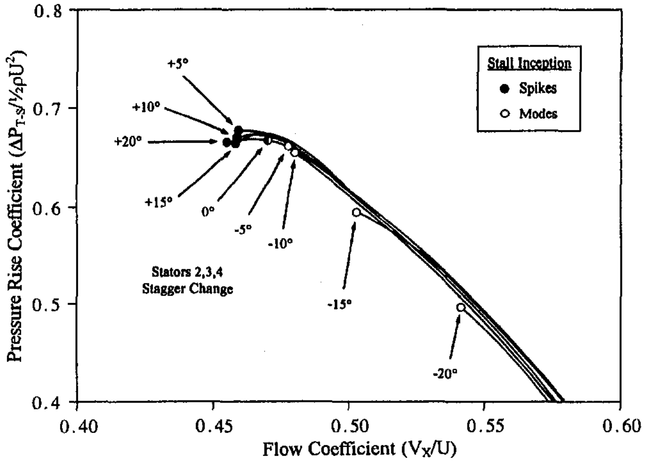

Fig. 13 Stage 1 characteristics as a function of stator 2, 3, and 4 stagger angles

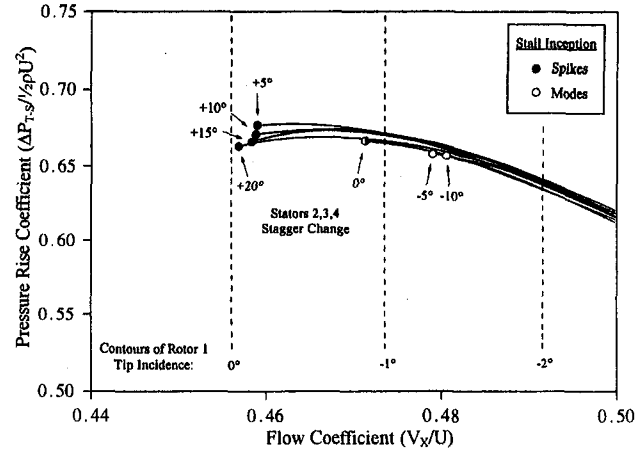

Fig. 15 Enlargement of Fig. 13 showing contours of rotor 1 tip incidence

To check on the generality of these observations, a second experiment was performed in which, rather than changing the flow conditions into the first stage, the geometry of the first stage was held fixed while the conditions in the downstream stages were varied. For this the stagger angles of the IGVs and stator 1 were held at their design settings, while the angles of stators 2, 3, and 4 were changed by equal amounts in steps of 5 deg from −20 to +20 deg relative to their datum positions. The pressure rise characteristics of stage 1 are shown in Fig. 13, while the characteristics of the group of stages 2–4 are shown in Fig. 14. The symbols marking the stall points on these characteristics show that when the stagger angles of stators 2–4 were increased relative to their design setting (increasing the negative slopes of their stage characteristics and thereby increasing the stability of stages 2–4 relative to stage 1) modal oscillations did not occur before stall; instead spike stall cells were initiated at rotor 1. However, when the stagger angles of stators 2–4 were decreased, stages 2–4 became more highly loaded than stage 1 and stall was generally preceded by a modal oscillation. (For stagger changes of −5 and −10 deg, spikes sometimes initiated at rotor 4.)

Figure 15 shows enlarged regions of the stage 1 characteristics overplotted with contours of rotor 1 incidence. Again it can be seen that spikes were initiated in rotor 1 at an approximately constant value of rotor incidence and that when modal behavior appeared before stall this occurred at lower values of average rotor 1 incidence. The value of rotor 1 incidence at which spikes were initiated (about 0 deg) is the same value at which spikes were initiated in the first experiment, as shown in Fig. 11. Figure 16 shows the total-to-static characteristics for the overall compressor near the stall points. Again modal oscillations appeared prior to stall when the slope of the overall pressure rise characteristic was zero or just positive. When the compressor stalled on a negatively sloped region of the characteristic, spike-type stall cells were seen to grow from an axisymmetric flowfield.

Simple Model and Discussion

The low-speed results described above lead to two important observations:

1 When stall is preceded by modal oscillations, the slope of the overall compressor characteristic is zero or slightly positive at the stall point. Alternatively, when stall initiates without a modal oscillation being present, this occurs on a negatively sloped region of the characteristic.

2 When stall is initiated by spike stall cells in rotor 1, this occurs at a limiting value of rotor 1 incidence angle. When stall is preceded by modal oscillations, rotor 1 incidence angles at the stall point are below this limiting value.

From these observations it is possible to propose a simple model to explain why, in some cases, stall is preceded by modal oscillations while in others stall cells grow from an initially

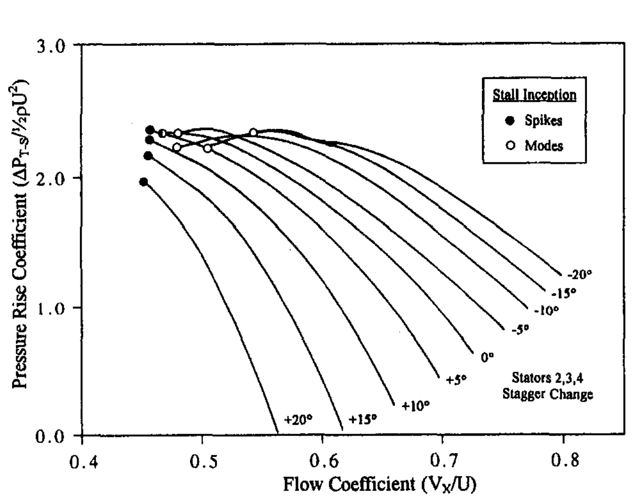

Fig. 14 Stage 2–4 characteristics as a function of stators 2, 3, and 4 stagger angles

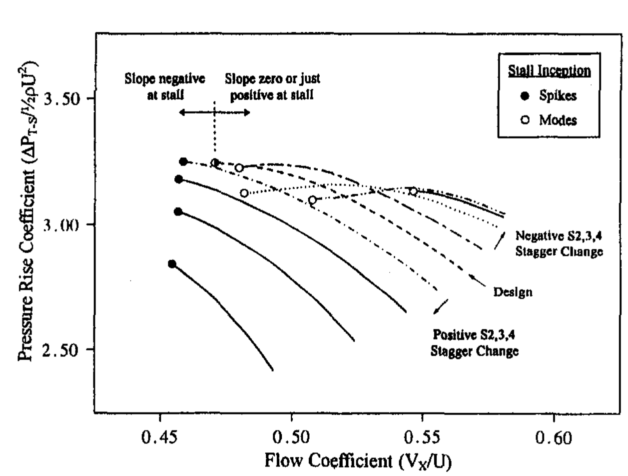

Fig. 16 Stage 1–4 characteristics as a function of stator 2, 3, and 4 stagger angles

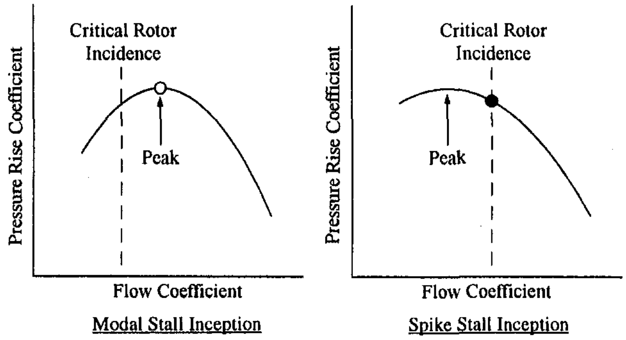

Fig. 17 Diagram illustrating the model

axisymmetric flow-field. The same model explains why some compressors stall at the peak of the characteristic while others stall while the characteristic is still rising. Stated simply, if the peak of the overall characteristic is reached before the critical value of rotor incidence is exceeded, then modal oscillations will occur. If, on the other hand, the critical rotor incidence is exceeded before the peak of the overall characteristic is reached, then spikes will appear in the overloaded rotor before modal activity has had an opportunity to develop. Whichever of these two stability limits is reached first will determine the stalling pattern of the compressor. Figure 17 illustrates these ideas diagramatically.

The general validity of the model is confirmed by Fig. 18, which summarizes the results of tests at various other IGV and stator stagger settings. In these tests the stagger angles of stators 2, 3, and 4 were not constrained to be the same, but were varied independently. The ordinate in Fig. 18 indicates the rotor 1 tip incidence, while the abscissa shows the gradient of the total-to-static pressure rise characteristic at the stall points. Each test symbol indicates whether modal oscillations were observed prior to stall or whether spike-type stall cells grew from a previously axisymmetric flow-field. Of all the measurements taken, the only data to be excluded from Fig. 18 are the few cases where stall was initiated by a spike mechanism at a rotor other than rotor 1. The figure shows that the symbols are grouped according to whether or not modal oscillations appeared before stall. The data points corresponding to stall without modes (spike inception at rotor 1) lie around a line of constant rotor 1 incidence and at positions that indicate negative slopes of the characteristics at stall. In contrast, points corresponding to modal behavior lie close to a line of zero characteristic slope and at rotor incidences below the critical value at which spikes initiate.

Single-Stage Experiments. In the experiments described above, which were conducted in a high hub–tip ratio compressor, the radial distribution of incidence was little affected by changes in stagger setting. There is of course no reason why the occurrence of critical incidence at the rotor tips could not be changed by changes in the twist of the blades or by changes in the radial distribution of the flow. This point has recently been demonstrated in a single-stage compressor where the appearance of modes or spikes was influenced by artificially skewing the flow toward the casing or the hub. A small amount of additional wall blockage was introduced in the vicinity of the stator blades, which had the effect of slightly altering the radial distribution of the flow approaching the rotor. The results show modal activity accompanying a "turned over" characteristic when the flow was diverted to the casing (increased tip velocity and therefore reduced incidence), and clear spikes on a rising characteristic when the flow was diverted to the hub (reduced tip velocity and increased incidence).

High-Speed Compressor Evidence. In a high-speed compressor the axial matching, and hence the rotor incidence, may be changed not only by adjusting variable blade rows but also by changing the shaft speed. If the compressor is well matched at the design speed, then at lower speeds the blading in the front stages will be subject to higher incidence than the rear stages while at higher speeds the blading in the rear stages will experience the highest incidence angles as stall is approached. According to the model discussed above, one would expect different stall inception mechanisms to occur at different speeds. Over a middle range of speeds where the compressor is well matched at the stall point (somewhat lower than the design speed if the design point is some distance from the stall line), stall will be preceded by modal oscillations of the flow-field. At lower speeds modes will not occur because spikes will initiate in the front stages, where the incidence angles are greatest. Likewise at higher speeds we would expect spike-type stalling in the rear stages.

Results from high-speed compressor tests conducted under the BRITE-EURAM Civil Core Compressor Project (Day et al., 1997) show that modal activity is indeed confined to the mid-speed range. Figure 19 shows a set of results from the MTU three-stage compressor studied in this project. At 60 percent speed the compressor stalls with spikes on the first rotor, while at 80 percent speed there is clear modal activity prior to stall. At higher speeds, spike-like behavior was sometimes observed, but owing to the very fast breakdown of the flow-field, the precise initiation mechanism was difficult to classify. Measurements from other compressors in the European project also show a clear trend from spikes to modes as the speed of the compressor is increased from low to medium speeds of rotation. It is interesting to note that the Rolls-Royce Viper compressor tested in the same project yielded modal activity in only a very narrow speed range between 85 and 87 percent. It seems that this occurred because the Viper compressor is relatively long, having eight stages, and therefore it is rare that equal loading on all stages occurs without one stage experiencing higher incidences than the rest.

Influence of Blade Design on Stalling Pattern

As described above, spikes are associated with flow breakdown at the rotor tips and so occur more readily in "tip critical" blading, such as the Cambridge compressor. For the case of blading that is not tip critical, it is thought that the increase in blockage and loss associated with hub corner separation leads to a "turned over" characteristic, which in turn promotes two-dimensional modal behavior. Indeed, the blockage created by separation near the hub tends to divert the flow toward the outer

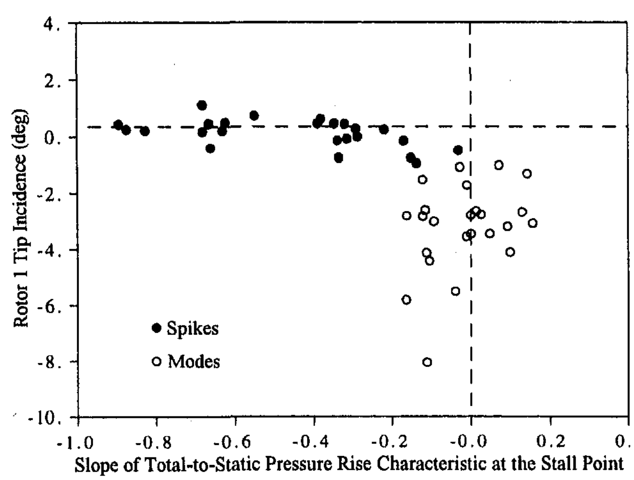

Fig. 18 Stall inception results for many different compressor configurations

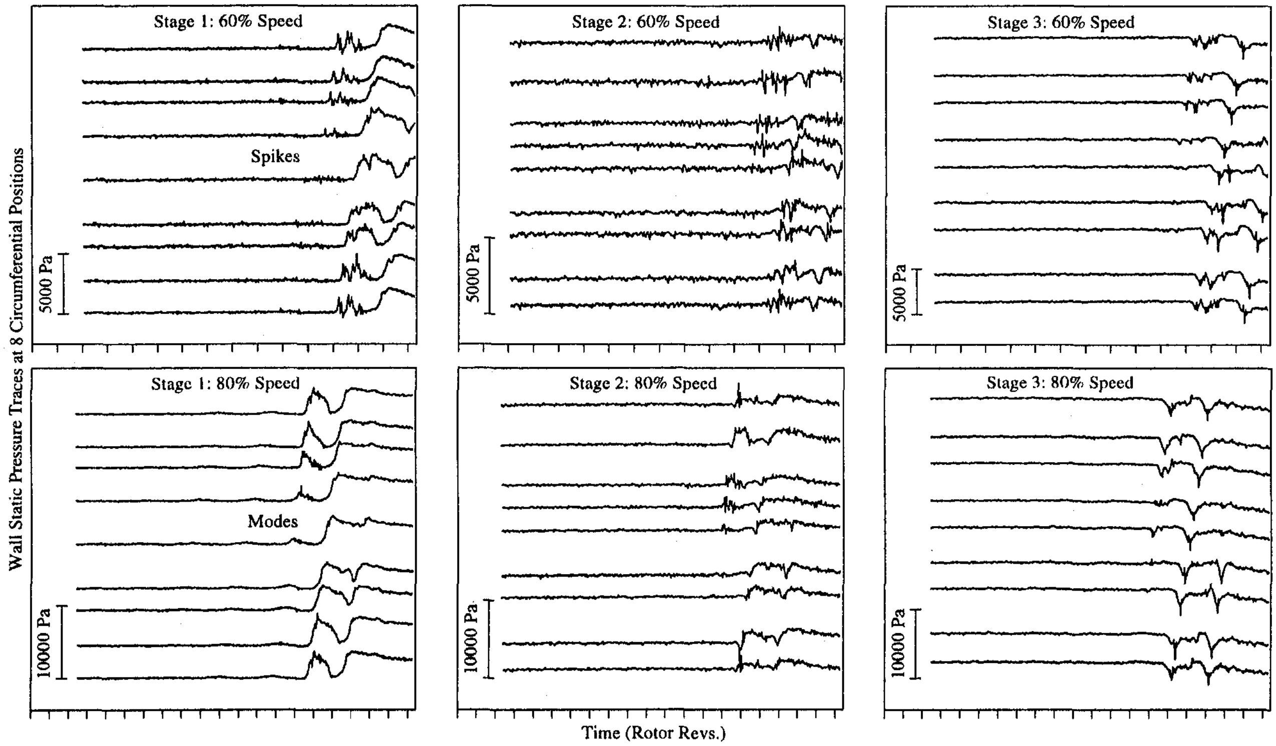

Fig. 19 Casing static pressure traces from the MTU three-stage compressor showing changes in the stall inception pattern with shaft speed

casing, thereby reducing rotor tip incidence and suppressing spike formation (as found in the single-stage experiments described above).

To go one step further, in a situation where the characteristic has turned over and modes have developed, the velocity deficit associated with the modes will, when large enough, precipitate flow breakdown and rotating stall. The actual breakdown process will include spikes if, at this stage, the tip is closest to stall, Fig. 3, or will take the form of a broad, slow-moving separation if the hub is the first to separate, as shown in Fig. 2. (The formation of a broad stall cell originating from separated flow at the hub has been reported by McDougall et al., 1990.) This argument does not take into account the contribution of the stator rows to the overall flow and hence to the shape of the characteristic. It has, however, been shown above that corner separations in the stator row at the casing will promote rotor spikes, while separation at the hub will favor the formation of modes.

The view presented here of the relationship between blade design and the stall inception pattern is simplistic and is open to a number of criticisms of detail, but it is consistent with most of our observations from low-speed single-stage and multistage testing.

Conclusions

This paper gives details of the short and long length-scale disturbances that precede stall in low and high-speed compressors. Using the variable geometry features of a four-stage low-speed compressor, we have investigated the situations in which each type of disturbance occurs, and have formulated a descriptive model, which may be used to explain the type of stall inception in a given situation. The generality of this model is supported by results from a single-stage compressor and from multistage high-speed compressors. In detail, the work has shown that:

1 Spike stall inception describes localized flow separation in one particular blade row of the compressor and occurs when a critical incidence is exceeded. This form of stall inception appears to be restricted to the tip region of rotor rows.

2 Modes are a circumferential oscillation of the flow-field occurring at the peak of the total-to-static pressure rise characteristic. When present, a modal oscillation may give rise to local flow conditions in which the incidence onto a particular blade row exceeds the critical value, giving rise to spikes. Alternatively a modal oscillation may give rise to separations in a larger number of blade passages, leading to the formation of a broader, more slowly rotating stall cell. (This type of cell formation is thought to be attributed to flow separation near the hub.) The amplitude to which a modal oscillation will grow before stalling begins depends on how close any particular blade row in the compressor is to being critically overloaded.

3 Modes and spikes may occur in the same compressor, sometimes simultaneously. The stability criteria for each are different: Long length-scale disturbances are related to a two-dimensional instability of the whole compression system, while short length-scale disturbances indicate a three-dimensional breakdown of the flow-field associated with high rotor incidence angles. Tests show that the occurrence of each phenomenon can be influenced by the stage matching, which in turn may be altered by changing blade stagger angles or, in the case of a high-speed compressor, by changing the shaft speed. A change between modes and spikes in a single-stage compressor has also been demonstrated by changing the radial distribution of rotor incidence.

4 A simple model has been proposed, based on the experimental results, which may be used to explain the occurrence of modes or spikes in a given situation. Stated simply, as the compressor flow rate is reduced, if the peak of the overall characteristic is reached before the critical value of rotor incidence is exceeded anywhere in the compressor, then modal oscillations will occur. However, if the critical rotor tip incidence is exceeded before the peak of the overall characteristic is reached then spikes will appear in the overloaded rotor, initiating stall before modal activity has had an opportunity to develop. The model thus explains why some compressors stall at the peak of

the characteristic and why others stall while the characteristic is still rising.

Acknowledgments

The authors would like to acknowledge the support of Rolls-Royce plc, who funded much of this research. We would also like to acknowledge the BRITE-EURAM program and to thank MTU for allowing the publication of the high-speed test data shown in Fig. 19. The suggestions made by several colleagues during the course of the work were very valuable and we would particularly like to thank Prof. N. A. Cumpsty, Prof. E. M. Greitzer, and Dr. D. Wisler.

References

Camp, T. R., 1995, "Aspects of the Off-Design Performance of Axial Flow Compressors," Ph.D. dissertation, Cambridge University Engineering Department.

Day, I. J., 1993a, "Active Suppression of Rotating Stall and Surge in Axial Compressors," ASME JOURNAL OF TURBOMACHINERY, Vol. 115, pp. 40–47.

Day, I. J., 1993b, "Stall Inception in Axial Flow Compressors," ASME JOURNAL OF TURBOMACHINERY, Vol. 115, pp. 1–9.

Day, I. J., 1996, private communication (to be published).

Day, I. J., Breuer, T., Escuret, J. F., and Cherrett, M., 1998, "Stall Inception and the Prospects for Active Control in Four High Speed Compressors," ASME JOURNAL OF TURBOMACHINERY, Vol. 120, in press.

Epstein, A. H., Ffowcs-Williams, J. E., and Greitzer, E. M., 1986, "Active Suppression of Aerodynamic Instabilities in Turbomachines," AIAA *Journal of Propulsion and Power,* Vol. 5, No. 2, pp. 204–211.

Feulner, M. R., Hendricks, G. J., and Paduano, J. D., 1996, "Modeling for Control of Rotating Stall in High-Speed Multistage Axial Compressors," ASME JOURNAL OF TURBOMACHINERY, Vol. 118, pp. 1–10.

Garnier, V. H., Epstein, A. H., and Greitzer, E. M., 1991, "Rotating Waves as a Stall Inception Indication in Axial Compressors," ASME JOURNAL OF TURBOMACHINERY, Vol. 113, pp. 290–301.

Haynes, J. M., Hendricks, G. J., and Epstein, A. H., 1994, "Active Stabilization of Rotating Stall in a Three-Stage Axial Compressor," ASME JOURNAL OF TURBOMACHINERY, Vol. 116, pp. 226–239.

Hendricks, G. J., Bonnaure, L. P., Longley, J. P., Greitzer, E. M., and Epstein, A. H., 1993, "Analysis of Rotating Stall Onset in High Speed Axial Flow Compressors," AIAA Paper No. 93-2233.

Hoying, D. A., 1996, "Blade Passage Flow Structure Effects on Axial Compressor Rotating Stall Inception," Ph.D. Dissertation, Dept. of Aeronautics and Astronautics, M.I.T.

McDougall, N. M., Cumpsty, N. A., and Hynes, T. P., 1990, "Stall Inception in Axial Compressors," ASME JOURNAL OF TURBOMACHINERY, Vol. 112, pp. 116–125.

Moore, F. K., and Greitzer, E. M., 1986, "A Theory of Post-Stall Transients in Axial Compression Systems: Part I—Development of Equations," ASME *Journal of Engineering for Gas Turbines and Power,* Vol. 108, pp. 68–76.

Paduano, J., Epstein, A. H., Valavani, L., Longley, J. P., Greitzer, E. M., and Guenette, G. R., 1993, "Active Control of Rotating Stall in a Low Speed Axial Compressor," ASME JOURNAL OF TURBOMACHINERY, Vol. 115, pp. 48–56.

Simon, J. S., Valavani, L., Epstein, A. H., and Greitzer, E. M., 1993, "Evaluation of Approaches to Active Compressor Surge Stabilisation," ASME JOURNAL OF TURBOMACHINERY, Vol. 115, pp. 57–67.

Tryfonidis, M., Etchevers, O., Paduano, J. D., Epstein, A. H., and Hendricks, G. J., 1995, "Pre-stall Behavior of Several High-Speed Compressors," ASME JOURNAL OF TURBOMACHINERY, Vol. 117, No. 1, pp. 62–80.

APPENDIX 1

Details of the Cambridge Four-Stage Compressor

The multistage compressor used for the low-speed experiments described in this paper was the C106 compressor at the Whittle Laboratory, Cambridge University. This compressor has four identical stages preceded by a lightly loaded row of inlet guide vanes. The stator vanes in this compressor are cantilevered. The hub/casing radius ratio is 0.75 and the speed of rotation is 3000 rpm, giving a maximum Reynolds number of 2×10^5, based on rotor chord. Details of the blading are given in Table 1. The inlet guide vanes and each row of stators are fitted with a variable stagger mechanism, similar to that used in high-speed compressors, which allows these blade rows to be easily restaggered. Further details are given by Camp (1995).

Table 1 Details of the C106 compressor

	IGV	Rotor	Stator
No. of Aerofoils	60	58	60
Mid-Height Chord (mm)	36.0	35.6	36.2
Mid-Height Stagger (deg)	6.5	-44.4	23.1
Mid-Height Camber (deg)	-17.3	17.8	37.6
Mid-Height Solidity	1.55	1.47	1.55
Aspect Ratio	1.76	1.78	1.75
Tip Clearance (% chord)	1.0	1.0	1.0
Axial Gap (% chord)	42	46	45
Reynolds Number	1.04×10^5	2.00×10^5	1.46×10^5
Va / Umid	0.52		
Δho / Umid^2	0.40		
Reaction (%)	68		
Static Pressure Rise Coef.	0.70		
Rotor Tip Mach Number	0.23		

APPENDIX 2

The Calculation of Critical Rotor Incidence

To estimate the values of rotor incidence, contour plotted in Figs. 11 and 15, it was assumed that the IGV deviation angle was constant at its design value and that the axial velocity at the rotor tip was equal to the area-mean axial velocity, calculated over the whole annulus. The increase in rotor incidence due to the casing boundary layer was therefore not accounted for, and thus the incidence values given in these figures underestimate the true rotor tip incidence, but by a relatively constant amount. To check that the restaggering of the IGVs did not cause significant changes to the casing boundary layer or to the IGV deviation, area traverses of a hot-wire probe were performed at the IGV exit plane at operating points near stall for each stagger setting. It was found that the IGV stagger angle had little effect on the thin casing boundary layer at this compressor inlet plane and that the axial velocity profiles remained flat outside the boundary layers. The assumption of a constant IGV deviation angle was also confirmed by the measurements. In summary, the IGV restaggering produced clean changes to the rotor 1 incidence without significant changes to the inlet boundary layers or to the radial distribution of the flow.

In this description of results and in the proposed model, the initiation of stall cells in a previously axisymmetric flow-field (spike stall inception) was linked to a critical value of blade incidence being exceeded. The question arises as to whether blade incidence angle is the most appropriate parameter with which to predict the formation of stall cells. It is unlikely, for example, that the critical value of incidence would be the same for all profile types and all values of camber, space-chord ratio, etc. The formation of stall cells implies separation of the flow from the blade surfaces, endwall or (perhaps more likely) a corner. Therefore it might be thought that a loading parameter, such as diffusion factor, would correlate better with spike formation. It was found, however, that when Fig. 18 was replotted using rotor diffusion factor instead of incidence, the collapse of the datapoints was not as good. Further work is required to clarify which particular measure of blade or endwall loading is the most appropriate guide to spike stall inception.