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Stall Inception in Axial Flow Compressors

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ABSTRACT

Studies have been conducted on two laboratory test compressors to investigate the process leading to the formation of finite amplitude rotating stall cells. The measurements were obtained from circumferential arrays of hot-wires and were spatially and temporarily analysed to show that modal perturbations are not always present prior to stall, and when present, sometimes have little direct effect on the formation of the stall cells. The measurements lead to the conclusion that the occurrence of modal perturbations, and the formation of finite amplitude stall cells, are two separate phenomena; both occurring under roughly the same conditions at the peak of the pressure rise characteristic. The measurements also underline the hitherto unsuspected importance of short length scale disturbances in the process of stall inception. Examples are given of different ways in which stall cells can develop and the conclusions are backed up with a summary of current test data from various machines around the world.

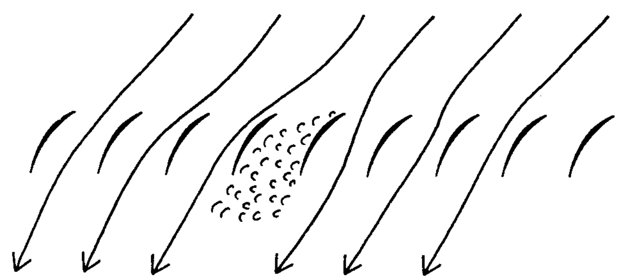
INTRODUCTION

The present trend in aero-engine compressors is towards increased pressure ratio per stage. The need to retain an adequate operating range under these conditions has stimulated renewed interest in compressor instabilities. Rotating stall cells and transient reversed flow (surge) are limiting factors at peak performance of the compressor. New attempts, using better instrumentation and improved analytical models, are therefore being made to understand these complex flow phenomena.

Past efforts in this field have met with limited success. Both rotating stall and surge can be reasonably well described in terms of what happens in the compressor when either disturbance is well established. However, a gap exists in the understanding of how the disturbances come into being and we are still unable to predict the precise operating point at which this instability will occur. In recent years it has become clear that the idea of using only steady state parameters to determine the stall point of a compressor is too simplistic and the focus of attention has now shifted to problems of aerodynamic stability. The current experimental work on stall inception is aimed at providing a sound physical understanding of the phenomenon on which to base improve modelling of compressor stability.

A heightened interest in stall inception has also been aroused with the recent attempts to extend the compressor operating range through the use of active control techniques. The idea was first published in the open literature by Epstein et al (1986) and since then successful experiments have been carried out by Day (1991), and Paduano et al (1991). The objective in applying active control to compressor instabilities is to feed back damping perturbations into the flow field which will suppress the development of the stalling disturbances.

Two flow models of particular interest to the current work can be found in the literature. It will be useful to review these before going on to look at the experimental results. The first flow model is that derived from a picture originally drawn by Emmons et al, (1955). This picture suggests a possible cause of stall inception and offers an explanation of how the stall cell propagates along the blade row.



Sketch A

In a row of highly loaded blades, a minor physical irregularity, or flow non-uniformity, can result in momentary overloading and separation. This separation, or blockage, will restrict the flow through the blade passage and will therefore divert the incoming streamlines. On one side of the blockage the streamline diversion will cause increased incidence, and on the other side decreased incidence. The blade passage subjected to increased incidence, the one on the left in Sketch A, will therefore become stalled, while the one already stalled will unstall. The stall "cell" will thus propagate from blade to blade around the compressor. The propagation speed of the cell is always lower than the rotational speed of the blades and, although the stall cell will move to the left relative to the rotor blades in this picture, both cell and blades move to the right in the absolute frame of reference. Although not originally intended this way, the above picture has been drawn so as to deliberately emphasise the point that a stall cell such as this need only affect the flow around a few blade passages. A cell like this would be called a disturbance of short length scale.

The second flow model is also found in the above paper by Emmons et al, but has recently been extended to multi-stage machines by Moore and Greitzer (1986) In this work modal perturbations are of primary interest and these, sometimes referred to as pre-stall waves,

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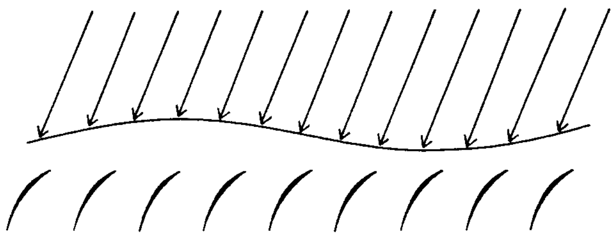
may be visualised as small sinusoidal velocity fluctuations which rotate about the annulus at a steady speed.



Sketch B

In Sketch B only a limited number of blades are shown but these are intended to represent a complete blade row. A velocity perturbation with a wave length equal to the circumferential length of the annulus would be referred to as a mode of order 1. Here the blades move to the right and so too does the perturbation, but always at a speed lower than the blade speed. In this view, such a perturbation would grow or decay depending on stability criteria related to the operating point of the compressor. If the compressor were to be throttled towards the stalling point the perturbation would grow in intensity, without any abrupt change in either amplitude or frequency, until a fully developed stall cell is formed. In other words the stall cell would grow smoothly out of the pre-stall perturbation. Initially the wave amplitude would be infinitely small and not discernible above the background noise, but as the instability point is approached growth of the wave would be rapid, and to the observer the compressor would appear to stall instantaneously.

In recent years the Moore and Greitzer model has received some direct experimental support from work done by McDougall et al (1990) and by Garnier et al (1990). McDougall was the first to show that modal perturbations actually exist and his measurements lead him to conclude that localized disturbances of the first type (Sketch A) have no part to play in the stall inception process. Garnier was also able to detect modal perturbations in low and high speed compressors and examined the build up of these perturbations prior to stall. His work did not, however, concentrate on the process by which the perturbations ultimately become stall cells. The present work also shows modal perturbations prior to stall, but by looking more closely at the cell formation process it has been found that modal perturbations are not the only path into stall. Disturbances of a shorter length scale (as in Sketch A) have now been identified and their effect on cell formation is evident in the majority of cases tested.

The work documented here focuses on the process of stall cell development and presents detailed measurements from both single stage and multi-stage machines. It also takes advantage of a compressor facility which incorporates devices by which the onset of stall can be delayed using active control techniques. Using this equipment the flow field can be perturbed so that different aspects of the transient behaviour in the compressor may be studied. The findings undoubtedly reveal a more complicated picture than the accepted picture of stall inception, and provide new details on short length scale disturbances which will have to be taken into consideration in future modelling of compressor stability. The results also help explain some of the difficulties encountered in recent active control experiments and underline the fact that methods for active control of stall will need to be more complex than originally thought.

EXPERIMENTAL FACILITY AND DATA PROCESSING

The stall inception measurements have been obtained from two low-speed compressors at the Whittle Laboratory, namely the Deverson rig and the C106 compressor. A detailed description of both machines is given by Li (1990) with some additional statistics included in an appendix to this paper. The Deverson rig is a large single stage low speed machine ideally suited to detailed measurements of flow in the blade passages (Fig. 1). As a stage it is intended to have axial absolute flow into the rotor and therefore has no inlet guide vanes. The tip clearances of the rotor blades can be varied from 0 - 3.0%, based on chord length, by using spacers under the blades at the hub. The

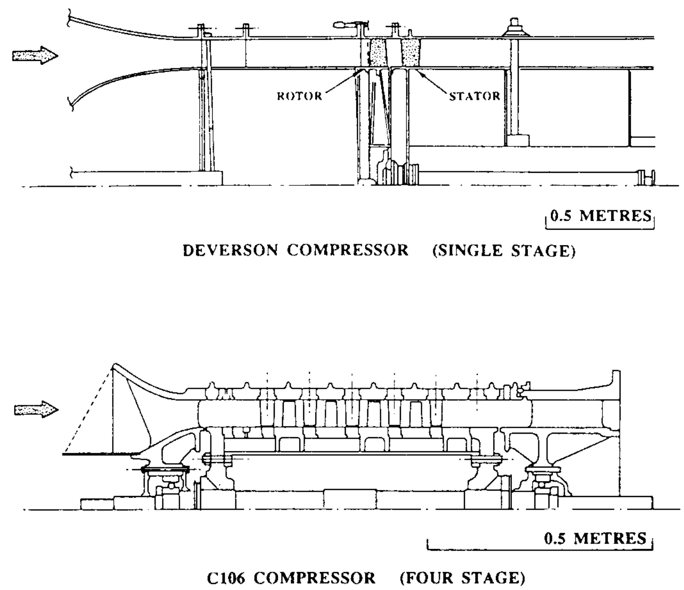


Fig. 1 Cross Sectional Views of the Deverson and C106 Compressors.

C106 compressor on the other hand is a multi-stage machine with four identical stages (Fig. 1). The blading has been designed to model a modern HP compressor and the stage loading is comparatively high.

The C106 was used for preliminary studies on the active control of stall and in the course of this work the compressor was fitted with an array of twelve fast acting air injection valves. These valves were equally spaced around the circumference of the machine between the IGVs and the first rotor and were designed to inject a small puff of air into the flow field near the rotor tips. The velocity of the injected air was about 1.5 times the free stream velocity and the amount of air from each valve was less than 0.1% of the compressor flow rate. (A fuller description of these valves is given by Day (1991)). The valves could be individually opened and closed and when linked to a computer could be instructed to inject any desired sequence of disturbances into the compressor.

Data acquisition procedures were essentially the same for both compressors. Hot-wire anemometers were used throughout, although wall static transducers were available to back up the velocity measurements. Where the probes were equally spaced around the circumference of the machine the number of probes varied between 4 and 8 with 6 being the preferred number. In the four stage machine the probes could be placed at almost any axial location but most of the time they were positioned just upstream of the first stage rotor; about half a chord from the rotor face. In the Deverson rig this spacing was reduced to about a quarter of a chord length ahead of the rotor face. This approach means that the probes were always in relatively undisturbed flow (as opposed to the turbulent flow behind the rotor), and therefore small changes of blockage or shifts of flow in the rotor passages could easily be detected. Moving the probes further upstream provides spatial damping from blade related potential effects, but also reduces the possibility of picking up small disturbances in their early stages.

Numerous processing techniques are available for enhancing various aspects of the transient hot wire data, but in most cases the examples used in this paper have been specifically chosen so that unprocessed time traces can be used for the sake of clarity. The time taken and the distance travelled around the circumference are easily obtained from these figures to give an indication of speed of rotation. Another useful method of looking at the data is to determine the amplitude and phase angle of circumferential perturbations by Fourier analysing the velocity distribution around the annulus obtained from the hot-wire probes. Repeated analysis at each sampling time step thus

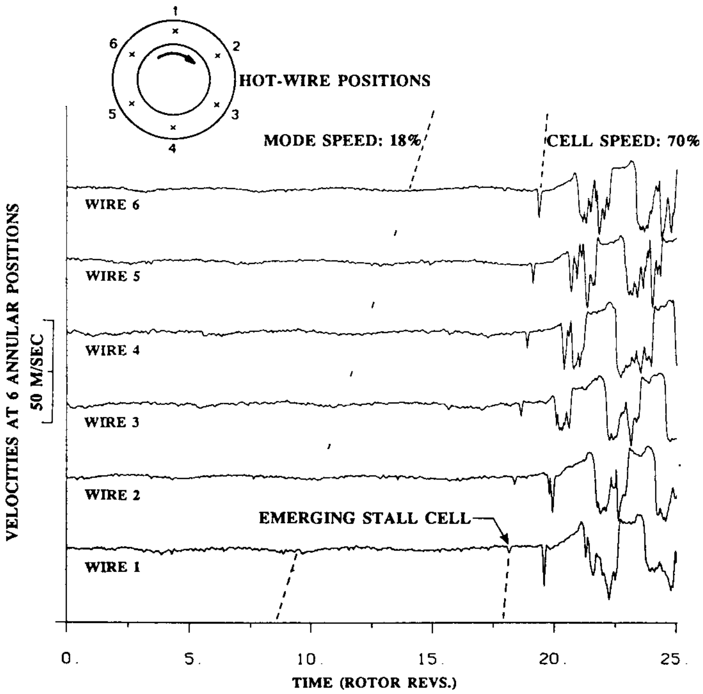


Fig. 2 Hot-wire measurements from the C106 Compressor (1.5% clearance) showing modal perturbations prior to the formation of a single stall cell. (Mean flow velocity just before stall: 28m/sec.)

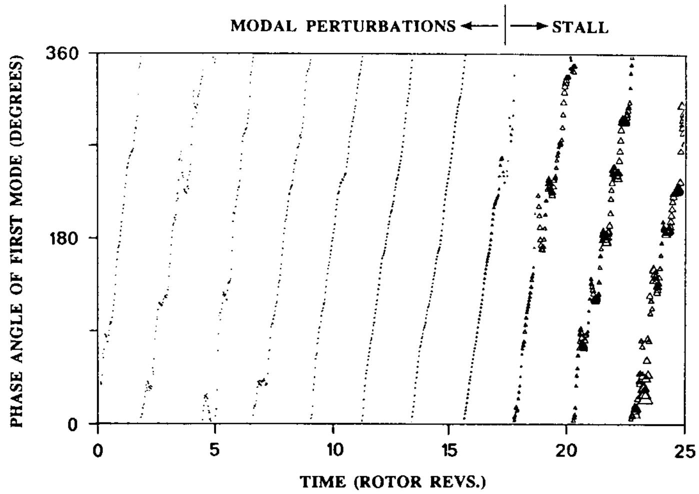


Fig. 3 Results from the Deverson Compressor (0.7% clearance) showing the phase angle versus time of a first order modal perturbation.

allows waves rotating in the annulus to be tracked as they move and develop with time. This technique was developed by Longley (1988) at the Whittle Laboratory who first plotted phase angle against time, varying the sizes of dots to indicate the amplitude of the disturbance. In the present work this technique has been used extensively to highlight the coherence of small perturbations which would otherwise have been masked by background noise.

EXPERIMENTAL RESULTS

The experimental results can be roughly divided into two parts, the first dealing with modal perturbations and the second with the formation of finite stall cells. The interaction between these two phenomena will be considered separately.

On a point of terminology, there has been a tendency in the past to refer to modal perturbations as 'pre-stall waves'. However, as the intention of this paper is to show that the modal perturbations and the formation of finite stall cells are not necessarily consecutive events, the term 'pre-stall wave' will not be used. In addition, the nature of the modal perturbations is so closely linked to the idea of aerodynamic resonance that the words "perturbation" and "oscillation" are sometimes interchanged depending on the emphasis required.

1. Modal Perturbations

Modal perturbations were first observed by McDougall et al (1990) and since then have been seen in other compressors by Garnier et al (1990). An example of these perturbations occurring in the C106 compressor is given in Fig. (2). Six hot wires just ahead of the first rotor positioned at mid blade span were used to obtain these results. From the beginning of the trace a wave like disturbance is present in the flow right up to the point where the compressor stalls at rotor revolution 16. By following a trough in the waves as it moves upward and to the right from one wire to the next it can be seen that the perturbation is travelling in the direction of rotation, and that the incoming flow field must look something like that shown in Sketch B above.

Another example of modal perturbations occurring before stall is given in Fig. 3. In this case the data, taken from the Deverson rig at 3.0% clearance, has been spatially decomposed using the Longley procedure and is plotted in terms of 1st mode phase angle versus time. The size of the dots represents the amplitude of the modal perturbation at each point in time and the grouping of the dots into periodic lines represents the rotation of a coherent modal wave. The unprocessed data from which this plot was derived is shown later in Fig. 10, and it can be seen from both these figures that the stall cell develops smoothly out of the modal wave. (This is not the case in the previous figure (Fig. 2) but this topic will be covered in greater detail later on.)

The growth of the modal perturbations prior to stall in the examples of Figs. 2 & 3 is suggestive of a damped oscillatory system becoming less damped as the compressor is pushed further and further towards the stall point. This is consistent with the Moore & Greitzer (1986) analysis which shows rapid growth of the oscillations at the peak of the total to static pressure rise characteristic.

The C106 test rig with its array of air injection valves offers a unique opportunity to explore the natural resonance of the compressor. To do this a looping sequence of valve openings was employed to give as near as possible a sinusoidal disturbance of constant amplitude to the incoming flow field. The valve sequence was computer generated and the amplitude of the forcing disturbance was kept constant by stabilising the air injection rate. The compressor was set running at a flow rate as low as possible without prematurely dropping into stall and the looping sequence was set in motion. The disturbance was started at a slow speed, about 10% of rotor speed, and slowly accelerated to about 50% of rotor speed. Five consecutive sweeps of this frequency range were executed so that an ensemble averaged result could be obtained. The results from these tests for a single hot wire positioned well away from the immediate influence of the valves is shown in Fig.4. The abscissa is labelled in percentage of rotor speed and shows that the peak amplitude of the induced disturbance occurs in a frequency range between 15 and 20%. This is in close agreement with the modal frequency of 18% measured from Fig.2. where the compressor went into stall naturally.

This experiment was backed up by a series of tests where measurements were taken around the compressor annulus immediately after a short sharp pulse was injected into the flow by one of the valves. The familiar modal disturbance was observed and the frequency was found to be the same as when the mode develops spontaneously. The amplitude of the mode decayed quite quickly after the pulse and was clearly linked to the compressor operating point. Further tests were therefore done to determine how the damping of the system decreases as the peak of the characteristic is approached.

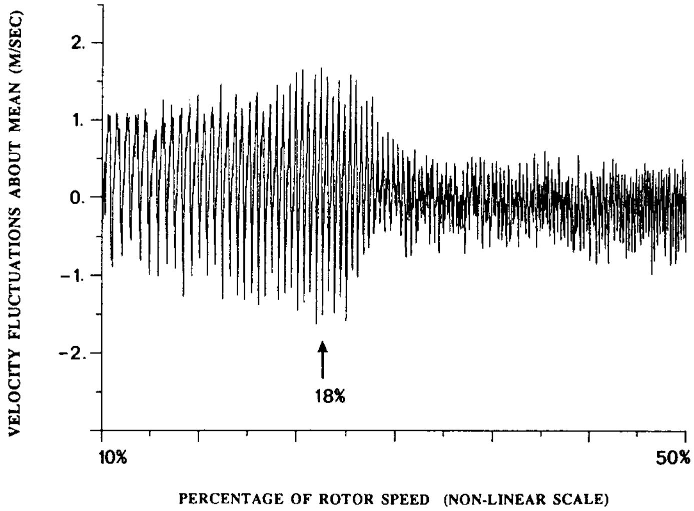


Fig. 4 Velocity fluctuations in the C106 Compressor (1.5% clearance) induced during unstalled operation by a rotating disturbance of constant amplitude and increasing rotational frequency.

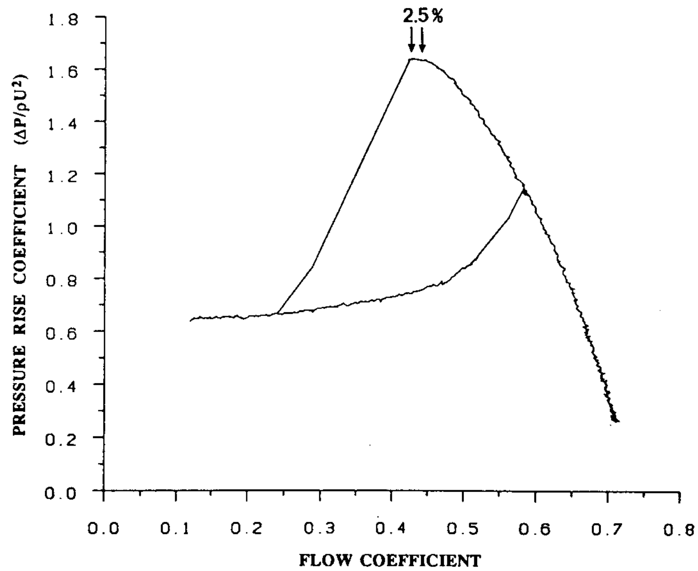


Fig. 6 Total to static characteristic of C106 Compressor showing how narrow the range is over which modal oscillations can be excited.

Accepting the natural frequency of the system to be about 18% of rotor speed, the valves were then used in a sinusoidal fashion at this frequency to excite a response in the compressor while operating at flow rates closer and closer to the stall point. The results are shown in Fig. 5. The amplitude of the excited waves increased almost linearly as the flow rate was reduced. A more interesting point, however, is the limited range over which any sign of excitation could be detected. Fig 6. shows this range on the measured total to static compressor characteristic. These experiments emphasis the limited range over which modal perturbations can be sustained, and this knowledge will prove useful later on in explaining why modal perturbations are observed prior to stall in some instances and not in others.

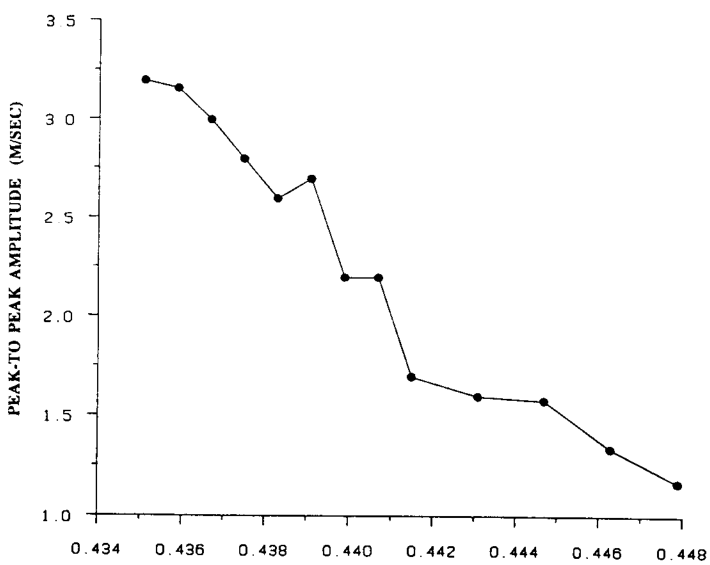
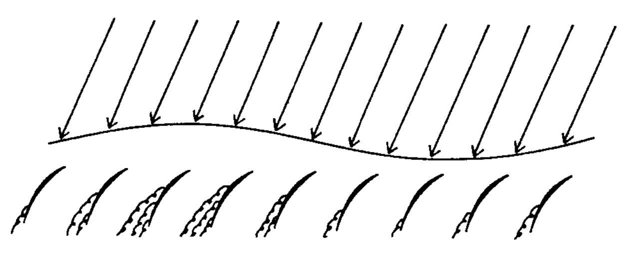


Fig.5 Amplitude of induced modal oscillations in the C106 Compressor at various throttle settings. (Frequency of forcing disturbance equal to the natural frequency of the compressor, ie 18% of rotor speed.)

During some of the testing on the 4 stage compressor the modal perturbations were of quite noticeable amplitude, even when the air injection valves were not in use. It was therefore reasonable to ask if these waves were not perhaps the manifestation of some other disturbance deeper in the compressor. Measurements were therefore taken with hot wires downstream of the third stator row, i.e. upstream of the last rotor. These probes revealed large periodic velocity fluctuations which appeared in perfect unison with the modal wave measured simultaneously at the front of the compressor. It was subsequently found that the probes happened to be in such a position relative to the upstream stator blades as to routinely catch the fluctuating wakes shed from these blades. Other probes used at the same time were well clear of the stator wakes and so showed no disturbance at all. The conclusion to be draw from these tests was that the rise and fall of through-flow velocity occurring with the rotation of the modal wave was causing a sympathetic thickening and thinning of the corner separations attached to each stator blade. The idea is illustrated in Sketch C. where the effect of the velocity perturbation on the wake size is clearly visible.



Sketch C

To demonstrate that this picture is in fact correct, 5 hot wires were spaced across one blade passage near the outer casing behind the third stage stator row with the first and fifth probes in repeating relative positions. A spanwise position near the outer casing was chosen for this test because the flow separation being examined is usually most marked in the secondary flow regions at the extremities of the blades. The results in Fig. 7 show that hot-wire 2, nearest the stator wake,

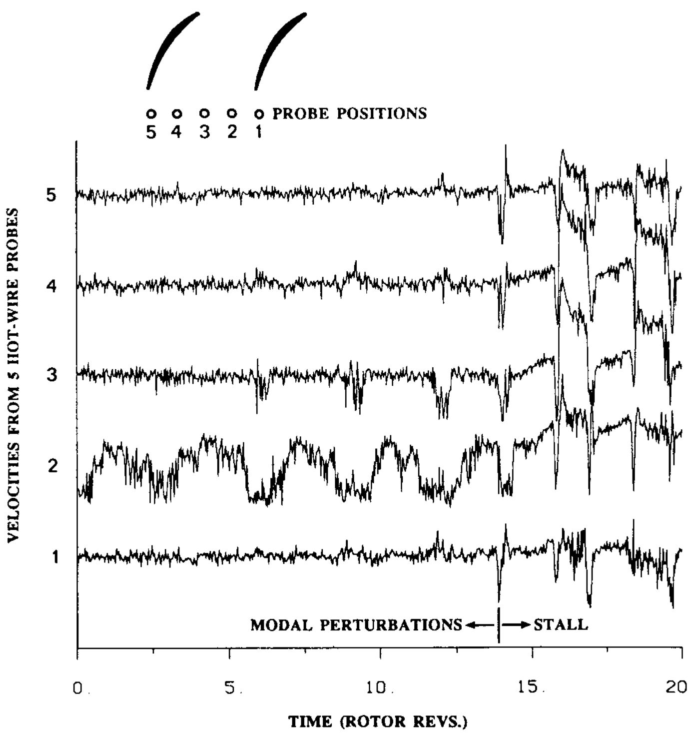


Fig. 7 Data from the C106 Compressor showing how the fluctuation in stator wake thickness affects the output from hot-wires in various positions relative to the upstream stator blades.

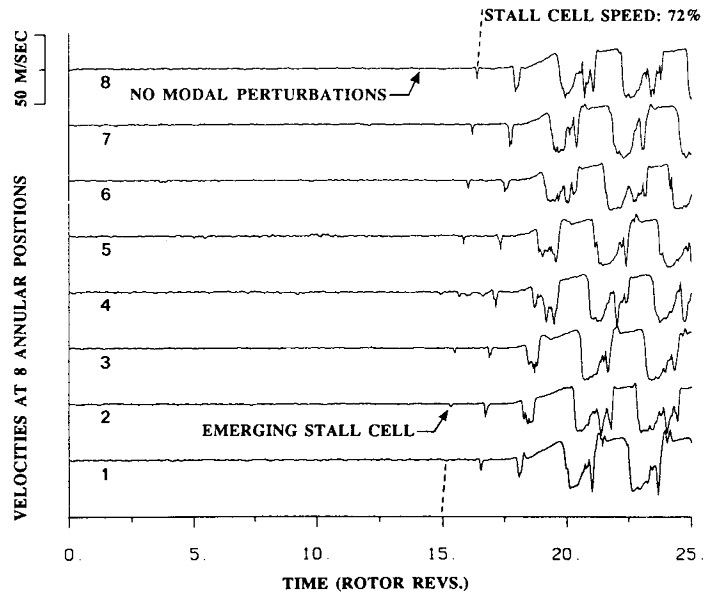


Fig. 8 Hot-wire measurements from the C106 Compressor(1.2% clearance) showing a stall cell emerging from a flow field in which no modal perturbations can be detected.

was indeed picking up the mode associated velocity fluctuations mentioned previously, while wires 1 and 5, which were well clear of the wakes, showed the least disturbance prior to stall. Once the compressor stalled of course the stall cell was picked up equally by all the probes.

It should be noted that corner separations, sometimes quite extreme ones, are a regular feature of the flow in a compressor operating near stall and it is therefore not surprising that the extent of the separation should be influenced by small changes in through-flow velocity. The fact that these separations exist does not imply that the machine itself is stalled nor does the presence of the modal perturbations mean that the compressor is irretrievably on its way into rotating stall. At any point prior to the formation of a finite stall cell, even when strong modal fluctuations are present, the machine can be returned to normal operation simply by backing off the exit throttle. The kind of fluctuating corner separations seen here are present throughout the whole machine and occur in both the rotor and stator blading but with the stators being most severely affected in this compressor, i.e. the C106.

2. The Formation of Finite Stall Cells

In order to discuss the formation of finite (ie not infinitesimal) stall cells, it is necessary to be clear about the distinction between a stall cell and a modal wave. A modal wave, as seen experimentally, is a "reversible" disturbance which can be made to come and go by slight changes in throttle setting and may be visible in the machine for as much as 200 rotor revolutions before stall (Garnier et al, (1990)). During this time the pressure rise across the compressor will remain nearly constant. A stall cell, on the other hand, is a comparatively "irreversible" disturbance and once formed will lead inexorably to a collapse of the pressure rise - usually within 4 to 6 rotor revolutions.

In the preceding discussion it was shown that modal perturbations may appear in the compressor as the stability line is approached. Alternatively a finite stall cell may also appear in the compressor, independently of the modal perturbations, and at about the same operating condition. It will be shown below that cell formation may precede the occurrence of modal perturbations; in which case the modes are never seen. Alternatively, cell formation may occur after the appearance of modal perturbations in which case some degree of coupling between these two phenomena, both rotating in the same direction, may occur.

In a somewhat artificial manner a distinction is drawn below between two types of stall cells which may appear in the compressor: those of short length scale and those of a longer length scale. Although the distinction is made in terms of the size of the cell when it first forms, the real difference between these two types of cells is the speed at which they rotate. The smaller cell will rotate more quickly, while a larger one will rotate more slowly. The importance of this distinction is discussed below.

a) Cells of short length scale. We start by considering an example from the C106 compressor, set at 1.2% tip clearance, in which a stall cell appears before there is any sign of a modal wave. Fig.8. shows the machine operating steadily, with no hint of modal perturbations prior to stall. Modes could not be detected using the Longley FFT procedure either. At approximately $t = 15$ in this figure, a small stall cell of limited extent is formed which rotates around the machine, growing slowly at first and then more quickly to become a fully developed stall cell. Initially the cell starts out moving at about 70% of rotor speed but slows down to 38% when the cell is fully formed. This pattern of a small localised stall cell rotating fast and slowing as it grows is a dominant feature of the measurements from the C106 compressor and is indicative of a stall inception pattern which is similar to that suggested in Sketch A.

Another example of cell formation occurring without the presence of modal perturbations comes from the Deverson rig as shown in Fig.9. It will be recalled that this compressor exhibited modal waves when the tip clearance was set at 3.0% (Figs. 3&10). Now with the tip clearance at 1.2% the picture is quite different. In Fig. 9 the incoming flow is steady and detailed analysis confirms that there is a complete absence of any modal oscillation prior to the formation of the stall cell. Once again the cell is quite localised when it starts out, only 2 or 3 blade passages wide, and its speed of rotation is close to 70%. The tip clearance for this build is 1.2%, as it was in the C106 in Fig. 8, but for want of hard evidence the similarity of clearance may just be a coincidence.

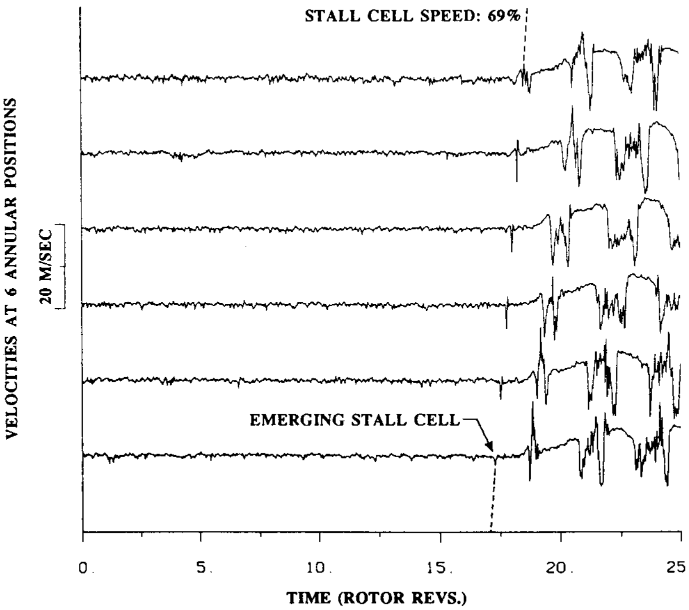


Fig. 9 Hot-wire measurements from the Deverson Compressor (1.2% clearance) showing a stall cell emerging from a flow field in which no modal perturbations can be detected. (Mean flow velocity just before stall : 16m/sec)

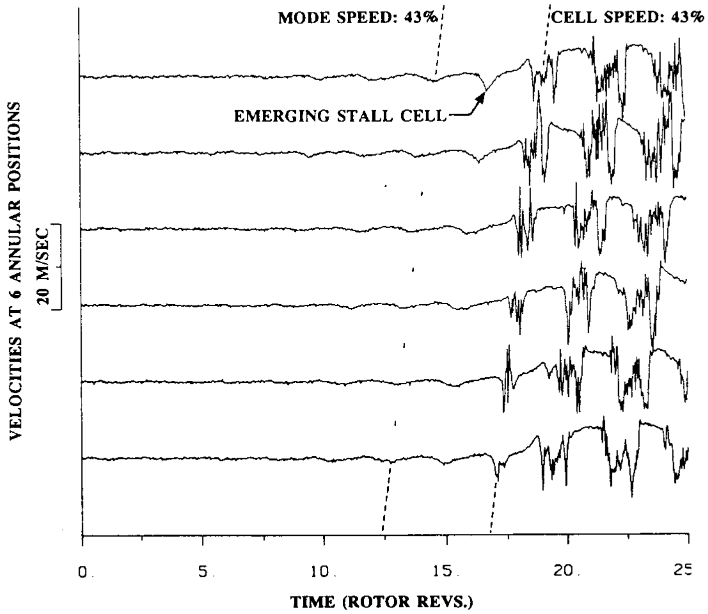


Fig. 10 Hot-wire measurements from the Deverson Compressor (3.0% clearance) showing a stall cell emerging smoothly out of a flow field containing a modal perturbation.

If the tip clearance in the C106 compressor is increased slightly to 1.5% by changing the outer casing ring over the first rotor, modal perturbations appear before stall. A clear example of this was given earlier in Fig.2. Despite the presence of these perturbations the stall cell formation pattern is just as it was in the previous two examples, i.e. a small localised cell, appearing suddenly and rotating fast. The lack of influence of the modal perturbations on the formation of the cell is emphasized by the fact that the modal speed is 18% whereas the stall cell starts out at nearly 70% of rotor speed.

In Fig.2 the stall cell is initially very localised and has a small circumferential length scale compared to that of the preceding modal wave, i.e. $L_{cell} \sim .05\ L_{mode}$. The coupling between the two phenomena would therefore be expected to be small, as is in fact the case. It has been shown by repeated measurements, however, that the modal wave does affect the formation of the stall cell in that the cell usually begins near the trough of the wave where the through-flow velocity is lowest. The stall cell rotates much faster, and is a more vigorous disturbance, than the modal wave and therefore the wave only remains coherent for a very short time after the stall cell appears. Once the stall cell grows in size the modal wave is naturally obliterated.

b) Cells of longer length scale. The above discussion of finite cell development concentrates on what happens when the cell is initially of small length scale, i.e. the disturbance is localised to just a few blade passages. If, on the other hand, the stall cell is larger in circumferential extent when it first forms it tends to rotate more slowly and coupling between the modal wave and the stall cell will then be more effective.

An example of effective coupling is taken from the Deverson rig at 3.0% tip clearance. In this configuration the compressor produces a lower pressure rise for the same flow coefficient than before and the compressor stalls more gently with a much smaller hysteresis loop. The results of the stall inception measurements are shown in Fig.10 where a modal perturbation appears prior to stall. When the stall cell forms in this case, it is larger in circumferential extent and rotates more slowly. The length scales of the mode and the cell are more comparable here and the cell now appears to grow progressively out of the modal wave. The abrupt change in speed seen previously between the modal wave and the emerging stall cell does not occur in this case. This type of stall inception pattern supports the Moore & Greitzer model shown in Sketch B and is in fact the case studied by McDougall et al (1990).

c) Cells of both length scales. An unusual example illustrating the difference in rotational speeds of cells of short and long length scales is given in Fig. 11. This data was obtained from the Deverson compressor at 0.7% tip clearance when the compressor sometimes stalled in an uncertain manner. Here a short length scale cell is joined soon after inception by a larger cell which rotates much more slowly. In this figure the smaller cell has been coloured in black to make it

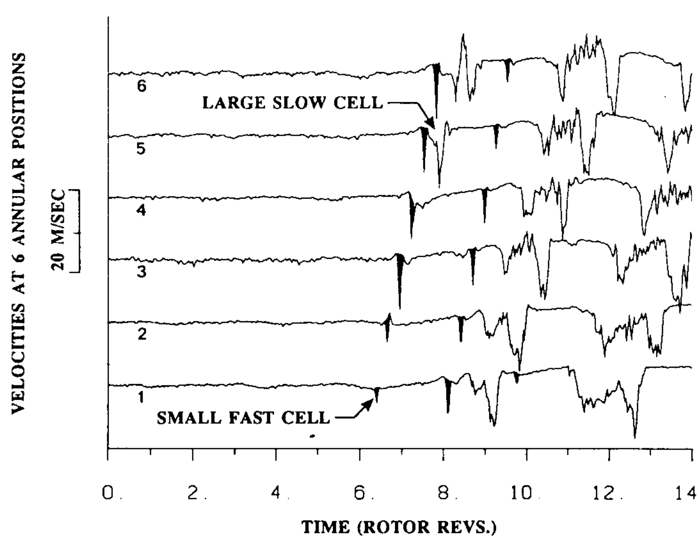


Fig. 11 Hot-wire measurements from the Deverson Compressor (0.7% clearance) emphasizing the difference in speed between small and large stall cells.

easier to follow its progress around the the machine. Because of its greater speed the small cell quickly catches up with the larger one and after two complete revolutions it runs into the back of the slower moving disturbance. (On closer inspection it can be seen that after one revolution the small cell begins to decrease in size. This is because the part of the flow field through which it is moving is beginning to operate at a higher through flow velocity due to the blockage caused by the larger cell.)

SUPPORTING MEASUREMENTS

The examples given above are intended to show that modal perturbations can influence the process of cell formation but that these perturbations themselves are not a necessary prerequisite for the formation of stall cells. A number of other experimental observations support this point and these are listed below.

1) It was shown previously how the range over which modal oscillations appear in a compressor is very limited and is restricted to the peak of the characteristic. Since many compressors stall while the characteristic is still rising it is not surprising that the first sign of a disturbance might be the stall cell itself and that the modal oscillations are therefore never seen. This is often the case in the C106 compressor. In this compressor, however, it is possible to inject small quantities of air directly at the tips of the first rotor (in a steady manner) and this has the effect of delaying the formation of the stall cells while creating a nearly horizontal extension to the compressor characteristic, Day (1991). Any modal oscillation in the compressor can therefore be greatly amplified simply by delaying cell formation and so allowing the compressor to reach an operating point further to the left on the characteristic. The results of such a test are summarised in Fig. 12 where the output from one, rather than six, hot-wires is used to illustrate the difference. The upper trace shows the compressor going into stall naturally with only a slight hint of a modal wave before stall. The lower trace shows a much clearer modal wave when the characteristic is extended by just 1.5%.

It is worth pointing out again that when the stall cell is small and sharply defined, the coupling between it and the modal perturbation is very weak. The patterns of cell development in the upper and lower traces in Fig.12 are almost identical despite the fact that a much stronger modal perturbation is present in the lower trace.

A further test along these lines was done using one air injection point only to trip the machine into stall prematurely. By swivelling the chosen injection valve to point in an unfavourable direction, a short sharp puff, effective for about 2.5 milliseconds, is sufficient to trigger the formation of a small finite stall cell of the type seen above and this causes the compressor to go straight into stall. This premature stalling can be achieved at a flow rate about 5% greater than that at the natural stall point. This experiment emphasises the range over which stalling can occur and implies that, under natural conditions, some minor physical or aerodynamic irregularity could easily stall the machine before the point on the characteristic is reached at which modal oscillations come into play. The data from this last experiment also shows that the action of the valve is the immediate cause of cell formation; the valve does not first create a modal oscillation which then turns into a stall cell.

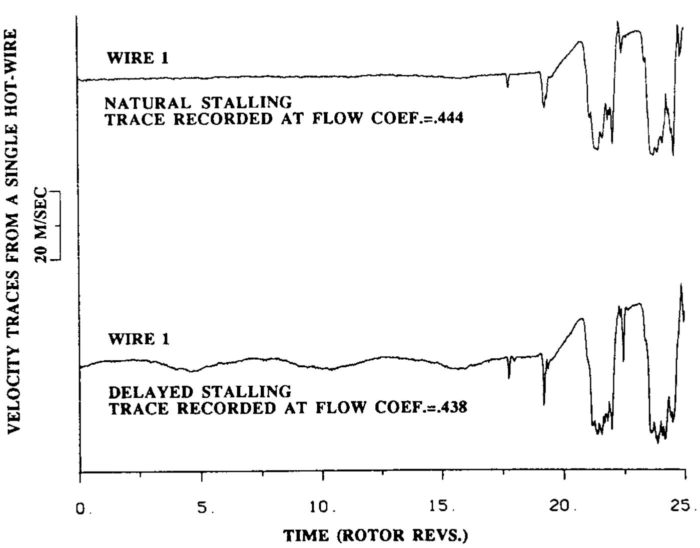


Fig. 12 Hot-wire traces recorded in the C106 Compressor (1.5% clearance). The upper trace shows natural stalling while the lower trace shows stalling at a flow coefficient 1.5% below normal.

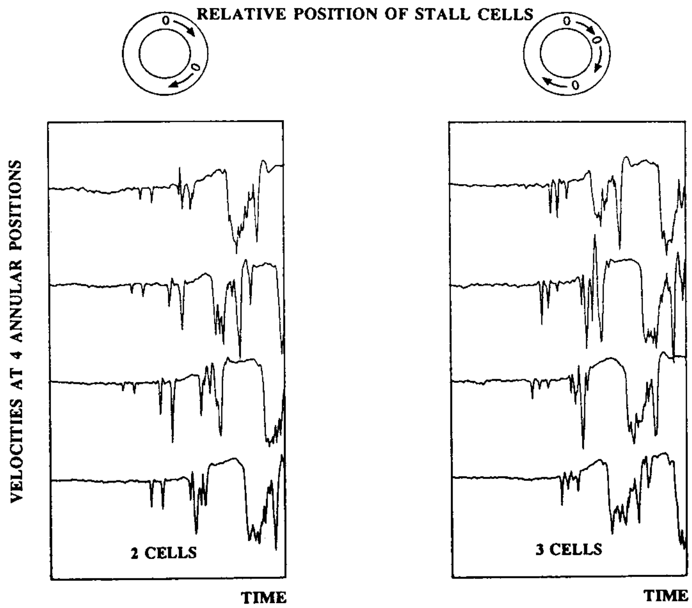


Fig. 13 Examples of the uneven distributed of stall cells around the annulus in the C106 Compressor (1.2% clearance).

2) The small sharply defined stall cells of the type which appear in the Deverson and C106 compressors have a very short circumferential length scale. The disturbing influence of these cells is restricted to just a few blade passages and therefore any number of cells could form in the compressor, and these could be randomly distributed around the annulus. Fig.13 from the C106 compressor shows some examples of short length scale cells which form in just this way. The random distribution of the cells would be far less likely to occur if organised modal perturbations were a necessary prerequisite for cell formation.

3) In many of the experiments the short wavelength stall cells appear to originate at precisely the same circumferential position in the annulus each time the machine is stalled. This phenomenon has been observed in the C106 compressor and in the Deverson rig (1.2% clearance), but in the Deverson rig particularly, the orientation of the rotor is also involved in the timing of cell formation. A series of 21 consecutive tests were conducted in which six hot wires were used to determine the starting location of the stall cell, and at the same time a once per revolution signal was used to track the orientation of the rotor. In all the tests the stall cell clearly originated near the same spot on the casing and,furthermore, the rotor was always in the same relative position when the cell started. In other words the formation of the stall cell was delayed until a particular part of the rotor was opposite a particular part of the casing. The results of the rotor position measurements are shown in Fig.14 where relatively little scatter in rotor orientation is observed at the time of cell formation. No measurable eccentricity could be found in the compressor when these

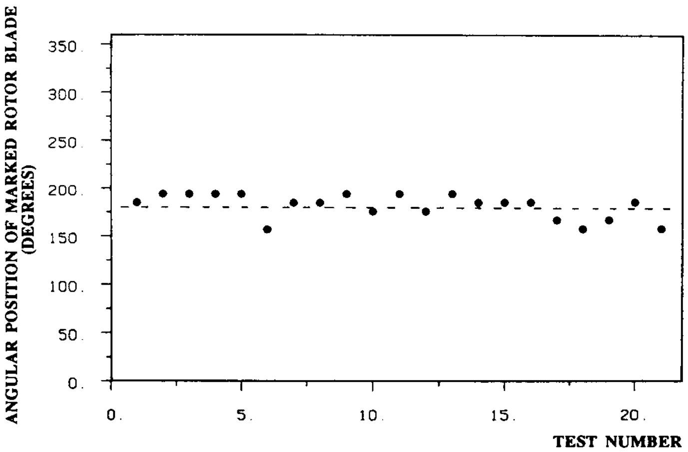


Fig. 14 Series of repeated tests on the Deverson Compressor (1.2% clearance) showing the angular position of the rotor at the time of stall cell formation.

tests were done. Again the Sketch A picture of cell formation is most suitable here where, in a highly loaded situation, an imperceptible irregularity would be all that is necessary to trigger the formation of a localised cell.

Some tests were done by McDougall et al(1990) in the same compressor to try and fix a point on the rotor where the stall cell would begin. Substantial eccentricity (up to 8 re-staggered blades) had to be built into the rotor before rotor position played a significant part in the timing of cell formation. The difference between their tests and those reported here is that their tests were done at 3.0% clearance. At 3.0% clearance modal oscillations of large length scale regulate cell formation (see Fig. 10), whereas at 1.2% clearance the cell length scale is much shorter and localised irregularities are likely to be of greater consequence.

DISCUSSIONS

The natural occurrence of modal perturbations in a compressor operating near the stability limit has been demonstrated, as has the narrowness of the range over which the perturbations will appear. Finite stall cells of independent origin have also been shown to appear in the compressor under much the same operating conditions. Coupling between the two types of disturbance may occur if the modal perturbation becomes established prior to the formation of the stall cell. If the stall cell forms first, however, the flow field will quickly be disrupted and the modal perturbation will not be seen. The measurements therefore suggest that modal perturbations and the formation of finite stall cells are two separate phenomena.

A modal oscillation is essentially a perturbation superimposed on an underlying axisymmetric flow field. The individual blades in the compressor may all have partly separated flow and the size of the separated region may grow and shrink in sympathy with the oscillation, sometimes even giving the impression of propagation, but still the flow field is essentially axisymmetric . A stall cell, on the other hand, represents a definite break in the symmetry of the flow field. It is not practical to define a stall cell as a disturbance of this or that flow structure, but it is unequivocal to define it as a disturbance which puts an end to the axisymmetric nature of the flow; the crack which stops the bell from ringing.

Two types of stall cells have been observed and these have been classified according to the size of the cell when it first appears i.e. short or long length scale cells. In the past, part span and full span cells have been identified and it might be thought that this classification would perhaps explain the differences observed here. Extensive testing in the C106 and the Deverson compressor has, however, shown that in all cases the first sign of real disruption of the flow field always occurs near the outer casing. In some instances the disturbance spreads to the hub more quickly than in others but in general it may be said that all the stall cells examined here start out as part span and then become full span. Some additional information on this topic is given in a discussion by the author to the paper by McDougall et al, (1990). (It is also interesting to note that in the four stage compressor the stall cell always originates near the tips of the first rotor.)

At the present time it is not possible to say which physical features of the compressor are most important in terms of deciding which stall inception scenario will apply. Tip clearance has been found to play an important part, but this is the only parameter which has been varied in any significant way so far and its influence is not clear. Tip clearance is known to affect conditions in the blade passages and therefore its primary effect on stall cell behaviour may well be in this part of the flow field. Some recent tip clearance experiments have produced conflicting results and therefore the references to tip clearance in this paper should be treated as build labels rather than parametric details.

By way of a summary, the stall inception details from the various compressors for which information is available have been tabulated below.

Compressor Build	Modal Speed (%)	Cell Starting Speed (%)	Cell Final Speed (%)
Deverson (3.0%)	43	43	36
Deverson (1.2%)	-	69	36
Deverson (0.7%)	44	65	38
C106 (1.2%)	-	72	38
C106 (1.5%)	18	70	40
Cranfield (4 stage)	-	67	49
Rolls Royce (8 stage)	-	66	48
MIT (build JP)	-	45	22
MIT (build VG)	35	35	38

It can be seen that in only the first and last cases, where the mode and the cell have equal speeds, does a relatively smooth transition take place from the modal perturbation stage right through to the fully developed stall cell. In all the remaining cases, whether modal perturbations are present or not, the stall cells always starts out going relatively fast and only then slow down. Further than this the data as a whole underlines the complexity of the problem and highlights the importance of short length scale disturbances in the stalling process.

CONCLUSIONS

1. From the experimental results it would appear that modal oscillations and the formation of finite stall cells are separate, and physically different, events. A modal oscillation is a disturbance of circumferential proportions superimposed on a fundamentally axi-symmetric flow field. An emerging stall cell on the other hand is a localised disturbance which represents the start of the process by which the symmetry of the flow field is destroyed.

2. The occurrence of modal oscillations and the formation of finite stall cells are not necessarily consecutive events. Both types of disturbance develop near the limit of the pressure rise characteristic, but either one might be the first to appear.

3. If a finite stall cell is the first to appear in the compressor, the symmetry of the flow field is rapidly destroyed and modal oscillations do not develop. If a modal oscillation develops prior to the formation of a finite stall cell, the oscillation may influence the stalling process to a greater or lesser extent depending on the size of the cell when it forms.

4. It would appear that stall cell formation can be divided in to two different patterns. In the one case the stall cell is restricted in circumferential extent, and the initial speed of rotation is high. In the second case the cell is larger in circumferential extent, originates less abruptly and rotates at a speed more like the final stall cell speed. (The separation of cells into categories of small or large is probably artificial, and a continuous spectrum of sizes is more than likely possible.)

5. When cell formation occurs in an already established modal flow field, some measure of coupling between these two phenomena may occur. If the cell is initially of small length scale its formation will not be affected by the larger length scale modal perturbation. In these circumstances the only influence of the modal waves is to provide a localised dip in the through-flow velocity where the stall cells are most

likely to originate. When the stall cell formation is larger in circumferential extent the stall cell will appear in phase with the modal wave and will develop smoothly without an abrupt change in amplitude or speed.

6. The data seen so far suggests that it is the initial emergence, and subsequent spreading, of a short length scale disturbance that is the most commonly followed route into fully developed stall.

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APPENDIX

Table 1. Deverson Single Stage Compressor. Mid-Height Blading Parameters and Other Details

	Rotor		Stator
Solidity	1.31		0.95
Aspect Ratio	1.37		1.34
Chord (mm)	111.0		114.0
Stagger (deg.)	47.9		14.3
Camber (deg.)	26.5		42.9
No. of aerofoils	51		36
No. of IGVs		0	
Axial Spacing (mm)		50.0	
Tip diameter (mm)		1524	
Hub/Tip ratio		0.8	
Speed of Rot. (rpm)		500	
Reynolds Number		3.1×10^5	

Table 2. C106 4-Stage Compressor. Mid-Height Blading Parameters and Other Details

	Rotor		Stator
Solidity	1.47		1.56
Aspect Ratio	1.75		1.75
Chord (mm)	35.5		36.0
Stagger (deg.)	44.2		23.2
Camber (deg.)	20.0		40.6
No. of aerofoils	58		60
No. of IGVs		60	
Axial Spacing (mm)		13.0	
Tip diameter (mm)		508	
Hub/Tip ratio		0.75	
Speed of Rot. (rpm)		3000	
Reynolds Number		1.7×10^5	