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# Stall Inception in Axial Compressors

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## ABSTRACT

Detailed measurements have been made of the transient stalling process in an axial compressor stage. The stage is of high hub-casing ratio and stall is initiated in the rotor. If the rotor tip clearance is small stall inception occurs at the hub, but at clearances typical for a multistage compressor the inception is at the tip. The crucial quantity in both cases is the blockage caused by the endwall boundary layer. Prior to stall disturbances rotate around the inlet flow in sympathy with rotating variations in the endwall blockage; these can persist for some time prior to stall, rising and falling in amplitude before the final increase which occurs as the compressor stalls.

## NOMENCLATURE

$p_2$ downstream static pressure

$p_{o1}$ upstream stagnation pressure

$U_m$ mean blade speed

$V_x$ axial velocity

$\phi$ flow coefficient $V_x/U_m$

$\psi_{TS}$ total-to-static pressure coefficient

$\omega$ rotational frequency

## INTRODUCTION

One of the limiting aspects of axial compressor behaviour is the breakdown of the flow into either rotating stall or into surge. In either case the unstalled compressor becomes unstable as the flow is reduced and as a result of stall the flow pattern is altered into one of these forms. (The word stall will be used here to describe the breakdown process following instability, while the stable operating process which can occur after the machine has stalled will be termed rotating stall.) Rotating stall and surge are alternative outcomes of instability of the same compressor, as was conclusively demonstrated by Greitzer(1976); it is not just the compressor but the parameters of the entire compression system which determines which will occur. More recently Moore and Greitzer(1986) and Greitzer and Moore(1986) have calculated the breakdown into both types of flow but only when the variation of pressure rise with flow rate is prescribed for the compressor, in other words it does not address the nature of the flow in the compressor itself. What this paper is about is the fluid mechanic processes which occur during stall.

One of the clearest descriptions of the conditions for a compressor to become unstable is given by Stenning(1980). The treatment is two-dimensional so it is strictly only valid for compressors with high ratios of hub to casing diameter. It also assumes that the flow is axially symmetric and incompressible, the latter being reasonable for most subsonic stages since it has been shown that the effects of Mach number are small for well matched compressors. The analysis makes no assumption about the behaviour of the compressor or the number of stages except that there can be no circumferential flow inside the machine - a reasonable assumption for most compressors which have small axial gaps between rotors and stators. The prediction of this simple analysis is that the compressor will be unstable when

$$\partial(p_2-p_{01})/\partial V_x > 0$$

where $p_2$ is the static pressure at the outlet face of the compressor, $p_{01}$ is the inlet stagnation pressure and $V_x$ is the axial velocity. Although the assumptions of the analysis are all reasonable the common experience is that most compressors stall when $\partial(p_2-p_{01})/\partial V_x$ is still significantly less that zero, i.e on the "stable" part of the operating characteristic when $p_2-p_{01}$ rises as flow rate is reduced. In practical terms this means that the compressor is stalling at flows larger than would seem necessary and that the operating range would appear to be unnecessarily reduced. The work described in this paper was an attempt to establish why this is so.

The analysis of compressor stability has usually avoided including any consideration of the compressor aerodynamics, choosing to treat the pressure rise-flow rate variation as an input

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parameter. However in describing the universal tendency of a stall cell to rotate Emmons et al(1955) did postulate a mechanism. They envisaged the cell to cause the separation of the flow from the suction surface of one blade; the blockage caused by this affects the upstream flow so that on one side of the cell the incidence is reduced and on the other it is increased. This succeeds in producing a disturbance which moves in the correct sense. In general rotation will occur when loss or pressure rise vary with incidence.

At the outset of this research there was an explanation for the stall occurring when $\partial(p_2-p_{01})/\partial V_x < 0$ which was to be investigated. The idea for this was that each blade passage was

Rotor blade row

| Radius (m) | camber (deg) | stagger (deg) | solidity |
|---|---|---|---|
| 0.610 | 40.3 | 38.8 | 1.48 |
| 0.686 | 26.5 | 47.9 | 1.31 |
| 0.762 | 18.5 | 54.4 | 1.18 |

51 blades, chord=111 mm.

Stator blade row

| Radius (m) | camber (deg) | design stagger (deg) | unloaded stagger (deg) | solidity |
|---|---|---|---|---|
| 0.610 | 44.4 | 16.6 | 26.8 | 1.07 |
| 0.686 | 42.9 | 14.3 | 24.5 | 0.95 |
| 0.762 | 42.9 | 12.7 | 22.9 | 0.86 |

36 blades, chord=114 mm.
Rotor and stator both C4 profile on circular-arc camber line, thickness-chord ratio =0.10

TABLE 1
Compressor stage geometry

similar to a diffuser and diffusers are known to achieve peak pressure recovery when they have flow which is transiently separated. It was a conjecture that the compressors operate in this way near stall and since the separations are stochastic the instability would occur when sufficient random blade separations occurred simultaneously. This explanation looked back to the Emmons et al model for the stall cell, putting attention on blade separation. As will be seen in the present paper it was an erroneous view and gave insufficient weight to other evidence which already existed about the importance of endwall boundary layers and tip clearance. It has been widely recognised that increasing tip clearance can move the stall inception point to significantly higher flow rates, for example Smith(1969) and Freeman(1985).

Two different types of measurement are reported here. One set looks at the flow field outside the rotor passage, mainly upstream. The other was made in or at the exit from the blade passages. The interpretation of the in-passage measurements take advantage of the detailed measurements of the time-mean flow in the same compressor reported by McDougall(1989). (The subject of that paper and the present one, together with the facilities and techniques, are described in greater detail by McDougall,1988.) McDougall(1989) showed that changing the rotor tip clearance from 1.2% to 3% of the tip chord raised the flow coefficient at stall inception from about 0.37 up to 0.40 as well as reducing the pressure rise. The overall pressure rise and stall inception point were essentially identical for tip clearances of 0.5% and 1.2%. McDougall also showed that there could be significant regions of separated flow in either the rotor or stator passages without stall occurring. All the significant separated regions where three-dimensional in character, originating near the hub or casing wall.

Because of the change in ideas as the work progressed it is more straightforward to drop any pretence of chronological order in this work but instead to try to give a clear picture of the real processes at work.

## FACILITIES AND TECHNIQUES

The present experiments were performed on a single stage, low-speed compressor with a hub-casing ratio of 0.8. A summary of the compressor geometry is given in Table 1 and details of the aerodynamic behaviour are described by McDougall(1989). The stage was one for which the absolute velocity into the rotor is axial and as a result the reaction was high with about 80% of the pressure rise at design point being across the rotor. To remove any risk that stall was being initiated in the stator the stator blades were staggered closed 10° from their design value so there was still less pressure rise across them. (Later tests showed identical stall behaviour with the stators at their correct design stagger and this extra unloading was unnecessary.) The overall pressure rise-flow rate characteristic is shown in Fig.1 for several ratios of tip-clearance to blade chord.

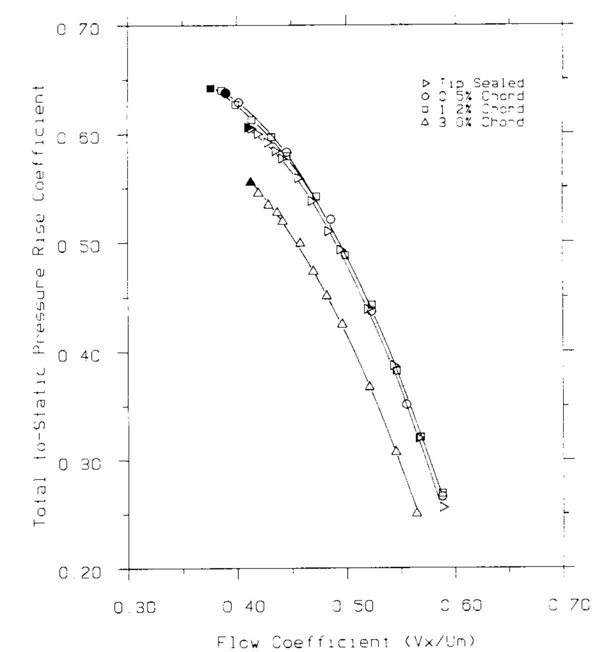


FIG. 1 EFFECT OF VARIATIONS IN ROTOR TIP CLEARANCE (Solid Symbols - Stall Point)

A traverse gear was mounted on the rotor so that the flow could be surveyed with a hot-wire or Kiel probe. In addition up to seven sub-miniature hot wires with probe stem diameter of 1 mm could be mounted on the rotor blades. Whenever hot-wires were used they were aligned to be normal to the expected local mean flow. All the unsteady data was logged by a computer. Stall is inherently stochastic and the instant when this would occur could not be predicted. A logging system was adopted whereby the data was logged continually with the computer memory being overwritten. When a stall was detected the contents of the memory were frozen so that events prior to the event could be recovered. In this way a high data sampling rate could be used with modest computer memory.

For these tests stall was initiated in one of two ways. In one case the throttle was closed to a point very near to stall and the compressor was left running until stall was initiated "naturally": in fact stall was effected by inlet distortion from the room. In the other case, which was used for the ensemble averages, the compresor was brought to a point very close to stall and stall was initiated by placing a hand over the outlet from the throttle, this being just enough to bring more or less immediate stalling.

With the data logger it was possible to collect very large bodies of data but harder to present it in a compact and understandable way. One approach is illustrated in Fig.2 where a cut-off threshold is specified and marks made on a bar chart when this was crossed. This simple technique has been extensively used because it allowed the most important effect to be displayed from several wires simultaneously.

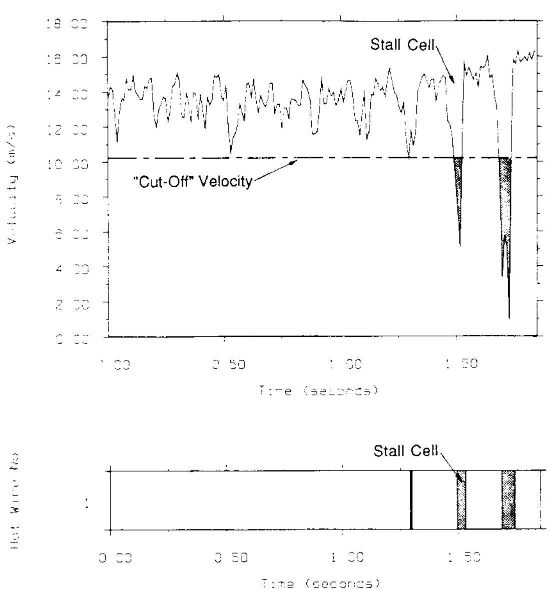


FIG. 2 RAW VELOCITY-TIME TRACE AND BAR CHART EQUIVALENT

## THE EFFECT OF TIP CLEARANCE ON STALL

The aerodynamic background to the differences illustrated by the overall pressure rise-flow rate characteristics, shown here as Fig.1, is discussed by McDougall(1989). The performance was very similar for tip clearances equal to 0.5 and 1.2% of chord, but that at 3% of chord the behaviour was strikingly different. Not only was the pressure rise lower but stall occurred at a higher flow coefficient with the larger clearance. McDougall also showed contours of axial velocity at the rotor exit for flow coefficients near to stall. With the smaller clearances there was a build up of low velocity flow from the hub over half the annulus, with a small pocket of reverse flow near the suction surface not far out from the hub. With the 3% clearance the boundary layer was thin near the hub but the casing boundary layer was thicker. The blockage produced by the clearance flow has reduced the blade pressure rise and as a result reduced the blockage at the hub.

For the build with 0.5% clearance Fig. 3 shows velocity traces measured with hot wires mounted near the trailing edge on the suction surface of a rotor blade. At the time labelled A the compressor is stalled and from the point of view of inception the flow is no longer of interest. It is very clear from this that the periodic disturbance leading to stall can be seen much earlier near the hub. In

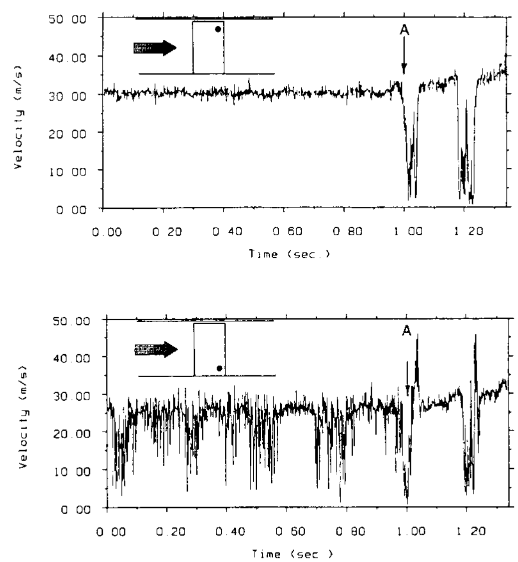


FIG. 3 VELOCITY-TIME TRACES RECORDED ON THE ROTOR DURING STALL INCEPTION AT THE HUB

bar-chart form Fig.4 shows a similar test with 1.2% clearance. This time each case compares an on-blade hot-wire with a hot-wire upstream of the blade near the casing. The hot-wire on the blade near the hub again shows up the effect long before it is visible near the tip on the blade. (It was found for all tip clearances that with the hot-wire in the correct spanwise position near the exit from the rotor the inception could be detected much earlier than upstream or downstream.)

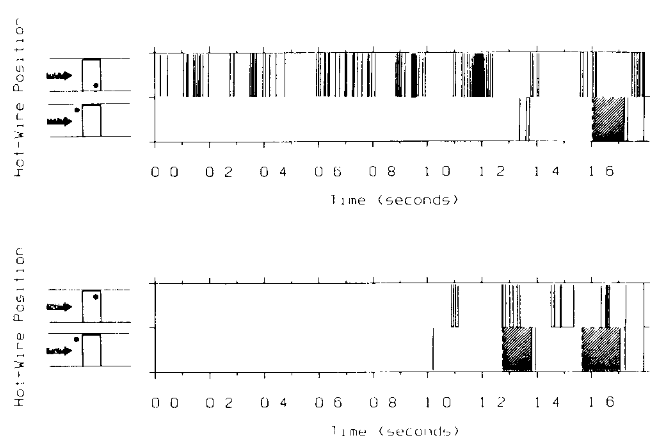


FIG. 4 VELOCITY MEASUREMENTS DURING HUB STALL SHOWN IN BAR CHART FORM (Rotor tip clearance 1.2% chord).

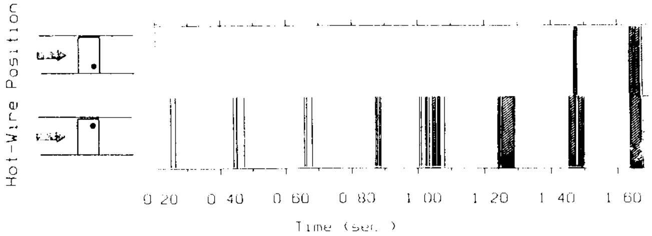

FIG. 5 VELOCITY MEASUREMENTS DURING TIP STALL SHOWN IN BAR CHART FORM (Rotor tip clearance 3.0% chord).

Figure 5 compares on-blade measurements with a clearance of 3% of chord. This time the behaviour is quite different with the hot-wire near the tip showing the incipient stall long before that at the hub. The fluid dynamic processes seem to be quite different with large and small tip clearance, although blockage is the critical quantity in each case. The precise value of clearance at which the change takes place must depend on many aspects of the design, not least the hub-casing radius ratio. Stall is normally initiated at the tips of rotors, see for example Freeman(1985), and 3% is believed to be a clearance likely to be encountered in multistage compressors. Most of the tests were therefore conducted with this geometry.

## THE FLOWFIELD UPSTREAM OF THE STAGE

Hot wires were installed at a number of positions upstream of the rotor aligned normal to the axial velocity. As will be shown below this was not the best place to spot the first signs of stall, although it was the only option open to Jackson(1987 ). The region ahead of the rotor does have advantages, compared with downstream, amongst which is its comparative freedom from turbulence and random unsteadiness. Spectral analysis had shown evidence of a spectral peak at about half rotor frequency, but this was not very clear. A better result came from a cross-correlation of signals from two wires 0.6 mean radii upstream of the rotor, set 90° of the circumference apart. The cross-correlation, which used only the unsteady part of the signals, was performed in the standard manner. The cross-correlation was not normalised so that an estimate for the disurbance magnitude can be obtained. The result is shown in Fig.6. For a flow coefficient well away from stall there is only a small cross-correlation with a time delay appropriate to the speed of the rotor: this then denotes small non-uniformities of the rotor at the shaft frequency. Close to stall, however, there is a strong peak corresponding to a frequency of about 4 Hz, or about half the shaft frequency. The peak cross-correlation shown in this figure corresponds to a velocity of about 0.3% of the freestream axial velocity. What this figure does not show is that short time averages reveal a cross-correlation which rises and falls with time. At some times there was essentially zero correlation and at other times it was as high as 0.006 $m^2/s^2$. Since the mean flow coefficient was about 0.4 and the mean blade speed was 36 m/s it is easy to show that a cross-correlation of 0.006 $m^2/s^2$ correponds to a perturbation of about 0.55% in the mean axial velocity. When the correlation was strongest the frequency could be more accurately determined as 4.1 Hz.

The correlation indicates that there is a pressure field rotating around the annulus at about 49% of rotor speed prior to the compressor stalling. It is only very close to stall that they can be detected. Nevertheless the amplitude rises and falls during this time so the existence of these disturbances is not the same as stall, nor does it indicate that stall is occurring or about to occur. The same pattern was observed whatever the tip clearance being used and there was likewise no significant difference in frequency. Since, as discussed above, the initiation of the stall was from the casing for large tip clearance and the hub for the smaller clearances this indicates that the rotating disturbances ahead of the rotor were largely independent of the fluid mechanic process of stall. The amplitude of the disturbances was approximately constant across the annulus, indicating that it was essentially two-dimensional in nature.

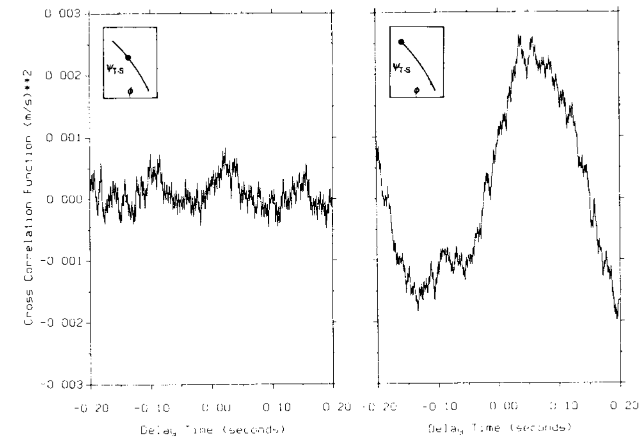

FIG. 6 VARIATION IN CROSS CORRELATION FUNCTION MEASURED UPSTREAM OF THE COMPRESSOR AS STALL IS APPROACHED

The cross-correlation raised the signal-to-noise ratio by using two sensors. An alternative way of looking at the upstream disturbances was suggested in discussion with Messrs Greitzer and Longley. The signals from six hot wires uniformly distributed around the circumference 0.6 mean radii ahead of the rotor were logged simultaneously as the compressor was allowed to stall. The signals were then Fourier analysed at each instant to resolve the circumferential harmonics, of which the dominant one was the first order, i.e. one cycle around the circumference. The result is plotted in Fig. 7 where the size of the symbols used is a measure of the amplitude. Figure 7 shows the phase of the signal against time (with arbitrary origins for each) and the slope of the lines gives the phase velocity. When the signal is weakest the lines are steepest and this corresponds to the non-uniformity of the rotor at first order rotating at rotor speed. When the first order mode is of larger amplitude the phase speed is lower, about 0.5 times as large. The record shown in Fig.7 is just leading up to stall, which occurs at about 4.25 seconds. It should be noted how the amplitude rises between about 1.5 and 2.0 seconds before falling, to rise again at about 3.25 seconds ultimately to stall. It is noteworthy that there appears to be no discontinuity in phase or frequency in changing from the small amplitude disturbance to the large amplitude stall cell. In both cases there is one cycle around the circumference, i.e. one cell, and the circumferential speed is about the same, just less than 50% of rotor speed. The same data, but this time displayed as the amplitude, is shown in Fig.8. An exponential curve has been fitted to the data in the period after 3.5 seconds leading up to the breakdown into stall.

## MEASUREMENTS IN PASSAGE DURING INCEPTION

Measurements of the flow inside the blade passage during stall inception were made with the largest tip clearance tested, 3% of

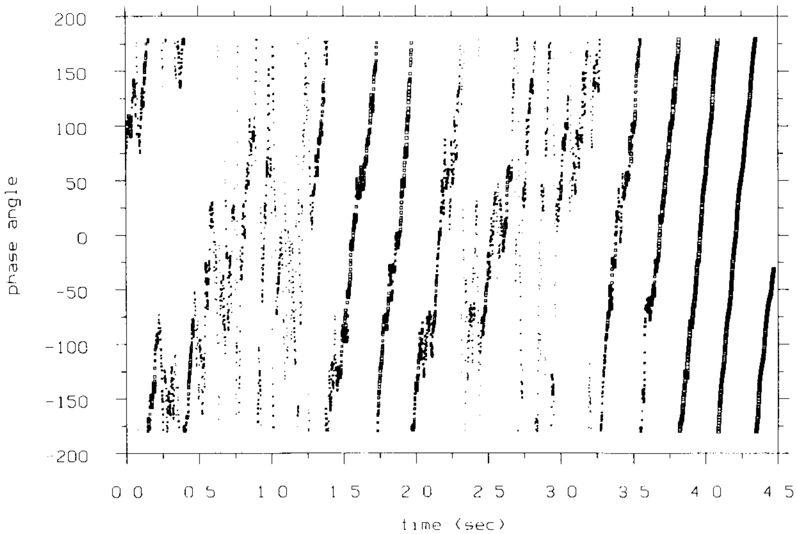


FIG. 7 VARIATION OF PHASE OF FIRST CIRCUMFERENTIAL HARMONIC WITH TIME DURING STALL INCEPTION (Arbitrary phase and time origins).

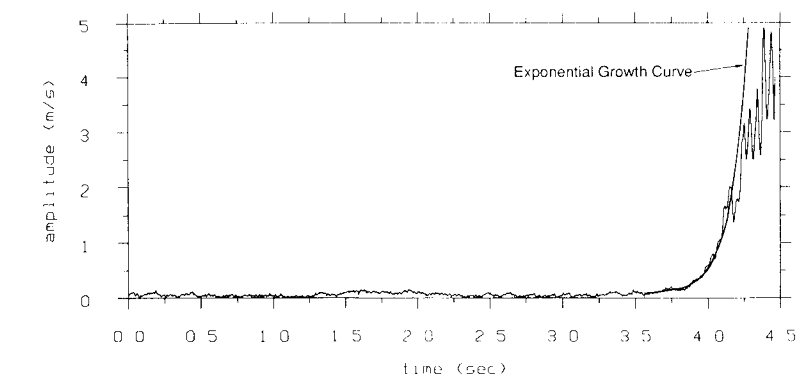


FIG. 8 VARIATION OF AMPLITUDE OF FIRST CIRCUMFERENTIAL HARMONIC WITH TIME DURING STALL INCEPTION (Arbitrary time origin).

chord. It was necessary to find where in the blade passage the stall could first be detected and the rotating-frame traverse gear was used to position a hot wire at different positions over the trailing edge plane while the compressor was stalled many times. A sub-miniature wire mounted on the suction surface of one blade was used as a reference. The position where the stall could be first detected is shown in Fig.9, just outside the annulus and suction surface boundary layers.

Stall initiation is stochastic, since its time is not known, and it can occur anywhere around a perfectly uniform rotor. To make the measurements more repeatable and to allow the on-rotor instrumentation to be in the correct position requires the position of stall inception to be fixed in a particular part of the rotor circumference. The first approach for fixing the position around the rotor of stall inception was to restagger one of the blades so that it was at slightly greater incidence than the others. A variation tried was to hang an obstruction in a passage. It became clear, however, that this was not achieving the goal of fixing the stall inception, indeed it was having no measurable effect on the stall. No change in the flow coefficient at stall and no apparent fixing of the stall inception was produced when the perturbation was taken as far as completely removing a blade!

An alternative strategy was then tried which was to restagger a group of blades by a small amount. Figure 10 shows clearly that there is an effect on the flow coefficient at stall for reductions in stagger of 1° or 2°; the effect increases progessively to reach its maximum value after about 8 blades. Even at its maximum the effect is still quite small, the flow coefficient at stall being increased by about 1%. The rotor had a total of 51 blades so 8 restaggered blades gives an arc of about 60°. Figure 11 shows the measured throughflow velocity contours at rotor exit with the rotor operating at virtually identical flow coefficients close to the stall point, in one case where the blade stagger is not reduced, in the other in the middle of the section of 8 blades restaggered by 2°. The boundary layer close to the casing and the rotor wake there have been thickened but generally the flow is very similar, indicating that the flow has not

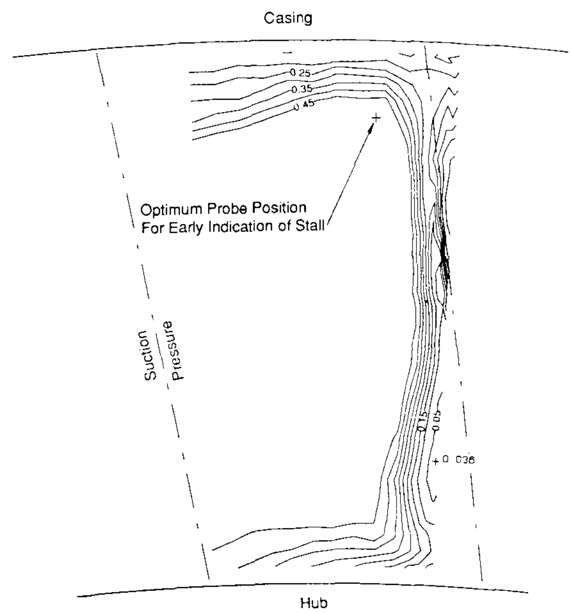


FIG. 9 AXIAL VELOCITY RATIO, ROTOR EXIT PLANE
$\phi$=0.429, Tip Clearance/Chord Ratio 3.0%.

been perturbed so seriously that it ceases to be similar to the unperturbed flow. Figure 12 shows bar charts obtained from hot wires mounted near the trailing edge on the suction surface. When the reduced stagger section was centered on hot-wire 1 the velocity recorded by this sensor dropped much earlier than wires 2 and 3; when the reduced stagger sector covered hot-wire 3 it was this sensor which showed the first signs of a fall in velocity.

With the phase of stall inception fixed with 8 blades at 2° reduced stagger it was possible to obtain an impressive degree of repeatability, allowing the formation of ensemble averages of the velocity in the period up to stall inception. An example of the raw and ensemble-averaged data is shown in Fig.13, taken at the trailing edge plane at mid pitch and 80% of the span from the hub. (It was found that little improvement accrued from using more than 10 ensembles though the running time increased linearly.) It was then possible to produce contours of throughflow velocity at different

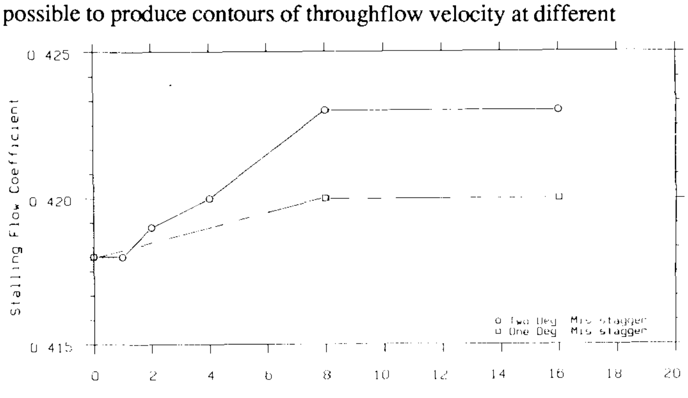


FIG. 10 VARIATION OF STALLING FLOW COEFFICIENT WITH SIZE OF MIS-STAGGERED SECTOR

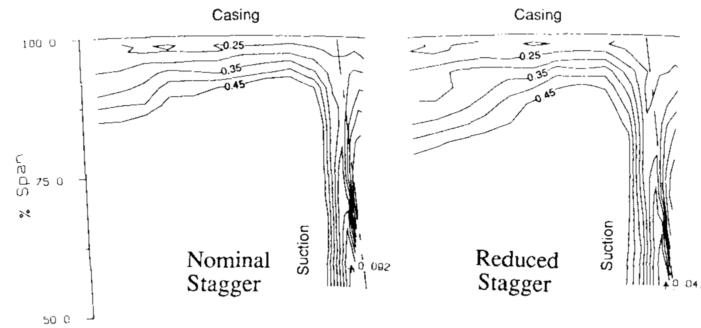


FIG. 11 THROUGH FLOW VELOCITY RATIO, ROTOR EXIT PLANE. Near Stall, Rotor Tip Clearance/Chord Ratio 3.0%

times during the stall inception ensemble averaged over 10 stall inceptions. Each of the contour plots is similar in appearance to Fig.11 but the annulus boundary layer varies in thickness, particularly near the suction surface, in an approximately cyclic

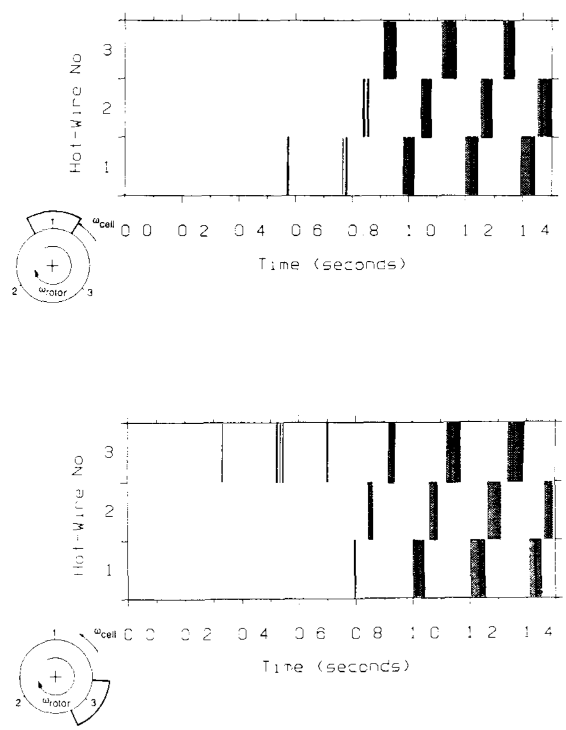


FIG. 12 BAR CHARTS RECORDED DURING STALL INCEPTION
Top — Probe 1 at centre of mis-staggered sector
Above— Probe 3 at centre of mis-staggered sector

manner. The frequency is that of the rotating upstream disturbances, transposed into the rotating frame of reference. During some parts of the cycle the boundary layer is thinner than in the steady state prior to the initiation of the inception process. The amplitude increases with time until stall has occurred.

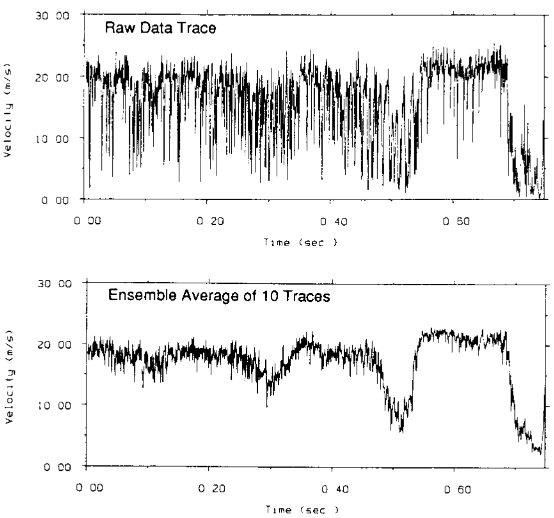


FIG. 13 EFFECT OF NUMBER OF SAMPLES ON NOISE CONTENT IN ENSEMBLE AVERAGED DATA

## DISCUSSION

There are two separate strands to the work described here. One is the behaviour in and around the rotor blades, the other is the rotating perturbations detected upstream.

The rotating upstream perturbations had not been observed before but since have been found in other compressors by, Day(1988) and by Garnier(1988). They are not in themselves surprising but represent the response of the upstream flowfield to the disturbance in the blades. They are then the modes of the system. If the blade disturbance were made to remain stationary, as with partial casing treatment in Cumpsty(1989), some of the upstream perturbation would be recognisable as a stationary pressure field. It is notable that the field appears to be essentially two-dimensional, not varying appreciably in the radial direction, and being similar for in-blade disturbances at the hub or at the casing. No explanation has been found for the dominance of the first-order mode with one cycle around the circumference, although the exponential decay will be larger for the higher modes. In tests on another machine Day(1988) has found the second-order mode to predominate. If the disturbance field is important it is not surprising that disturbances to one blade passage, even going as far as removing one blade, did not have much effect. This is because the coupling between the passage disturbance on a small circumferential scale and the first-order mode, which has the scale of the circumference some 51 times as large, is weak.

The amplitude of the upstream perturbations detected is quite small, around 0.5% of the mean axial velocity at its greatest prior to stall. This was detected 0.6 mean radii upstream of the rotor and assuming exponential decay would give about 0.9% at the rotor inlet. This, although small, is comparable to the magnitude of the changes in flow coefficient at stall achieved by restaggering the blades by 2°, Fig.10.

Many attempts have been made at predicting stall cell speed by considering small, two-dimensional disturbances. No method considers details of the flow in individual blade passages, but they rely instead on an actuator or semi-actuator disk to model each blade row. Their mixed success is not surprising in view of the measurements reported here. Nevertheless it seems clear that the rotational frequency is set not just by the flow processes taking place in the rotor itself but also by the dimensions of the upstream and downstream ducts and the boundary conditions to these ducts. In a multistage configuration the downstream blade rows will exert a large influence on this as well as on other aspects of the stall process.

The measurements made in and downstream of the passages have shown that stall inception occurs in different ways depending on the magnitude of the tip clearance. In each case it is the blockage caused by the endwall boundary layers, more properly the combination of the endwall and blade boundary layers. With large tip clearance the blockage near the casing appears sufficient to reduce the blockage near the hub and render the flow in this region relatively stable so that the inception occurs near the casing. At small tip clearances the flow rate at stall was lower, the blockage near the hub was large and it was near the hub that the first unsteadiness leading to stall occurred. The speed of the disturbances detected upstream was very similar in each case which may indicate that the wave speed is only weakly determined by the in-passage flow details.

At a tip clearance equal to 3% of chord the stall inception could be observed bringing about a fluctuation in the annulus boundary layer, particularly near the suction surface. The period of this disturbance corresponded, of course, to the period of the upstream disturbance after transposing to the moving frame of reference. The process leading to stall inception was highly repeatable, evidenced by the fact that ensemble averages could be achieved, and this makes it clear that the process, though not its position or start, are not random but highly deterministic.

A previous investigation related to the study of stall inception was reported by Jackson(1987) on a similar compressor. Jackson used hot-wires upstream and downstream of the rotor but was unable to make measurements in the rotating frame of reference. He restaggered a single blade to fix the stall cell inception but it is now apparent that any fixing that occurred was originating in some other way. By examining the velocity traces measured upstream of the rotor Jackson estimated the stall cell speed from its first appearance until fully developed: the cell appeared to begin as a small cell travelling at near rotor speed and to decelerate rapidly in the absolute frame of reference as it grew in size. This supported the view that stall inception occurred when a particular blade, believed to be the restaggered one, experienced a conventional separation type stall along the lines proposed by Emmons et al. The current measurements made in the rotating frame show that this is not how stall begins. Stall can be detected much earlier in the rotating frame yet from their first appearance of the disturbance rotates at about 50% of the rotor speed. McDougall(1988) has shown that Jackson was in fact looking at cells which were fairly well developed when first detected and that the measurements upstream of the rotor gave an erroneous indication of speed change as a result of the simultaneous increase in cell size.

The measurements described link blockage in the endwall regions with a perturbation field upstream of the rotor, the amplitude increasing as the compressor undergoes stall. It does not provide an explanation for the stall occurring when the overall $\partial(p_2-p_{01})/\partial V_x<0$. Other evidence, some experimental, some computational, helps to explain this. A number of years ago measurements had been made in Cambridge on a rotor with a casing treatment over part of the circumference. These are now reported by Cumpsty(1989) and show that close to stall the boundary layer thickness upstream of the rotor varies around the circumference with this geometry: in the same direction as the rotor the thickness increased over the portion with no treatment and decreases sharply where there is treatment. The only explanation which could be found for this is that close to stall the untreated section is allowing flow to spill forward from one passage to the next. These measurements also showed a non-uniform pressure field upstream and downstream of the rotor, analogous to the upstream perturbations found in this present work. The pressure rise through the compressor was least where the inlet blockage was greatest, near the end of the untreated section. When the pressure rise and flow rate changes that these upstream and downstream perturbations were added to the mean values of $p_2-p_{01}$ and $V_x$ it is plausible to interpret the local value of $\partial(p_2-p_{01})/\partial V_x$ as approximately equal to zero at the end of the no-casing-treatment section. In other words it is not necessary for the entire compressor annulus to satisfy the instability criterion but only a section. When these measurements were made there was no other evidence, experimental or theoretical, to support the idea of flow spilling forward of the rotor.

clearance vortex. At low flow coefficients the calculation would no longer converge. The lowest value at which full convergence could be obtained is $\phi$=0.48, for which the clearance flow is shown in Fig. 15. This time the clearance jet does not pass down the blade passage but is spilled forward of the next blade into the next passage. On reflection it seems reasonable that the jet would either go downstream or upstream of the leading edge region of the next blade but not on the leading edge because this is a region of high static pressure. The spillage of flow from one passage to another helps explain why the downstream velocity contours are more nearly uniform on the casing at flow rates near to stall because the relatively well defined jet is no longer present. It seems highly likely that with spilled flow small changes in pressure can bring quite large changes in flow entering the passages near the endwall. It also provides a mechanism by which the endwall boundary layer flow in one passage can affect others.

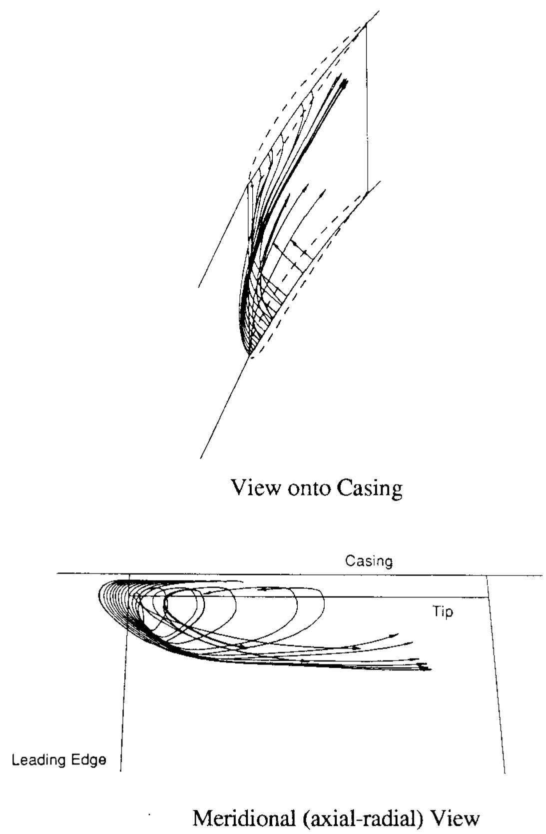


FIG. 14 PREDICTION OF TIP LEAKAGE FLOW— DESIGN POINT

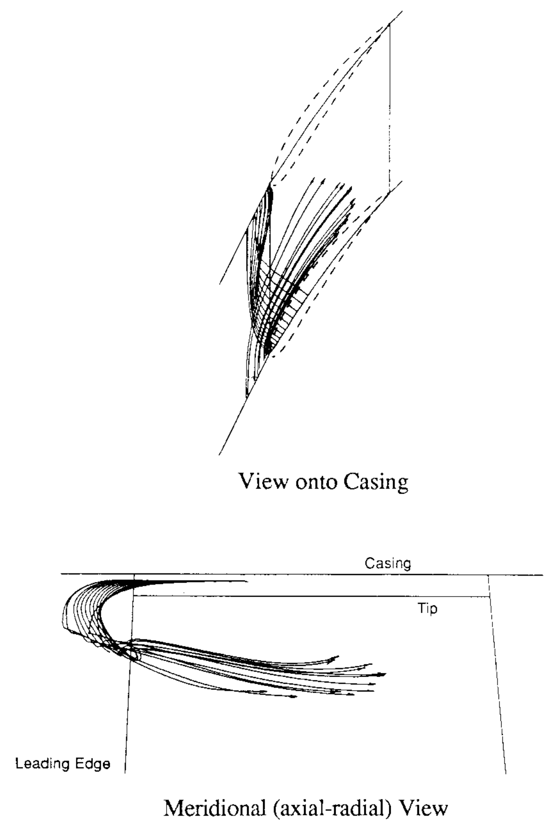


FIG. 15 PREDICTION OF TIP LEAKAGE FLOW— CLOSER TO STALL

Calculations of the flow around the tips of a rotor blade were performed using the Navier-Stokes solver written by Dawes(1987). (The success of this method in predicting a number of aspects of the flow was demonstrated by McDougall and Dawes, 1987 and also by McDougall,1989.) Figure 14 shows predictions of the tip leakage flow at a condition close to the design flow coefficient, $\phi$=0.55, with a tip clearance of 3% of the chord. In this case the tip clearance jet emerges with an upstream component of axial velocity, is turned by the main flow and becomes what is usually referred to as the

The prediction of spillage near to stall goes some way to explain the operation of the axial-skewed-slot type of casing treatment. Recent measurements with hub treatment under a stator have shown that sucking away the high loss fluid near the back of the blade or blowing near the front of the blade can delay stall, Lee and Greitzer(1989). Suction near the back was expected but the benefit of blowing near the front was a surprise. It seems probable that the high momentum of the flow leaving the treatment slot is able to entrain the clearance jet and prevent it from spilling ahead of the next blade, thereby delaying the onset of instability.

The evidence that has come from this work and that reported by Cumpsty(1989) is that the collapse of the flow leading to instability and stall takes place over a relatively small part of the annulus. Other evidence points to this section being much larger than one blade pitch. The collapse of the flow leads to a change in the slope $\partial(p_2-p_{01})/\partial V_x$ such that instability is possible. This is not in conflict with the theoretcial studies of system instability which take this gradient as an input variable. Hynes and Greitzer(1987), for example, show that it is the average of the gradient $\partial(p_2-p_{01})/\partial V_x$ around the circumference which determines whether the compressor is unstable: the present work does not conflict with this. It has shown that the slope can change quite markedly in some regions of the circumference as a result of local changes to the flow and that an overall value of $\partial(p_2-p_{01})/\partial V_x$ may not give a true indication of the nearness to stall.

A fashionable topic of research at the present time is active control of stall and surge: a control system will be linked to the compressor so that on recognising the incipient stall an appropriate control action will be undertaken. One of the problems is the recognising of stall in the noisy environment of a compressor operating close to stall. The results shown in Figs.7&8 are encouraging. They show that a low order mode is dominant and can be observed for some considerable time prior to stall. They also show that using a number of transducers and fourier analysing these to pick out any low-order modes which may be present is a way of eliminating much of the noise. It should be repeated, however, that the low-order modes are not the same as stall and that they were observed to rise and fall in amplitude for some time before stall inception occurred. What one can also say is that only very close to the conditions for stall were they ever of significant amplitude so their existence is evidence for the controller that remedial action to avoid stall is urgently required.

## CONCLUSIONS

1) With high reaction stages the rotor is much more likely to be the cause of stall. For the same overall geometry a compressor can stall at the rotor hub for small tip clearance and at the rotor tip for larger clearance. At the level of clearance typical of multistage compressors tip stalling is more likely.

2)Attempts to fix where around the rotor circumference stall was initiated were unsuccessful when only one blade or blade passage was involved. Only when a significant part of the circumference was altered did it affect the stall point: the full effect was achieved when the alteration was to about 60° of the circumference.

3) The crucial property of the flow in the blade passages for determining the stall point is the endwall blockage. For the compressor stage used for this investigation the blockage becomes large on the hub at small tip clearances, at large tip clearances it becomes large at the casing. The dependance of casing blockage on tip clearance is believed to be general to most axial compressors. The variation in blockage revolves around the annulus, the speed was about 49% of the rotor speed for this compressor.

4) In the upstream flow field rotating variations in axial velocity were measured. These were in the first-order mode, with one cycle around the annulus and almost constant in the radial direction. The mode rotated around the annulus in sympathy with the variation in blockage at exit from the rotor. The speed of rotation was found to be about 50% of the rotor speed, similar to the fully developed stall cell speed. (This similarity is probably a coincidence for this stage and does not occur on all compressors.) At the rotor inlet plane the magnitude of the axial velocity associated with the upstream disturbance was about 1% of the mean axial velocity near the stall point.

5)Calculations have shown that the flow pattern associated with the tip clearance changes markedly as the flow rate is reduced. The tip-clearance jet no longer turns into the adjacent blade passage but spills into the next one along the row. This is believed to be an important reason for the sensitivity of the row to very small changes in inlet flow rate. The greater inlet blockage caused by the spilled flow leads to a reduced pressure rise across the row.

6)It would seem that stall can be incurred when the flow in only part of the annulus is unstable and that this occurs first where the blockage is greatest.

7) The presence of stall could be detected much earlier with a sensor mounted on the rotor than with upstream or downstream measurements; maximum sensitivity was obtained with the hot-wire just outside the boundary layer. With on-rotor hot wires the speed of the pre-stall disturbances and of the stall cell itself were found to be similar and equal to about 50% of the rotor speed for this compressor stage. There is no evidence that the initially small stall cell is fixed to the rotor or rotates with an absolute velocity approximately that of the rotor.

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