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Active Suppression of Rotating Stall and Surge in Axial Compressors

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ABSTRACT

This paper reports on an experimental program in which active control was successfully applied to both rotating stall and surge in a multi-stage compressor. Two distinctly different methods were used to delay the onset of rotating stall in a four stage compressor using fast acting air injection valves. The amount of air injected was small compared to the machine mass flow, the maximum being less than 1.0%. In some compressor configurations modal perturbations were observed prior to stall. By using the air injection valves to damp out these perturbations an improvement of about 4.0% in stall margin was achieved. The second method of stall suppression was to remove emerging stall cells by injecting air in their immediate vicinity. Doing this repeatedly delayed the onset of stall, giving a stall margin improvement of about 6.0%. Further studies were conducted using a large plenum downstream of the compressor to induce the system to surge rather than stall. The resulting surge cycles were all found to be initiated by rotating stall and therefore the stall suppression systems mentioned above could also be used to suppress surge. In addition, it was possible to arrest the cyclical pulsing of a compressor already in surge.

INTRODUCTION

Compressor performance at the peak of the pressure rise characteristic is limited by aerodynamic instabilities which lead to rotating stall and surge. Rotating stall is a localised disturbance affecting the compressor only and leads to excessive blade vibration. Surge, on the other hand, is a system oscillation associated with reversed flow transients and these cause unusual stresses in the compressor. The accepted way of avoiding these dangers is to restrict the compressor to an operating point safely removed from the stability boundary. The stability boundary, however, is often ill-defined, being affected by engine acceleration, inlet distortion and deterioration with age, and therefore the safety margins allowed are generous. To decrease the need for wasteful safety margins, and thus increase the useful operating range of the compressor, Epstein et al (1986) first suggested in the open literature that active control techniques might be employed to delay the onset of stall or surge.

In the context of the proposals put forward by Epstein et al, stall and surge are seen as limit cycles whose final strength is set by non-linear effects. In origin, however, these disturbances start out as small perturbations and, if treated early enough, can be modelled by linear theory. Active control as envisaged by Epstein et al, would be applied at this linear stage of development and would require the feedback of additional disturbances into the flow field. These disturbances would have the effect of increasing the damping in the system and would thus restrict the growth of the original perturbation. The additional disturbances would be created by a system of actuators driven by a controller using real time measurements from inside the compressor. A control system such as this would require minimal input effort and would allow the compressor to operate safely at flow rates which would otherwise have been unobtainable.

Alternative stall suppression systems have been proposed in the past, using controllers which sensed the approach of stall and took remedial action fast enough to prevent the disturbances from developing. Ludwig and Nenni (1980) suggested using fast blow-off valves and Reis et al (1987) proposed a system using rapid re-scheduling of the stator vanes. This type of control is achieved simply by unloading the compressor and is termed avoidance control. In practice neither of these approaches falls into the category of true active control, as defined herein, since neither interferes with the unsteady aerodynamic damping in the compressor.

The conceptual difference between active control and stall avoidance control is illustrated in Fig. 1. In this comparison the avoidance control makes use of a fast blow-off valve, while the active control system is based on continuous stabilising feedback. Both systems allow the compressor to deliver at a lower than usual mass flow rate while avoiding stall, but in each case the true operating point of the machine is very different. With active control the blade loading is pushed to higher values without the basic fluid mechanic processes, by which the pressure rise is produced, being altered. In the case of the blow-off valve, the effective operating point of the machine is kept on the stable side of the characteristic by dumping excess air. The bleed valve approach is less efficient and, unlike active control, the pressure rise will be lower if a safety margin is to be maintained.

In the approach to active control suggested in the literature, the initial phase of stall development is modelled in terms of long length scale disturbances of small amplitude. Stall inception measurements by McDougall et al (1990), and Garnier et al (1990), have shown that in some instances stall cells do develop from such beginings, but this is not always the case. It has been shown by Day (1991) that emerging stall cells, representing short length scale disturbances, can originate without any detectable precursive build-up. This means that linear modeling will not cover the whole range of stall inception possibilities, and that active control will be more complex to apply than originally envisaged.

Disturbance of long length scale, ie modes, can be damped out through the feedback of counteracting disturbances. Stall cells emerging without precursive build-up cannot be treated in this way. It

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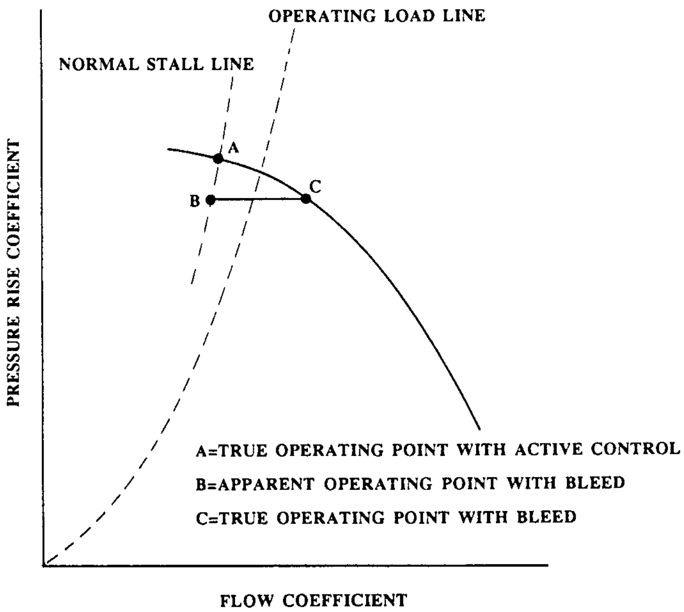
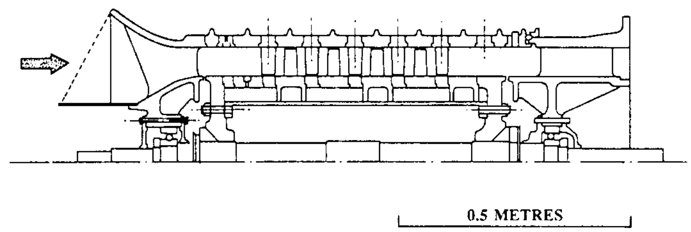


Fig. 1 Schematic of compressor characteristic emphasising the difference between active control and control using a bleed valve.



THE C106 FOUR STAGE COMPRESSOR

Mid-Height Blading Details and Other Parameters

	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	

Fig. 2 Cross-sectional view of the C106 Compressor with table of basic design details.

will be shown below, however, that these cells are limited in circumferential and radial extent and, as such, are amenable to localised corrective action Stabilisation in this case does not depend on increasing the damping in the system, but rather on energising the flow in the vicinity of the blade tips and thus making it more disturbance tolerant. This approach, like that applied to longer length scale disturbances, requires minimal input effort and therefore can be included in a broad definition of the term 'active control'.

The details of stall inception in the C106 compressor will be considered below, showing why two different approaches to stall suppression are necessary. The experimental results covering the damping of modal waves, and the elimination of unannounced stall cells will then be considered. Finally it will be shown that surge inception is initiated by rotating stall, and the use of stall suppression to avert surge will be demonstrated.

Stall Inception Measurements in a Four Stage Compressor

The four stage compressor chosen for these experiments is one of those used in a wider study of stall inception by Day (1991). The machine is a low speed research compressor with four identical stages preceded by a lightly loaded set of inlet guide vanes. The hub-casing radius ratio is 0.7. The blading is of modern controlled diffusion design and is intended to be representative of current H.P. compressor practice. A concentric throttle was used, either close coupled to the compressor for rotating stall studies, or mounted downstream of a large plenum for the study of surge. A schematic diagram is shown in Fig. 2 along with some basic design values.

To map out the stalling behaviour of the compressor, a number of hot-wires equally spaced about the circumference of the machine were used. The wires could be positioned at almost any axial location throughout the compressor and several axial stations could be examined simultaneously. It was found that the stall cells always originated near the tips of the first stage rotor blades and then spread in both the radial and axial directions. A stall cell could be detected at the back of the machine roughly one full revolution after being seen in the front; this was also about the time needed for the cell to spread from the casing to the hub on the first stage.

In terms of the way the compressor goes into stall, the pattern of finite cell formation is always the same, regardless of whether modal (long wavelength) perturbations are present or not. A small sharply defined stall cell, four or five blade pitches wide, forms abruptly near the first stage rotor tips and rotates around the annulus at about 70% of rotor speed. As the cell grows radially and circumferentially, the speed of rotation slows down until after about 4 complete revolutions the cell is fully formed and moving at about 38% of rotor speed. A typical example of cell formation, without a modal perturbation being present, is given in Fig. 3. The output from four hot-wires equally

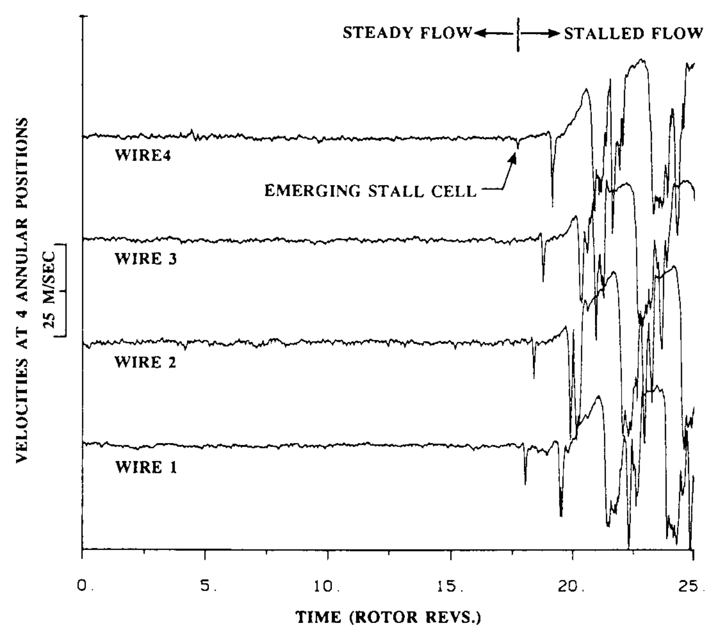


Fig. 3 Hot-wire measurements showing a small finite stall cell emerging from a steady flow field. (Mean axial velocity prior to stall ~ 30m/sec.)

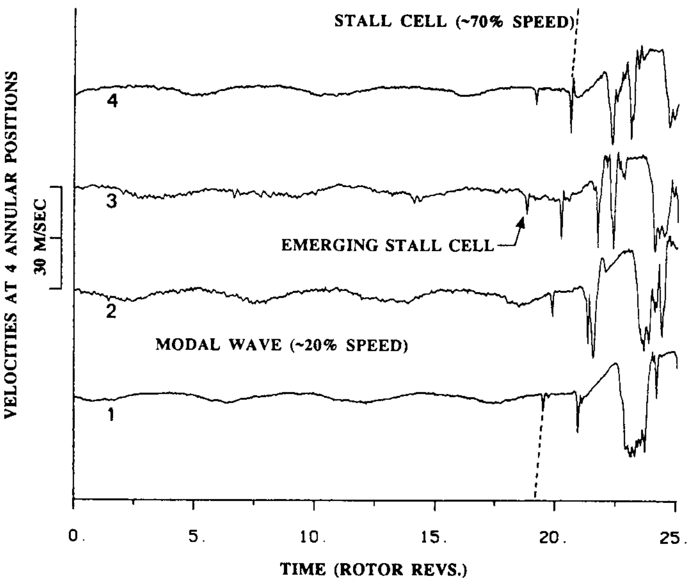


Fig. 4 Hot-wire measurements showing a small finite stall cell emerging from a flow field perturbed by a clear modal wave.

figure a stall cell appears abruptly at about rotor revolution 16. No form of precursive build up to the cell can be detected, even when the data is spatially and temporally decomposed.

In some experiments, where the tip clearance over the first rotor was increased from 1.2% to 1.5% chord, modal waves appeared in the compressor prior to stall. These waves represent a small amplitude velocity perturbation and may be thought of as the linear phase of the approach to rotating stall. The perturbation is detectable right throughout the compressor and in the incoming flow. An example of cell formation accompanied by these waves is given in Fig. 4. Besides the waves, the point to note is that when the stall cell appears (at time t= 17) its dimensions, speed of rotation, and rate of growth are the same as in Fig. 3. In other words the stall cell does not appear to be a continuous development of the modal wave, but rather a separate disturbance which appears despite the presence of the modal wave. Repeated measurements show, however, that the two phenomena are not totally unrelated. The trough of the modal wave represents a deficit in the through flow velocity and this often provides a natural starting place for the stall cell. (The details of stall inception in this compressor are discussed in greater detail in a companion paper by Day (1991).)

The experiments from the C106 therefore imply that, although modal perturbations are sometimes present, they do not appear to be the cause of stall. Fully developed stall only occurs after the formation of a short length scale stall cell, so that limited scope exists for a linear interpretation of stall build-up in this compressor. Recent experiments on a real engine confirm, however, that the C106 is a representative machine and as such provides an appropriate vehicle on which to examine stall suppression. In the work described below two approaches to stall suppression have been investigated. The first is based on suppression of the modal wave (when the modal wave exists) and the second introduces a new technique of flow energising applied directly to the emerging stall cell. The latter approach is applicable whether the modal wave is present or not.

THE CONTROL SYSTEM

To implement stall suppression, a system of actuators was necessary to interact with the flow field in the compressor. Numerous ideas, including tip clearance control and actuated inlet guide vanes, were considered but in the end an air injection system was selected. (Use of actuated inlet guide vanes is described by Paduano et al, (1991)) An array of twelve individually controllable valves was designed and built. These valves were to be positioned near the tips of the first rotor as this is where the stall cells were known to first appear. (An alternative to the 12 valve system was also developed for use in some of the early experiments. This system, described more fully below, consisted of a ring of 60 small air injection slots, fed by a single valve.)

To make room for the air injection valves (and the ring of slots) the inlet guide vanes were moved upstream by one chord length. These vanes are lightly loaded and tests showed that this move did not affect the stalling pattern in any way. The twelve trapdoor-type valves were then interposed between the inlet guide vanes and the first rotor. The valves were equally spaced around the circumference. The valve flaps were about 25 mm square, hinged along one side, and were flush with the outer casing of the annulus when closed. When open, about 3mm, air was emitted into the main stream of the compressor near the outer casing wall. Each valve was housed in a cylindrical housing and could be rotated about its axis so that the direction of the air injection could be varied. A schematic of the valves is shown in Fig. 5.

The injected air was supplied by an outside source. When all 12 valves were open at the same time, a total of 1% of the compressor flow rate at stall was consumed, ie each valve passed a flow equal to about 1/12th of 1%. This satisfies the basic requirement that the control effort expended should be minimal in comparison with the power of the compressor. The maximum velocity of the air leaving the valves, when measured just downstream of the valve opening, was about 1.5 times the mean axial velocity in the compressor. To drive the valves, an on-off signal was supplied either by an analog controller, or a mini-computer. Testing with a square wave signal generator gave a maximum frequency response of about 140 cycles per second, i.e. about 3 times rotor frequency or 7 times stall cell frequency.

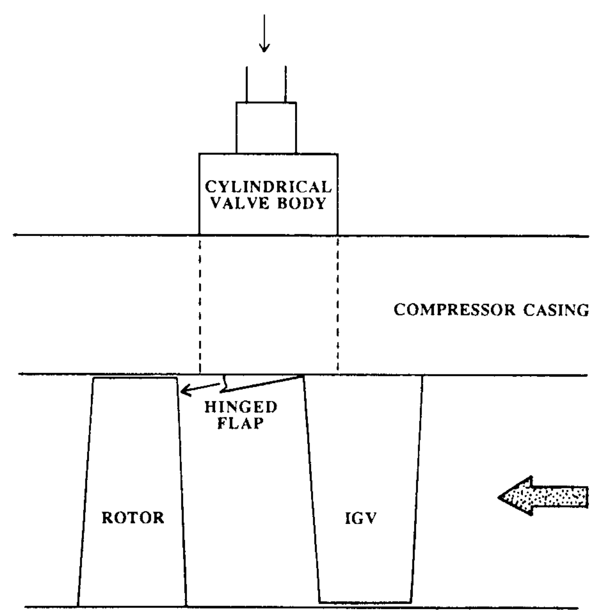


Fig. 5 Schematic picture of trapdoor-type air injection valve.

EXPERIMENTAL RESULTS

1. Suppression of Modal Waves

The air injection valves could be used in a variety of different ways to delay the onset of stall. We consider first the case where modal waves appear in the compressor prior to cell formation. As pointed out earlier, the modes in this compressor do not appear to turn into stall cells themselves, but rather generate a localised dip in the through-flow velocity where stall cells tend to start. By reducing the

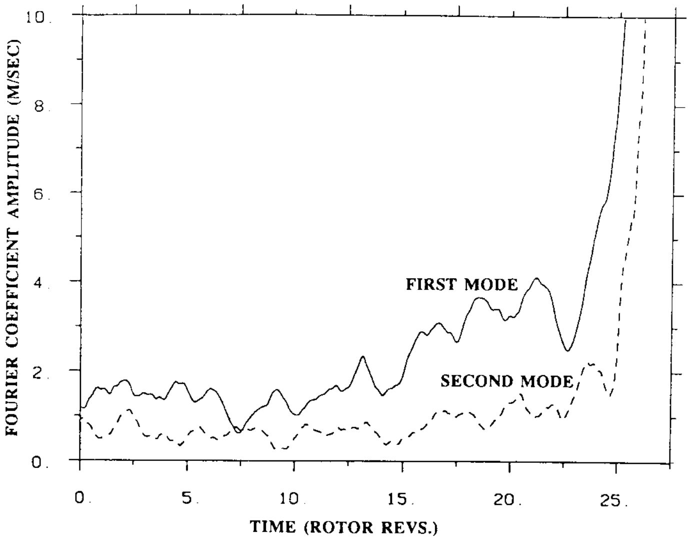


Fig. 6 Amplitude of first and second circumferential modes measured as the compressor goes into stall.

amplitude of the modal waves, and so removing the localised velocity deficit, the average flow rate through the machine can be lowered slightly without causing stall. As the axial velocity dips associated with the modal waves do not usually grow to more than a few percent of the mean axial velocity, it may be conjectured that the decrease in mass flow at stall onset would only be of the same order.

With sufficient hot wires around the circumference, the velocity distribution at any time can be treated with a discrete Fourier transform to obtain a measure of the amplitude and phase angle of any modal perturbation. Repeating this process at each sampling interval, the growth and position of the modal wave can be tracked from well before stall until the stall cells are essentially fully developed. An example is given in Fig. 6 where the amplitude of the first and second order modes are plotted for the compressor approaching stall. The first mode is clearly dominant in this case. The amplitude of this mode rises progressively until, at revolution 26, a finite stall cell is formed. The point of finite cell formation can be seen quite clearly at revolution 26 in the original data from which this figure was obtained. Furthermore the phase angle plots of this data show that the first order mode is well defined and rotates steadily around the annulus. The second order mode is ill defined and the phase angles show a total lack of coherence.

The 12 air injection valves could be used to damp the modal oscillation shown in Fig. 6. To achieve this, the output from six hot wires were used in conjunction with an analogue computer to open and close the valves. Full proportional control was obviously not possible using a discrete number of valves which are not amplitude controlled, i.e. the valves were either open or closed but could not be held at an intermediate position. Effective damping could, however, be achieved by judicious adjustment of the number of valves in use at any time and by varying the opening times and the supply pressure to the valves. The results using this scheme are shown in Fig. 7.

Fig. 7. is actually a composite picture made up of Fig. 6, trace A, plus two other sets of measurements using the air injection valves. Only the first order modal amplitudes are displayed in this case. The trace marked 'B' shows how, with control system in operation, the velocity perturbations can be kept to a relatively low value right up to the point at which the compressor stalls. The inset diagram shows that in this case the compressor characteristic was extended beyond the natural stalling limit; the benefit achieved represents about 4% reduction in stalling flow rate. The third trace in Fig. 7, trace C, shows the effect of deliberately offsetting the phase shift in the controller so that the action of the valves exacerbate the modal perturbations. When this is done the compressor stalls prematurely.

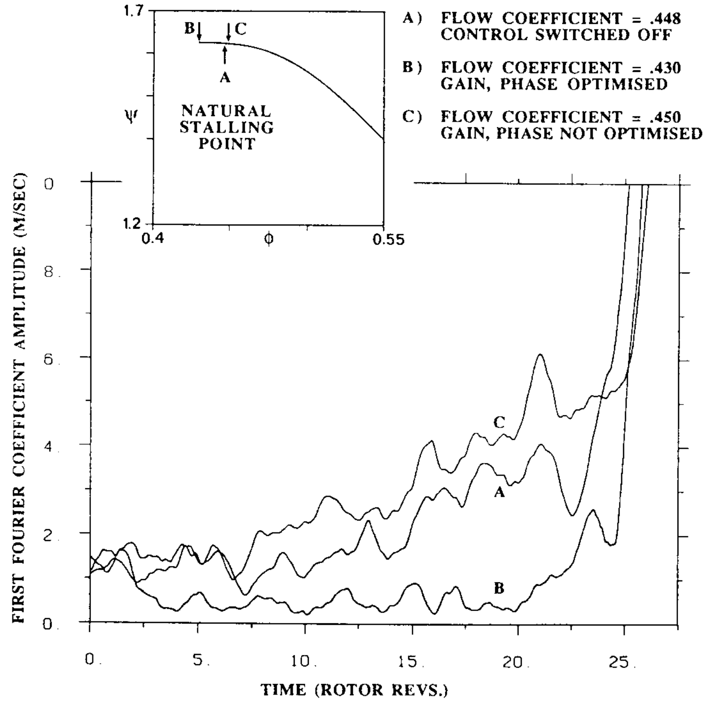


Fig. 7 Amplitude of first circumferential mode going into stall for different control options. Note that each complete data trace was recorded at a different throttle setting.

In interpreting Fig.7 it should be emphasised that the three sets of results each show a snap shot of what happens during the last half second as the compressor goes into stall and it should be emphasised that the results were obtained at very different throttle settings. In other words, the data traces have been overlaid on a common time axis for comparative purposes only.

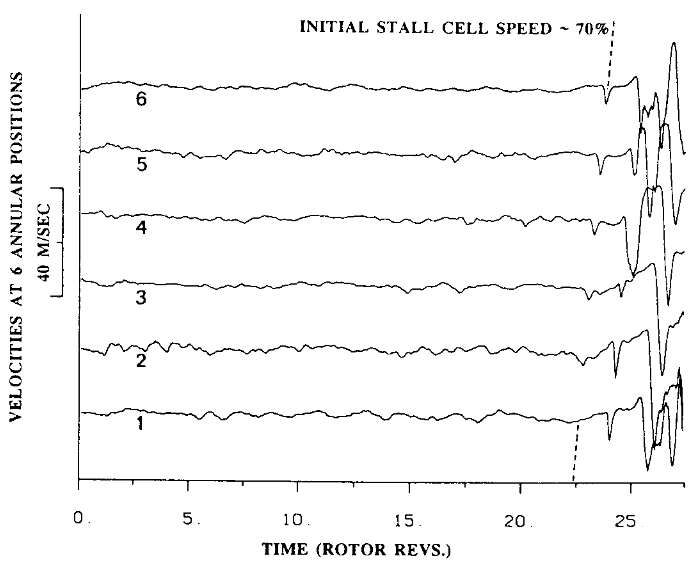
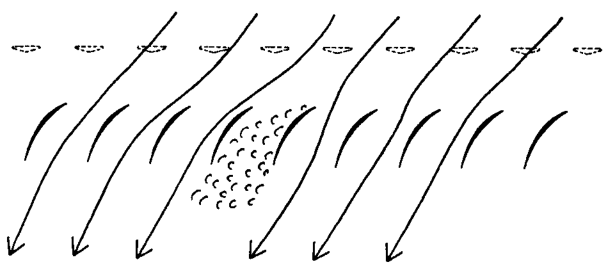


Fig. 8 Hot-wire measurements recorded while control system in operation. A coherent modal pattern is not detectable prior to the formation of a small finite stall cell.

When the compressor goes unstable with the control system working effectively (trace B) it is interesting to note that stall does not occur because the controller losses authority over the long length scale modes. It is rather that a short length scale disturbance, a stall cell, appears in the flow field and grows within about four revolutions to become a fully developed cell. The original data from which trace B was derived is shown in Fig. 8 to emphasises that this compressor stalls through the formation of a small fast moving stall and not through the exponential growth of a modal perturbations. (It should be stressed that the hot-wire signals shown in Fig. 8 were more severely filtered than in Figs. 3 and 4, and were AC coupled for compatibility with the control system. The emerging stall cell therefore appears less sharply defined than in the previous figures.)

2. The Suppression of Finite Stall Cells

Whether modal perturbations are present or not, the C106 compressor always ends up in rotating stall shortly after the appearance of a small sharply defined stall cell. Direct interference with the stall cell itself may therefore be an effective method of extending the compressor operating range. A useful way of visualising these small stall cells is to use a flow picture first suggested by Emmons (1955). In Sketch A the separated flow in one of the blade passages is shown diverting the incoming streamlines to either side of the affected area. Emmons suggested that the resulting increase in incidence on one side of the separation, and the decrease on the other, provides a mechanism for the disturbance to propagate from blade to blade around the machine. While this picture may not be accurate in all respects, it provides a useful means of visualising how air injection might be used to energise the flow field and remove the blockage. The direction in which the air is injected plays an important part in the process. As drawn, Sketch A also emphasises the fact that the stall cell is a localised disturbance and therefore localised corrective action may be appropriate.



Sketch A

To initially test the idea of using injected air to correct the flow in a localised disturbance it was decided to use a very simple injection system consisting of one fast acting valve, a manifold and a row of small directional slots. The relative size and position of these slots is indicated in Sketch A. The principle is the same as in the case of the trap-door valves, ie to inject air near the casing wall at the rotor tips where the stall cells are known to appear first. Using a manifold arrangement means that there is an inherent delay in the response time of the system but this was kept to a minimum. The quantity of the air injected through the slots was limited to a total of 1% of the compressor flow rate. An array of 8 probes was used to detect the emerging stall cell anywhere in the annulus and the central valve was opened immediately thereafter. The use of continuous air injection to delay the onset of stall is not a new concept. However, the use of air injection to remove a stall cell which has already become a finite amplitude disturbance has not been tried before. The results are important because they show that stall control after cell formation is a viable proposition.

Fig. 9 illustrates the operation of the system. The left hand side of the figure shows the compressor going into stall naturally (without the control system being used). The right hand side shows that opening the air injection valve as soon as the cell is detected will suppress the cell. The delay in the response of the system meant that the stall cell executed one complete revolution before the injected air could affect it.

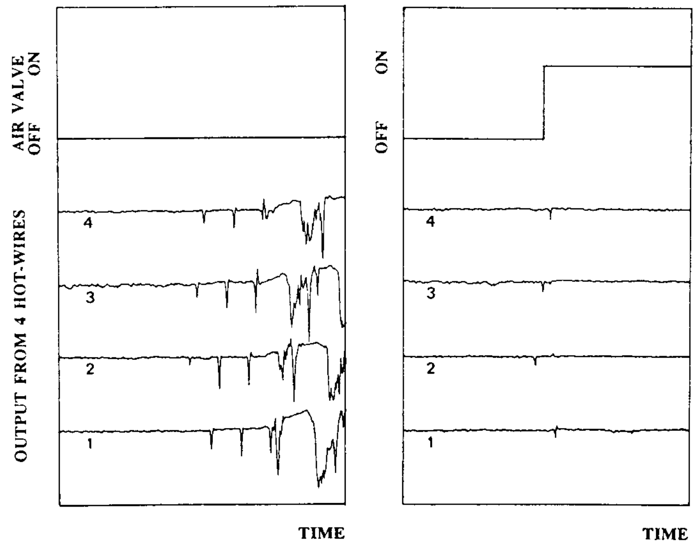


Fig. 9 Hot-wire measurements showing stall cell developing without air injection (LHS), and truncation of cell development with air injection (RHS).

This delay in response was acceptable in this case because the stall cell developed relatively slowly. Increasing the delay time in the detection system by just a half cell revolution more meant that the cell had time to grow bigger and the air injection system became ineffective.

In the right hand half of Fig. 9 the air supply was maintained after the stall cell had disappeared. If the supply were removed at some stage, a new stall cell would appear and the air supply would again be required. This on/off approach was tried by allowing the central valve to close again after a set number of milliseconds if no cells were detected. Fig. 10 shows the result of such a test where air is injected only when required; it also shows that if the detection system is switched off the compressor goes into stall. (It is interesting to note that after the cell is suppressed a new cell does not appear instantly.

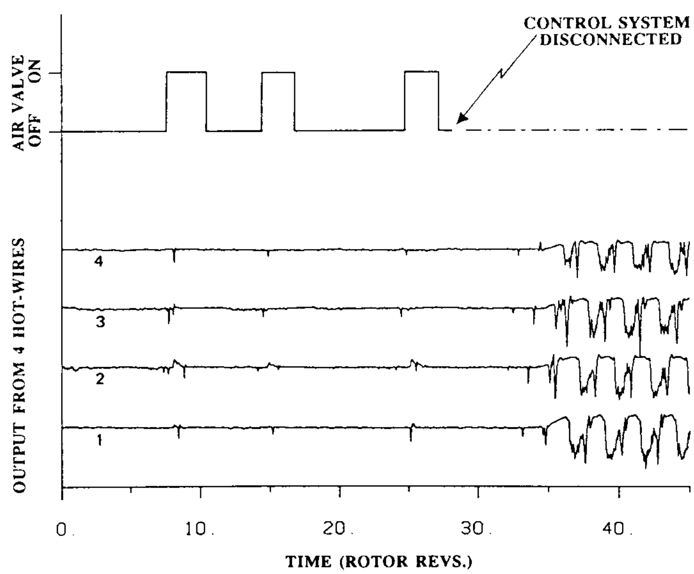


Fig.10 Measurements showing air injection being switched on only when necessary (LHS). At time t=29 the control system is disconnected and the compressor stalls.

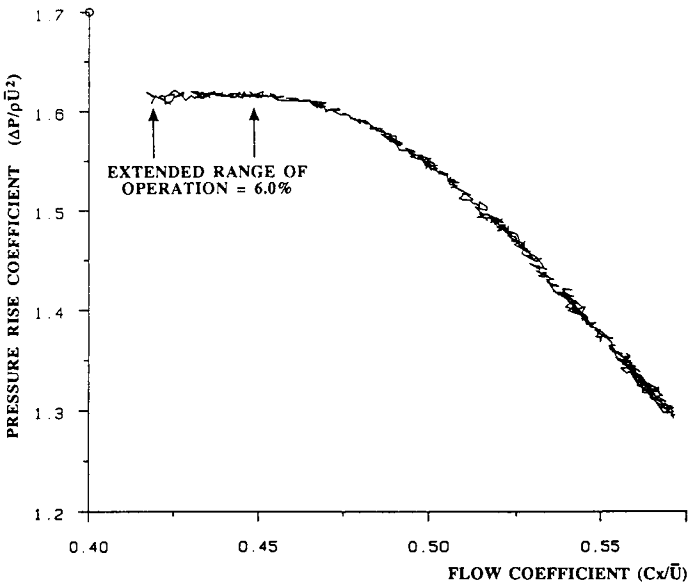


Fig. 11 Automatically recorded compressor characteristic showing the extended range of operation achieved with active stall suppression.

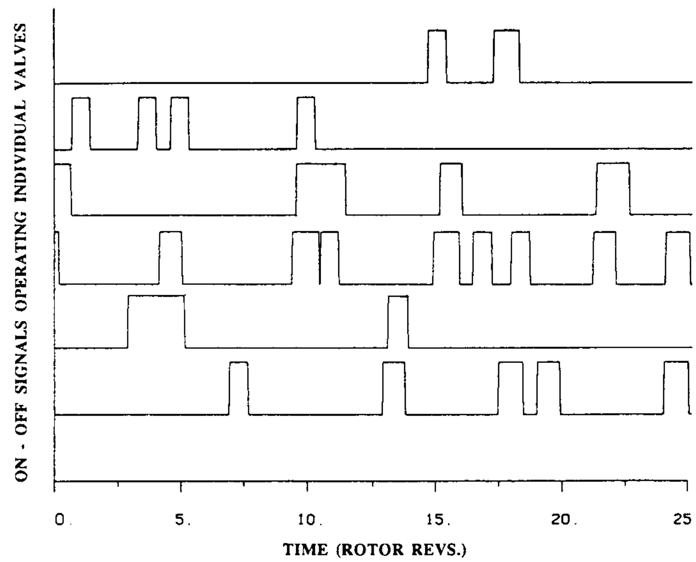


Fig. 12 The on-off signals driving six of the twelve air injection valves during operation beyond the natural stall limit.

There is usually a delay of erratic duration; anything between 2 and 8 rotor revolutions.) When the throttle is moved towards the point where the control system becomes ineffective, the stall cells still forms in the normal way, but their intensity deepens more quickly. Eventually a point is reached where the injected air only reaches the cells once they have already grown too big to be suppressed.

The decrease in stalling mass flow obtained with this type of on/off suppression system was about 6.0%, as illustrated in Fig.11. The characteristic shown was measured by sampling and averaging the output from the pressure rise and mass flow transducers as the throttle was being closed. No filtering was used and the jagged nature of the line is due to normal unsteadiness in the pressure rise and mass flow readings. While a 6.0% stall margin improvement is not particularly useful, it is an encouraging start, especially for a system which has not been optimised in any way and which only has control applied to the first of four stages.The control system does not appear to affect the overall trend of the characteristic, which is extended horizontally. This shows that the action of the air jets is primarily to suppress cell formation rather than to change the basic flow pattern in the machine.

Because the stall cells in the C106 compressor appear as localised disturbances it was hypothesized that the action needed to remove them could also be localised. Using the 12 independently controlled valves and 12 detection probes, a system was configured so that only the valves nearest the stall cell need open at any time. Knowing the typical speed of rotation of the cell, a suitable phase shift was introduced so that no matter where the stall cell first appeared, a valve could be opened in time to coincide with its passing. As before, each valve was automatically closed after a short time if no further disturbance was detected.

Fig.12 shows the opening and closing sequence of 6 of the 12 valves (each alternative valve) when operating in this way. It can be seen that in some cases the action of just one valve was needed to remove a cell whereas in other cases additional valves were brought into play.This localised approach to cell suppression worked well, especially when two valves were opened at a time, 180 degrees apart. (This strategy was employed because it balanced out the one sided effect of using just one valve.) The stall margin improvement was only 5.0%, less than in the previous experiment; but this is not surprising given that there were only 12 discrete injection points as opposed to 60 in the previous case. The localised approach has the advantage of using much less air, there seldom being more than two valves open at any time.

3. The Suppression of Surge

The compressor could be made to exhibit surge rather than pure rotating stall by using a large plenum between the compressor and the exit throttle. Surge is basically the interaction between stall and recovery in the compressor, and the dynamics of the gas stored in the plenum chamber. Two types of surge cycles have been observed; labelled as 'classic surge' and 'deep surge' by Greitzer (1976). Classic surge does not involve any significant reversed flow in the compressor while deep surge does. The C106 system only exhibited deep surge, and therefore on experiments were possible on classic surge. It should be mentioned, however, that deep surge is the more prevalent type, occurring at the high speeds and pressure ratios used in aero-engines.

Measurements from a typical surge sequence are shown in Fig. 13 where the plenum pressure and the output from a single hot wire at the front of the compressor, are plotted. At the left of this figure the compressor is operating unstalled and the throttle is slowly being closed, thus moving the operating point towards the surge line. At time t = 25 the compressor goes into rotating stall. In doing so it loses its ability to support pressure rise and the plenum air blows back through the compressor. This process eliminates the stall cell and sets up a (temporary) regime of axisymetric reversed flow. The plenum pressure then drops rapidly until a point is reached at which the compressor can re-establish forward flow and start filling the plenum again. The pressure in the plenum then rises back to peak value where the compressor again stalls and the same sequence of events re-occurs.

In Fig.13 the compressor momentarily exhibits rotating stall on two occasions, the first after a period of steady operation (t = 25), and the second after refilling of the plenum (t = 145).The number of stall cells which form on these two occasions may differ noticeably, but in each case the cells are of the type described previously, i.e. small, sharply defined and initially rotating at about 70% of rotor speed. A single stall cell is most likely to form at the start of the first cycle while groups of multiple cells are more likely thereafter. In each case, however, the surge cycle is initiated by this burst of rotating stall. An example is given in Fig.14 where the stalling and recovery phase of a typical first cycle is presented. Six hot wires were used here to illustrate how, at the start of the surge cycle, a single stall cell forms and grows and then decays as the annulus becomes engulfed in reversed flow.

The above description of surge inception shows that rotating stall is always the precursor of a surge cycle. Any active suppression technique which delays the onset of stall will therefore also be effective in delaying surge. Experiments carried out with the same stall cell

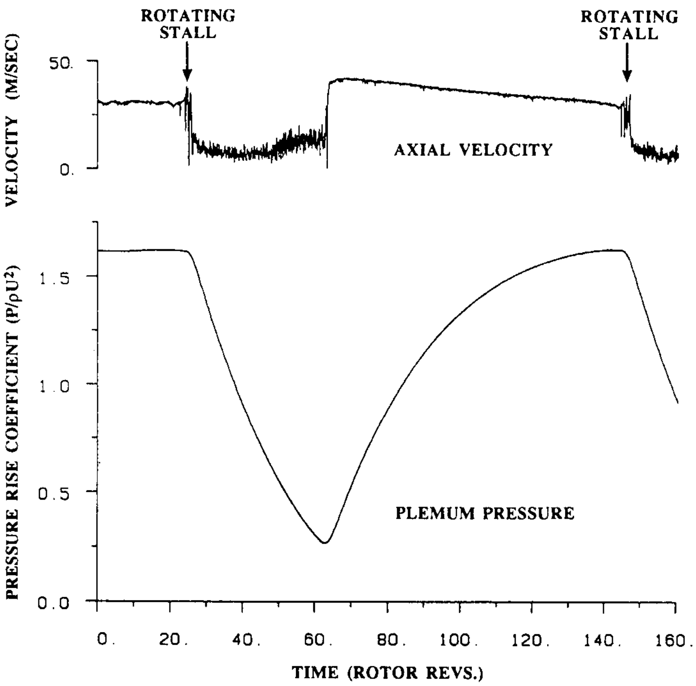


Fig. 13 Measurements of axial velocity and plenum pressure during a typical surge cycle.

detection and elimination procedures described above showed a mass flow decrease at instability of 5 or 6%, just as with pure rotating stall. If the compressor is allowed to surge without control and the throttle position is unchanged, repeated surge cycles will occur, each one being triggered afresh by rotating stall. Switching on the stall suppression system while the compressor is in such a repeating sequence should therefore arrest the next surge cycle and hold the compressor in a stable condition. The results of an experiment to demonstrate this point are shown in Fig.15, where the output from a

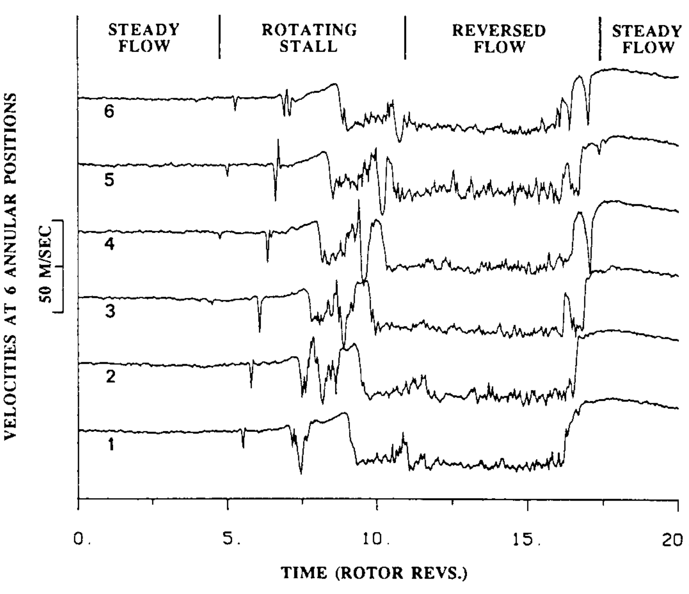


Fig. 14 Hot-wire measurements showing stall cell development and reversed flow during a typical surge cycle.

single hot-wire is displayed on a compressed time scale. The system can be seen to go through two surge cycles before the controller is switched on. In this case, the 12 trapdoor valves were used, and after being triggered to open by cell formation in the compressor, all the valves were held continuously open for 10 seconds before being closed. The reason for doing this was to be sure that the action of the valves is purely to hold off the formation of stall cells rather than to interfere with the dynamics of the surging process. Using the control system in the stall suppression mode to open and close the valves automatically, either all at once or one at a time, also worked successfully. To highlight the effectiveness of the control system, a horizontal line has been drawn in Fig. 15 to indicate the through-flow velocity at which the compressor first started to surge. During the first cycle the throttle was closed a bit further to show that the system could be stabilised at a lower flow rate lower than is normally possible. The surge sequence was therefore arrested even though the compressor operating point was to the left of the usual surge boundary.After a period of 10 seconds the controller was switched off and the surging sequence began again.

(In Figs. 13 and 15, the hot-wire velocity traces do not go below zero even though other measurements confirm that reversed flow does occur during part of the surge cycle. This is because the hot-wire system sees only the modulus of the velocity and so the negative velocity component is returned as a positive value. The change over from forward to reversed flow occurs during the period of rotating stall and therefore no clear zero velocity point is seen.)

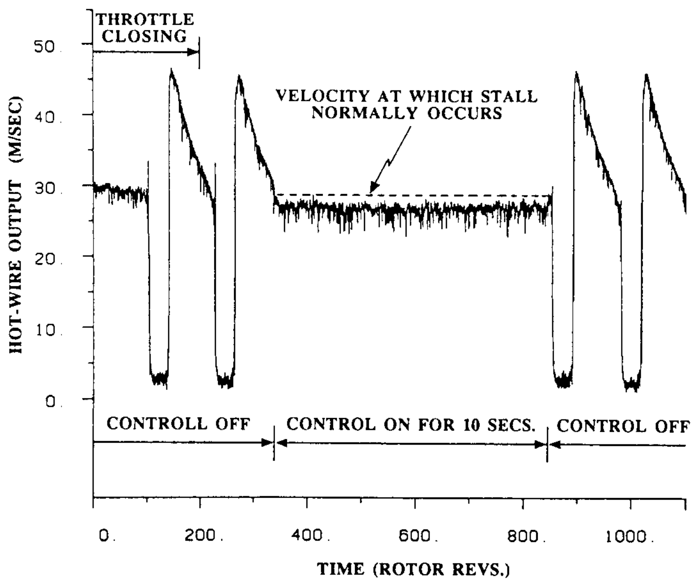


Fig. 15 Example of the use of stall suppression to stabilise a surging compressor. (Control system switched on for ten seconds only.)

DISCUSSION

The work presented here was started without formal ideas of how active suppression of stall should be implemented. The C106 compressor was chosen as the test vehicle and the subsequent experiments were based solely on the observed stalling behaviour of the machine. At the start it was unknown whether the C106 compressor might be unique in the way that it stalled. Additional work on stall inception by Day (1991) on other compressors, and recent measurements from a real engine have, however, shown that the behaviour of the C106 is representative and that the stall suppression techniques developed here can be applied in other situations as well.

In terms of stall inception, flow breakdown in the C106 compressor is always initiated by the abrupt appearance of a small localised disturbance which at first rotates at about 70% of rotor speed. The disturbance usually covers just the tip region of four or five blade passages on the first rotor. In some instances, prior to the formation of this kind of disturbance, a more global perturbation of the flow through the compressor is detected. This perturbation, usually referred to as a mode, rotates at a comparatively low speed (20% of rotor speed) and does not itself grow to become a finite stall cell. Instead the trough of the perturbation wave appears to provide a starting point for the formation of a localised disturbance. It is this localised disturbance, and not the the modal perturbation itself, which is responsible for the collapse of the pressure rise.

In response to this picture of stall inception, two basic experiments were performed; the first to damp the modal perturbation so as to remove its promotional effect on cell formation, and the second to remove the emerging stall cell itself. When modal perturbations occur, damping of these disturbances delays the onset of stall by allowing the average through-flow velocity to decrease without anywhere falling low enough to trigger the formation of a localised stall cell. Whether modal perturbations are present or not, repeated elimination the stall cells themselves also improves the stall margin by removing disturbances which would otherwise lead immediately to a complete collapse of the pressure rise. Ideally these two approaches to stall suppression should be applied simultaneously to obtain maximum benefit. In the current experiment the existing valve configuration, which only has twelve valves, is not versatile enough to operate in both the global and localised mode at the same time. The stall margin improvements which were achieved are nonetheless encouraging.

In discussing the approach adopted here, it should be emphasised that while a large number of compressors stall in the same way as the C106, there are others which do not. Measurements in the Deverson rig in Cambridge by McDougall et al (1990) and Day (1991), and at M.I.T. by Garnier (1990) suggest that in some instances modal perturbations can grow progressively into stall cells, i.e. stall cells may originate on a global rather than a localised scale. The work presented here, although focused mainly on stall cells of small dimension, suggests that air injection as a means of active control would also be applicable in machines which stall progressively. Air injection has the added advantage that in an aero-engine application high pressure air can easily be ducted from the rear of the compressor to suppress any stalling tendency originating at the front of the machine.

CONCLUSIONS

1. The suppression of stalling disturbances using fast acting air injection valves has been shown to be both practical and effective in two different stall onset processes.

2. Modal perturbations of long length scale have been detected in a 4-stage compressor and have been suppressed using feedback of controlled disturbances. In the compressor used here, the modes do not appear to be the primary cause of stall, but rather promote the formation of finite stall cells by producing a localised deficit in the velocity distribution. Suppressing the modal waves produced a 4.0% mass flow rate improvement in stall margin.

3. Short length scale stall cells, emerging without precursive build-up, are of localised extent and can therefore be removed by local, rather than global, action. Elimination of these stall cells has been proved possible and stall margin improvements of 5 or 6% have been achieved in a multi-stage compressor.

4. The need to control long <u>and</u> short length scale disturbances has been demonstrated by this work. The effective application of active control will therefore present a greater chalange than originally thought.

5. The control effort needed to suppress both types of disturbance has been shown to be minimal in comparison with the power of the compressor.

6. Measurements presented here have shown that in the case of surge, each cycle is preceded by a brief period of rotating stall. <u>Stall</u> suppression techniques have therefore been used for the first time to delay the onset of surge, and to suppress surge once a repeating cycle has been established.

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