THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

97-GT-281

STALL INCEPTION AND THE PROSPECTS FOR ACTIVE CONTROL IN FOUR HIGH SPEED COMPRESSORS

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ABSTRACT

As part of a European collaborative project, four high speed compressors were tested to investigate the generic features of stall inception in aero-engine type compressors. Tests were run over the full speed range to identify the design and operating parameters which influence the stalling process. A study of data analysis techniques was also conducted in the hope of establishing early warning of stall. The work presented here is intended to relate the physical happenings in the compressor to the signals that would be received by an active stall control system. The measurements show a surprising range of stall related disturbances and suggest that spike-type stall inception is a feature of low speed operation while modal activity is clearest in the mid speed range. High frequency disturbances were detected at both ends of the speed range and non-rotating stall, a new phenomenon, was detected in three out of the four compressors. The variety of the stalling patterns, and the ineffectiveness of the stall warning procedures, suggests that the ultimate goal of a flightworthy active control system remains some way off.

BACKGROUND

This project was started 1993 with the financial backing of the European Community under the auspices of the "Brite Euram' program. The four compressors to be tested were provided by Rolls-Royce, SNECMA (Societe Nationale D'Etude et de Construction de Moteurs D'Aviation), MTU (Motoren und Turbinen Union Muenchen) and DRA (Defence Research Agency, UK). Experimental co-ordination, and theoretical modelling (reported separately), were provided by the Whittle Laboratory, Cambridge. The project had three specific aims: to learn more about stall inception in high speed compressors; to evaluate various data analysis techniques with a view to obtaining early warning of stall onset; and to assess the range of incoming signals with which an active control system might be expected to cope. The background to each of these objectives is set out below.

The stall inception process

Previous work on low speed compressors has identified two fundamental mechanisms of stall inception; the first associated with modal oscillations of the entire flow field (long lengthscale disturbances) and, the second, with localised disturbances of blade passage proportions called spikes (short lengthscale disturbances). It was initially suggested by Day (1993) that the type of stall inception, either modes or spikes, was determined by the design of the compressor, in particular by the size of the tip clearance gap. According to this thinking, a particular compressor would always stall via the same stall inception process. Recent low speed work, carried out at the Whittle Laboratory in parallel with the current project, (Camp and Day, 1997) has shown, however, that stage matching, and radial flow distribution, are far more important parameters than tip clearance when considering the stall inception process.

In a high speed compressor, the stage matching not only depends on the design of the blading, but also on the speed of rotation of the machine and on the disposition of any variable stator vanes. Apart from some preliminary work by Wilson and Freeman (1994), and Day and Freeman (1994), no systematic investigation has been carried out on the effects of stage matching on stall inception in high speed compressors. The current work thus fills a gap by providing stall inception measurements covering the full speed range from idle to overspeed. Two of the compressors were fitted with variable inlet guide vanes and these were also used to influence the stage matching of the compressors.

All four compressors in this program were comprehensively instrumented, both circumferentially and axially, so that the stall inception process, whether beginning at the front or rear of the compressor, could be studied in detail. The work presented here concentrates on those aspects of the stall inception process which are common to all the compressors and highlights those features which are unique to an individual machine.

Presented at the International Gas Turbine & Aeroengine Congress & Exhibition
Orlando, Florida — June 2–June 5, 1997
This paper has been accepted for publication in the Transactions of the ASME
Discussion of it will be accepted at ASME Headquarters until September 30, 1997

Data analysis studies

Extensive data analysis was carried out during the course of this project with two primary objectives: 1) to assist with the interpretation of the data where important features of the flow are not apparent by eye in the raw data and, 2) to investigate the possibilities of using real time analysis of compressor signals to obtain some form of stall warning indicator. A clearly defined warning signal would make possible the semi-active management of an engine, where fast bleed valves, stator re-scheduling or fuel dipping could be used to pre-empt stall.

The analysis techniques used during the project are described in Section 4, but an evaluation of their respective merits and demerits is not presented here because this paper is primarily concerned with the physical processes associated stall inception.

The prospects for active control

In 1986 Epstein et al (1986) proposed a method of suppressing the onset of stall and surge by actively controlling the stall inception process. Four years later, when the current project was being put together, low speed laboratory experiments were just beginning to confirm that active control might provide useful improvements in compressor operating range; Paduano et al, (1993) and Day (1993). A decision had therefore to be made whether or not to include active control in the scope of the current program. Little information was available at that time on stall inception in high speed compressors and it was decided to concentrate on stall inception with the idea that the results would provide a solid basis for any future work on active control. (A follow-on project has not yet been negotiated.)

Four years have passed since the project began. During this time work on active control has continued in other centres throughout the world, some on low speed machines, Haynes et al. (1993) and Gysling and Greitzer (1994), and some on a working engine Freeman et al (1997). The results so far have been encouraging, but the current work suggests that there is still some way to go before all forms of compressor instability can be controlled.

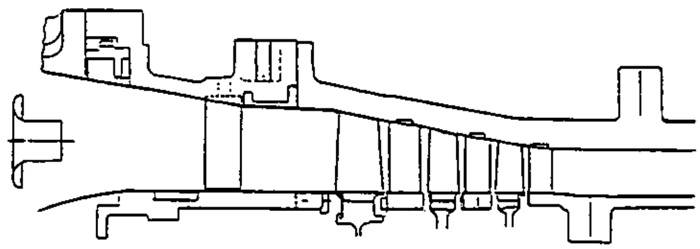
EQUIPMENT

The four compressors used in this study are shown in Figure 1. They are all of different design but each is representative of aero-engine practice:

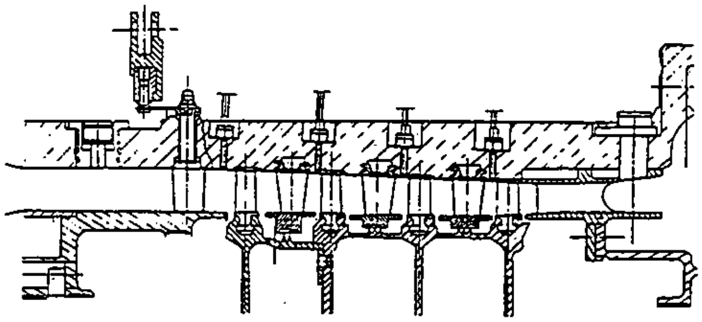
1) MTU Compressor: 3-stage compressor of Intermediate Pressure design, modern blading, relative tip Mach Number of first rotor 1.080 , overall pressure ratio 2.5, 20% true chord inter blade row axial spacing, no variables. (Pressure transducer locations: 8 probes ahead of each rotor.)

2) DRA Compressor: 5-stage compressor of military core design, modem blading, relative tip Mach Number of first rotor 1.001, overall pressure ratio 6.0, 60% true chord inter blade row axial spacing, variable IGVs. (Pressure transducer positions: 4 probes between each blade row and a special fifth probe ahead of each of the first four rotors.)

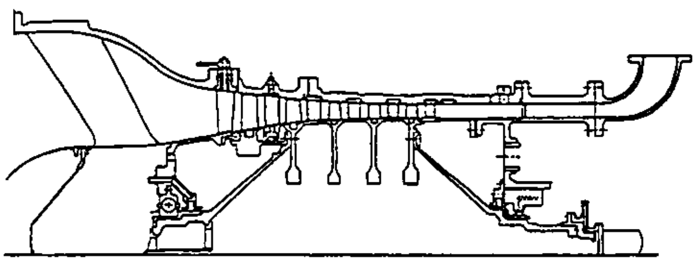
3) SNECMA Compressor: 4-stage compressor of High Pressure design, modem blading, relative tip Mach Number of first rotor 0.798 , overall pressure ratio 2.0, 50% true chord inter blade row axial spacing, variable IGVs. (Pressure transducer positions: 8 probes upstream of the IGVs and 8 probes ahead of each rotor.)



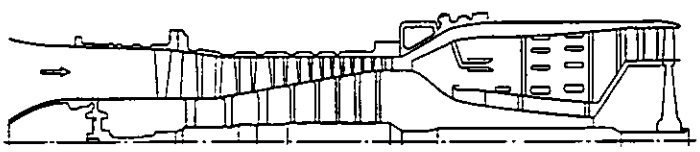
MTU COMPRESSOR



SNECMA COMPRESSOR



DRA COMPRESSOR



ROLLS-ROYCE VIPER ENGINE

Figure 1. The four compressors in the Brite Euram civil core compressor program.

4) Rolls-Royce Viper Engine: 8-stage single-shaft compressor. traditional blading, relative tip Mach Number of first rotor 0.929 , overall pressure ratio 5.0, 30% true chord inter blade row axial spacing, no variables. (Pressure transducer positions: 5 probes ahead of first rotor, 2 ahead of second rotor, 5 ahead of fourth, fifth and sixth rotors, 2 ahead of seventh rotor, and 5 ahead of last rotor.)

The high frequency transducers were arranged in circumferential arrays as equally spaced as mechanical restraints would allow. Higher circumferential densities were possible in the compressors with fewest blade rows. The speed of rotation, mass flow rate, stage-by-stage pressure rise, overall pressure rise and operating temperatures were all recorded using standard test-bed instrumentation.

The data logging was done according to a prescribed routine with the high response transducers coupled in both AC and DC modes. Each compressor was tested at speeds between 50 and 105% design speed with IGV scheduling, inlet distortion and reduced Reynolds Number tests being undertaken where possible. The data from all the tests has been archived in a standard format database.

REVIEW OF RESULTS.

In this section the occurrence of modes and spikes in the four compressors will be considered first. Both types of disturbance have previously been observed in high speed compressors (Tryfonidis et al, 1994, and Wilson and Freeman, 1993) and their appearance here is not particularly noteworthy. Of greater interest however is the role of stage matching in determining which of these instability patterns will occur in a particular situation. Clear trends were observed and are reported here for the first time. The influence of guide vane setting on stage matching, and hence on the stalling pattern, will also be considered. Following this we look at some of the other stall related phenomena that occur in these compressors, such as 'fixed-location stalling' and the occurrence of high frequency stall - both new phenomena. The effects of inlet distortion, Reynolds Number and shaft-order perturbations, will also be discussed.

Spikes and modes

Reference has already been made to the distinction between modal type stall inception, where the flow becomes unstable due to a large scale circumferential oscillation, and spike type stalling where flow breakdown originates from a distinctly localised disturbance. In general the physical features of the these processes can be summarised as follows: Modal perturbations revolve comparatively slowly, up to about 50% of rotor speed, and appear as gentle waves or undulations in the pressure traces. (Work by Hendricks et al (1996) suggests, however, that modes of higher frequency can exist in high speed machines.) Spike type disturbances, on the other hand, are characterised by sharp peaks or "glitches" in the time traces. As a general rule spikes rotate fairly fast when first detected, 70 to 80% of rotor speed, and slow down quite rapidly as they grow in size. Representative examples of modes and spikes from the current project are shown in Figures 2 and 3. In Figure 2 the modal perturbations rotates at 33% of rotor speed while the spikes in Figure 3 rotate at 70% of rotor speed.

The localised nature of spikes also means that the initial disturbance has a limited detection zone in the axial direction. An example from the SNECMA machine is given in Figure 4 where a spike is visible on rotor 1 almost a full revolution before it is detected by the probes in the second stage. A similar situation also occurs at low speed in the DRA compressor. The early detection of spikes is thus dependent on the positioning of the measuring probes.

The examples given here suggest that modes and spikes represent two distinctly different stalling mechanisms. This idea is supported by

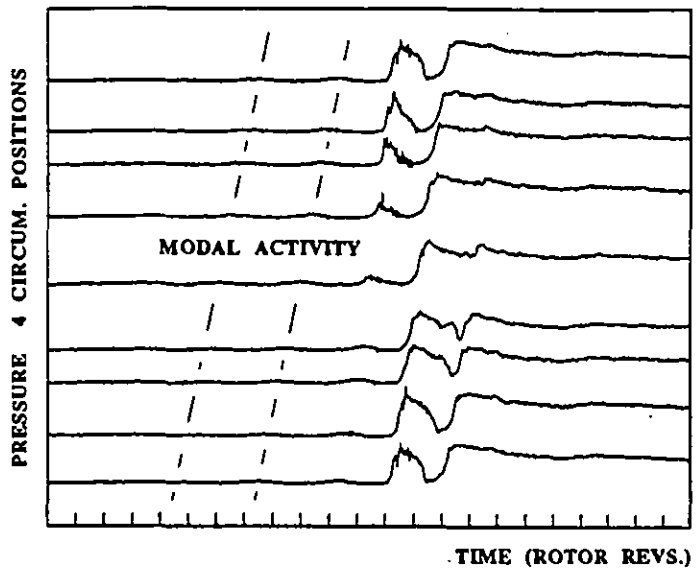


Figure 2. Example of modes in a high speed eompressor taken from the MTU measurements.

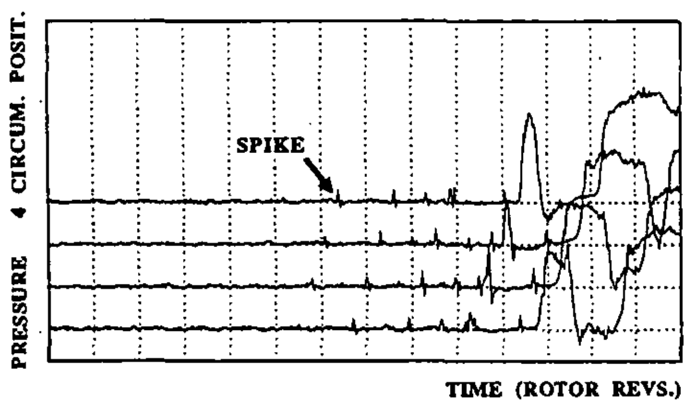


Figure 3. Example of spikes in a high speed compressor taken from the DRA measurements.

recent low speed work by Camp and Day (1997). This work also shows that both these disturbances can occur in the same machine and the questions now to be answered are whether or not this is true for high speed machines, and which compressor operating parameters are responsible for determining the type of stall inception which will occur.

Compressor Matching

The work of Camp and Day (1997) shows that both modes and spikes can occur in the same compressor and that stage matching is one of the parameters which can bring about a change from one stalling pattern to the other. In a high speed compressor the stage matching changes automatically as the speed of rotation changes. Compressibility effects mean that the position of highest loading shifts from the front to the rear of the compressor as the speed of rotation increases. At low speed the front stages are heavily loaded, whereas at

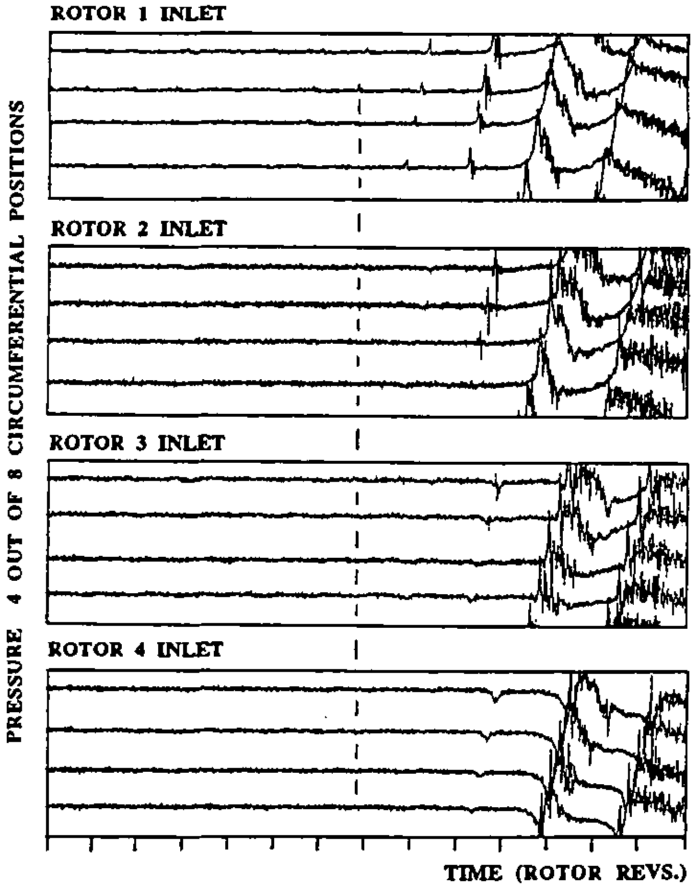


Figure 4. Example of the localised nature of spike stall inception in the SNECMA compressor.

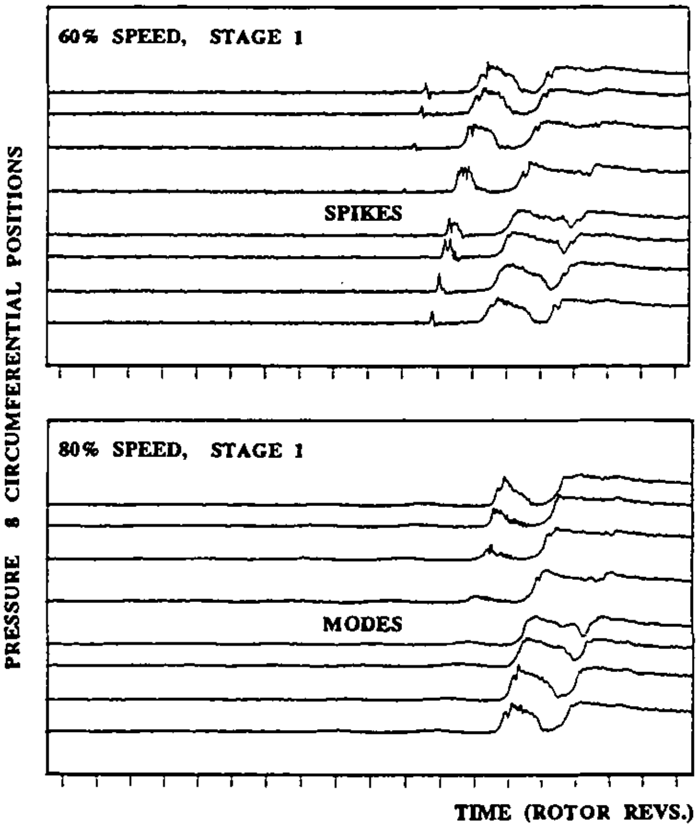


Figure 5. Measurements from the MTU compressor showing spikes at low speed changing to modes in the mid speed range

high speed the rear stages are the most likely to stall. In the medium speed range all the stages are evenly matched near the stall point.

As previously mentioned, spike type stalling is a localised phenomenon beginning as a very small disturbance in just one blade row of the compressor. It might therefore be assumed that spike type stalling will occur at low or high speeds of rotation where individual stages at the front or rear of the compressor are more highly loaded than the rest. At middling speeds of rotation, where all the stages are evenly matched, it might likewise be assumed that modal type stall inception would occur where the perturbations, being of larger proportions, can grow evenly throughout the machine. These ideas about the effect of compressor speed on the stall inception pattern are supported by the current tests.

Three of the four compressors in this study show a systematic change of stalling pattern with change in speed: clear spike-like behaviour at low speeds and modal-type behaviour in the mid-speed range. At high speed, spike-like behaviour is sometimes observed, but this is not always the case. In general the stalling process at high speed develops so quickly that it is difficult to classify the initiation mechanism. A clear example of the changing stalling patterns with a change in compressor speed is from the MTU compressor, shown in Figure 5. Here spike type stall inception occurs at low speed, and modal type in the mid-speed range.

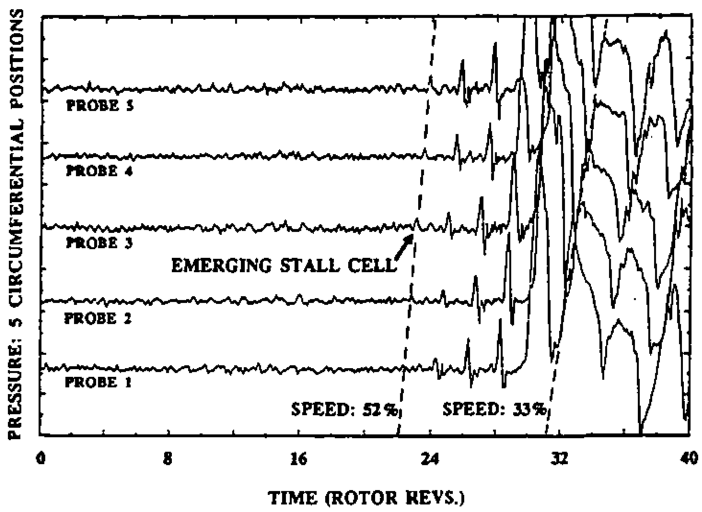


Figure 6. Spike stall inception in the Viper engine at 70% speed.

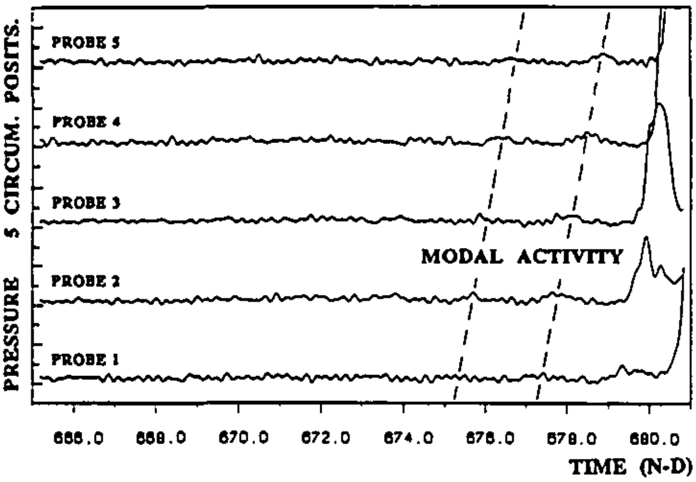


Figure 7. Modal stall inception in the Viper engine at 85% speed.

The Rolls-Royce Viper compressor also exhibits a change from spikes to modes as the speed of rotation increases. An example of spike-like behaviour is shown in Figure 6 at 70% speed and modal-type activity at 85% speed in Figure 7. In previous work by Wilson and Freeman (1994) and Day and Freeman (1994), the existence of modes in this machine was doubted It is only with the additional tests carried out in connection with the current project that modal activity has been discovered in a narrow band between 85 and 87 % speed. Only two clear examples were recorded both showing a modal pattero rotating at 45% speed. This compressor has 8 stages and the limited range of modal activity is ascribed to the narrowness of the speed range over which all 8 stages will be evenly matched. (At 100% speed in this compressor where spikes might again be expected, a rotating pattern does precedes surge, but the structure of the pattern is ill-defned.)

The DRA compressor also shows a clear change from spikes to modes as the speed of rotation is increased, but again the stalling process at full speed is too complicated to be analysed in terms of spikes and modes.

The fourth compressor in the series, the SNECMA machine, behaves differently from the others in terms of the change from spikes to modes as the speed increases. In this case stall is initiated by spikes throughout the whole speed range. The reason for this is not clear, except that this compressor has relatively large inter blade row axial gaps and a low overall pressure ratio. Spike type stall is known to be a localised phenomenon originating in a single blade row and therefore it is possible that this machine, with its widely spaced rows, may favour localised disturbances rather than modal activity.

It has thus been shown that both spike type and modal type stall inception can occur in the same compressor and that the change from one form to the other is associated with a change in rotational speed - and hence with a change in stage matching. The stage matching of a compressor can, of course, be changed independently of speed through the use of variable stator rows. In the current tests, effective changes in matching were unfortunately not possible because none of the compressors were fitted with a sufficient number of variables. Two of the compressors did have variable IGVs but the effects these produced on the stalling behaviour were not straightforward - as is shown below.

Effects of IGV setting

The DRA and SNECMA compressors were equipped with variable inlet guide vanes and tests at different settings were conducted throughout the speed range. In the case of the DRA machine the effect on stall inception of changing the IGVs was not significant, despite large (9°) changes in both the positive and negative directions. This result is somewhat surprising in view of the low speed Whittle Laboratory tests where changes in IGV setting produced a dramatic change in stalling behaviour. (Camp and Day, 1997).

In contrast, the SNECMA machine was very sensitive to changes in IGV setting. This compressor did not exhibit modal activity at any speed and therefore a change from spikes to modes with a change in IGV setting was not expected. However, the IGV setting did have an effect on the axial location of stall inception At the design setting of the IGV's, spikes appeared at the front of the compressor at low speeds and at the rear at high speeds. A small change in IGV settings was however sufficient to reverse this pattern. An example is given in Figure 8 where, at 92% speed, increasing the loading on the first rotor by changing the IGV's by 1 degree was sufficient to switch the point of stall inception from Stage 4 to Stage 1. Changes in stalling pattern were also observed at 87% and 100% speed.

Of the two compressors fitted with variable IGVs, one is thus very sensitive to IGV setting and the other not at all. The reason for this difference is not clear and will require further investigatioa. The anticipated change from modes to spikes, and vice versa, did not occur in either case, but had any of the compressors been fitted with more than one row of variables, the results might have been more revealing.

Stalling behaviour at low speeds

At low rotational speeds, around 40 to 60%, the DRA, SNECMA and Rolls-Royce compressors all exhibited so called "front-end" or "start-up" stall. This refers to a stable stall regime which

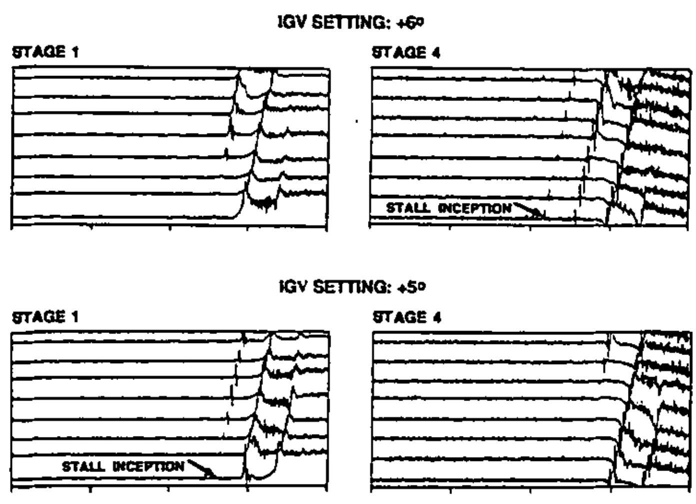


Figure 8. Example of a small change in guide vane setting affecting the axial location of stall inception.

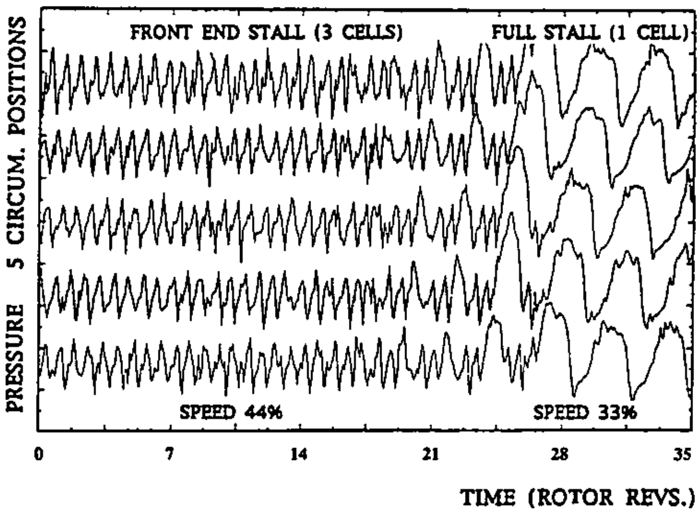


Figure 9. An example of multi-cell front end stall changing to single cell full-span stall in the Viper engine.

occurs naturally when the compressor is first started. The stalling comes about because the front stages of a long compressor suffer from low through-flow velocity at low rotational speeds and the blockage created by the stall cells is a self correcting mechanism for reducing the frontal area of the compressor and thus improving the matching.

Front-end stall is usually confined to the first few rows of a long compressor and the stall cells are nearly always of the part-span type. Multiple cells are a signature of this type of stall and the cells organise themselves in an equally spaced circumferential pattern. If additional throttling occurs while these cells are present, perhaps due to over fuelling in an engine, the cells coalesce through a merging process giving rise to a single stall cell of larger proportions. Figure 9 from the Viper compressor is a good example of three cells coalescing into one. While the cells remain in their multi-cell form, they are relatively benign and do little harm to compressor or engine performance. The larger full-span cell, when formed, has a bigger effect on compressor performance because it spreads throughout the length of the compressor and engine shut-down is necessitated. Under normal run-up conditions, where the part-span cells are not forced to coalesce, the cells shrink and disappear as the compressor speed increases and the front-to-back matching improves.

Only the MTU compressor in this series did not exhibit this type of stalling behaviour at low speeds. This machine operates without stall at 40% speed and when flow throttling is applied, a spike type pattern is observed prior to single cell rotating stall. Because front-end stall is a consequence of part-speed mismatching the effect is worst in long compressors. The MTU compressor has only three stages and therefore the first stage in this machine may not be sufficiently mismatched to sustain front-end stall.

Stall inception at full speed

Each of the four compressors stalls in a different way at high speed and it is difficult to make a clear statement of generic trends. All the compressors surge at this speed and in most cases the surge event is preceded by a brief period of rotating stall. The stalling process occurs very quickly - less than three rotor revolutions separate the time

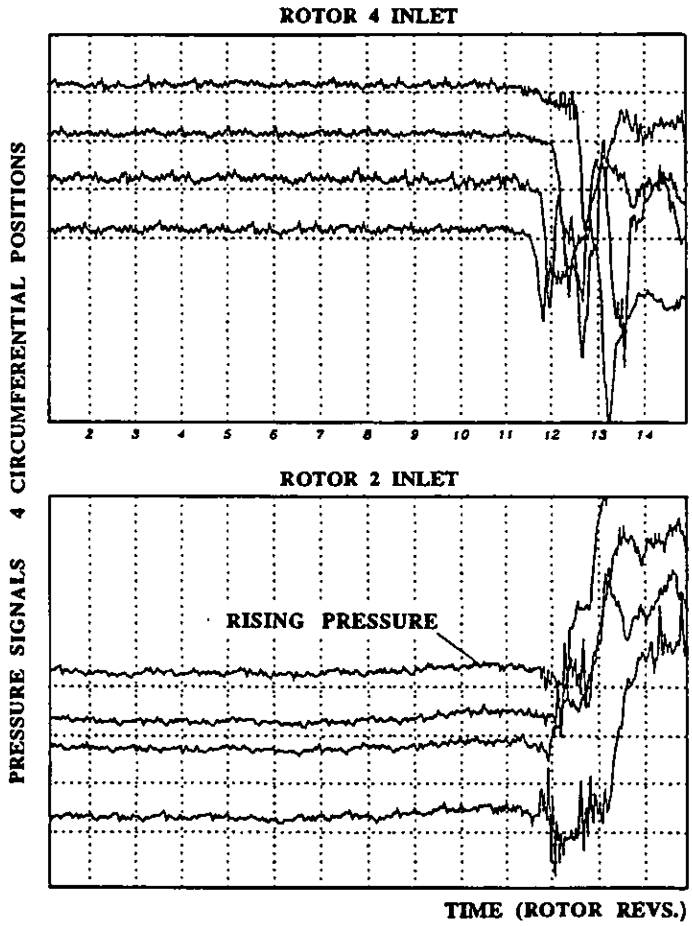


Figure 10. Non-rotating pressure ramping prior to stall at full speed in the DRA compressor.

at which the stall cell first appears and the time when reversed flow is fully established. The three rotor revolutions here should be contrasted with about five at lower speeds where the duration of each revolution is itself longer in time. Surge inception at high speed is thus a very rapid process. This is especially so in the DRA compressor where 10 milliseconds divides axisymmetric forward flow from axisymmetric reversed flow.

In all of the compressors, flow breakdown at high speed seems to begin at a fixed circumferential position in the annulus. This is certainly true for all the SNECMA tests where the spikes preceding stall always appear at the same circumferential position, as confirmed by Escuret and Garnier (1996). In the other compressors, the flow breakdown process is less well defined, but still the impression is that the breakdown process always begins at one particular point on the casing.

Circumferential bias is also a feature of another interesting phenomenon which has been observed for the first time in these tests. In some cases the compressors show signs of protracted, non-rotating pre-stall flow disruption. In the DRA, Rolls-Royce and MTU compressors, a gentle rise in pressure over a period of three or four rotor revolutions sometimes appears prior to stall. This rise is like a slow ramp and can occur over a wide circumferential area, even axisymmetrically. An example of this type of 'fixed-location stall' from the DRA compressor is shown in Figure 10. Here the rise in pressure appears in the forward stages of the compressor all round the circumference. There is nothing to suggest any alteration in flow conditions further back in the compressor - as can be seen from the steadiness of the pressure traces at rotor 4 inlet. In the other compressors the ramping sometimes occurs over a narrower circumferential arc, as in the Viper compressor where only one pressure transducer out of six records the rise, Figure 11. In this case the ramping occurs in the middle stages of the compressor and has no detectable effect at inlet or exit. (Further work is necessary to establish the physical structure of this type of disturbance.)

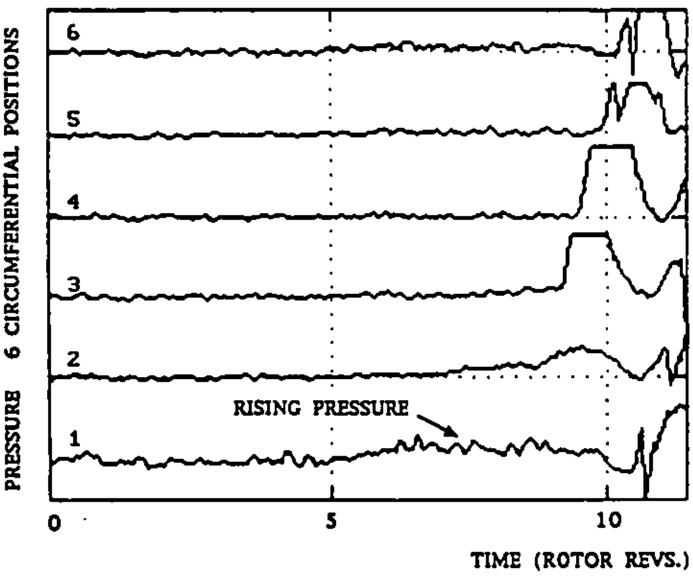


Figure 11. Fixed location pressure ramping in the fourth stage of the Viper compressor at full speed.

Considering all four compressors, the stalling behaviour of each machine can be summarised as follows: The MTU machine shows signs of both modes and spikes. When modal activity appears the speed of rotation of the disturbance is so slow that it could just as well be described as fixed-location pressure ramping. In the DRA compressor, the breakdown process takes place very quickly at the end of a period of fixed-location stall. In Figure 10 for example, the actual breakdown of axisymmetric flow takes place in just one rotor revolution. Other examples from this compressor show that the breakdown process can occur even quicker than this. The SNECMA compressor exhibits spike type stall even at high speeds. In the Viper compressor rotating stall precedes surge, but the actual stalling process is hard to categorise. Fixed-location pressure ramping often occurs before stalling takes place.

Cargill and Freeman (1991) suggested that a blast wave spreading forwards through the compressor could occur at high pressure ratios. In the MTU, SNECMA and Rolls-Royce compressors there is a clear period of rotating stall prior to flow breakdown and therefore blast wave effects are not expected to be present in these machines. In the DRA compressor, however, the flow break-down process is almost instantaneous and affects the entire circumference of the compressor, from front to rear, so quickly that sonic disturbances are a possibility. In Figure 10 there is some indication of a sequential flow breakdown process ahead of rotor 4, but this takes place so quickly, in less than one rotor revolution, that an ordinary stall cell could not keep pace. Some of the other stalling events for this compressor are even less structured and occur more abruptly than shown here. While it has been suggested that rotating stall always precedes surge, it is hard to see how this can be true for this compressor where the disturbance spreads faster than a stall cell can rotate.

The above description of the peculiarities of each compressor confirms that stalling at high rotational speeds is a very complicated process. Each compressor behaves differently and not always in the same way from test to test.

High frequency stall

High frequency signals, between 10 and 13 times rotor frequency, were observed during testing of the DRA and SNECMA compressors. For both machines, the disturbances were detected at or near full speed and were audible in the test cell. In the case of the DRA compressor the signal was sensitive to IGV setting. The audibility of the disturbances, and the sensitivity to IGV setting, suggest that data acquisition aliasing is not the origin of the disturbance. An example of the signals from each of the machines is given in Figures 12 and 13. In both cases the perturbations are strongest at the rear, but detectable throughout the compressor. They appear only at the peak of the characteristic, not at higher flow rates, and can last for hundreds of revolutions before stall.

In a compressor with a large number of blades in each row it is possible that signals of higher frequency than rotor passing can be generated by an array of very small stall cells. As a general rule, the smaller the stall cell, i.e. the fewer blades it covers, the faster it will rotate - rotor speed being the upper limit. Very small cells, like the spikes seen during stall inception, often rotate at more than 80% of rotor speed. An interesting example of a multi-cell stalling pattern, obtained during low speed testing at the Whittle Laboratory, is given in Figure 14. Here 11 part-span cells, affecting one out of every five blade passages and rotating at 82% speed, gives rise to a passing frequency of 9 times rotor speed. A similar pattern of 10 small stall cells has also been observed in a high speed industrial compressor, Day (1993a).

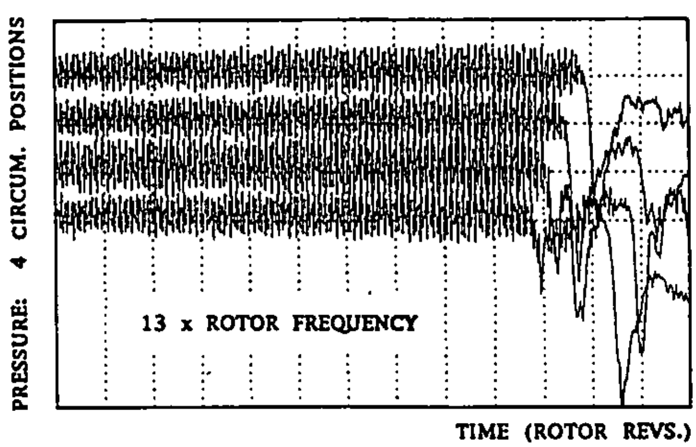


Figure 12. High frequency stall in the DRA compressor at full speed.

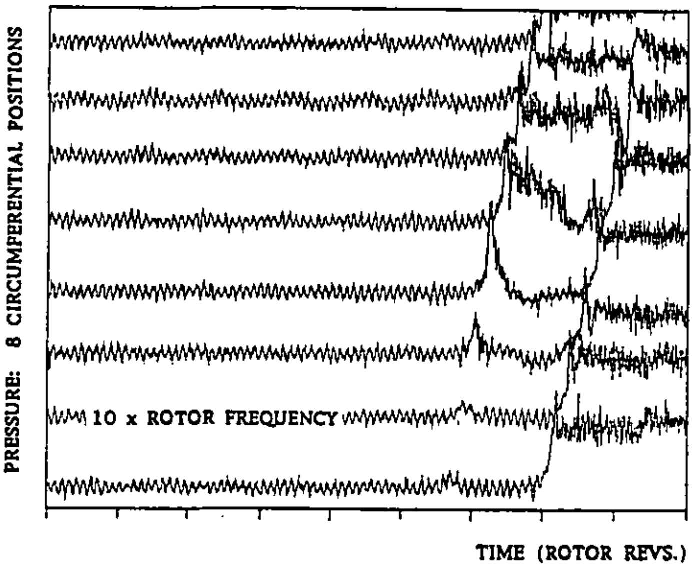


Figure 13. High frequency stall in the SNECMA compressor at full speed.

This example of small stall cells creating a high frequency disturbance seems to fit the SNECMA data very well. To begin with, the data shows that the disturbance is definitely rotating and not axisymmetric. With 70 blades per row and one cell to every five blade passages, as in Figure 14, we get a total of 14 stall cells. If, for argument sake, we assume a rotational speed of 82%, as in the case above, this would give a passing frequency of 11 times rotor speed - which is fairly close to the measured value of 10 times. For the DRA compressor, a plausible explanation is not so easily found. The disturbances at each point around the circumference appears to be in phase and so the disturbance may be axisymmetric rather than rotating. It is possible of course that the number of stall cells is an exact multiple of the number of measuring probes, only 4 in this instance, and so the disturbance would have the appearance of being axisymmetric. There are numerous other possible explanations for these disturbances, eg acoustic resonance or unsteady shock waves. Further work on this topic is necessary, especially because of the possible excitation of blade vibration.

Shaft order perturbations

Shaft order perturbations are present to a greater or lesser extent in all compressors. This is because the concentricity of rotor and casing, the size of the tip clearance gaps and the uniformity of the blade angles are never absolutely perfect throughout the entire compressor. All four machines in this study show signs of shaft order perturbations. The real point of interest here, however, is not the existence of the perturbations themselves, but whether or not the disturbance actually participates in the stalling process. It has been suggested by Hendricks et al (1996) that high speed modal perturbations may lock into shaft order disturbances with the combined effect of triggering stall. The results obtained from the current tests do not openly support this idea, but an in-depth study is in progress.

For the MTU compressor, shaft order disturbances are almost non-existent at the lower compressor speeds but become more prominent towards the top end of the speed range. At low speeds, the

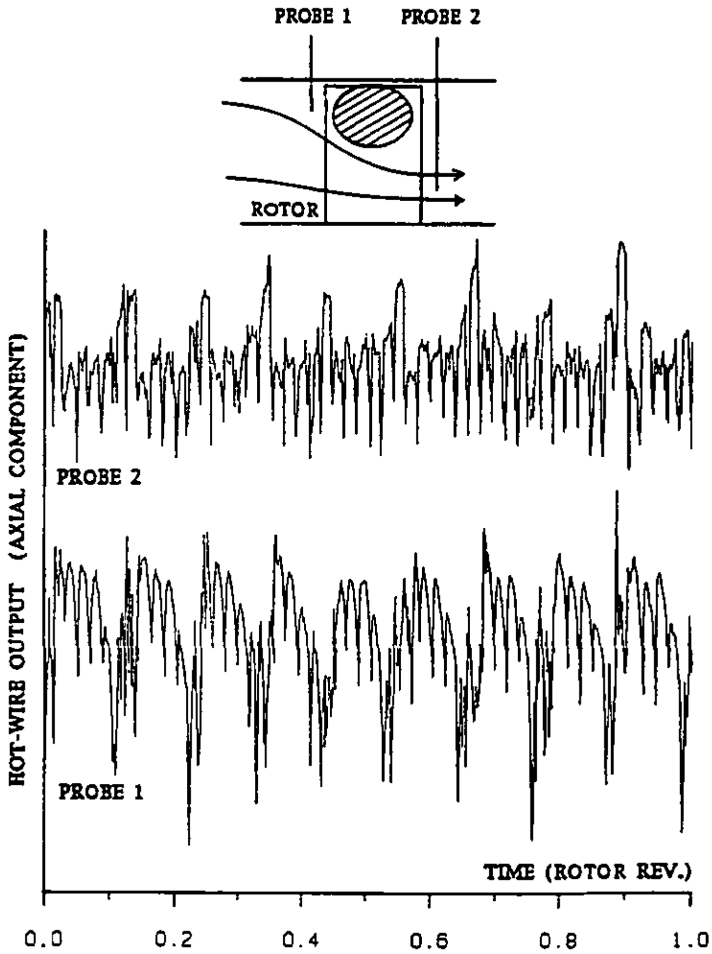


Figure 14. Low speed example of a multi-cell stall pattern in which every fifth blade is affected by a cell one pitch wide.

travelling wave energy analysis of Tryfonidis et al (1995) picks out a small once-per-revolution signal at the front of the compressor. At speeds closer to the maximum, the disturbance increases in intensity and becomes more pronounced at the rear. At full speed, the signal is clear enough to be visible by eye in the data traces. In terms of the growth of the perturbation as stall is approached, the travelling wave energy analysis suggests that the amplitude remains constant for hundred of revolutions before stall - the last twenty of which, at full speed, are shown in Figure 15.

Analysis of the SNECMA and Rolls-Royce data suggests a similar trend with the shaft order signal becoming more pronounced at higher rotational speeds. The DRA machine exhibits slightly different behaviour. In this case the shaft order perturbations are present at all speeds with a slight increase in amplitude at maximum speed. Little or no increase in the strength of the once-per-revolution signal could be detected during the approach to stall, except at 70% speed where a notable ramping of the signal occurred over the last thirty revolutions before stall.

The effects of Reynolds Number

Tests were conducted at MTU with the Reynolds Number reduced to about half the normal value by throttling the incoming flow.

A comparison of the stalling traces for normal and reduced inlet pressures shows no detectable differences in behaviour. The differences between the two sets of results are no more than the usual differences between one test and the next. For these tests the average Reynolds number was reduced from 7 x 10^5 to 4 x 10^5. The latter number is unfortunately still above the threshold at which Reynolds Number usually has an effect on blade row performance. (Tests at the Whittle Laboratory have shown that the stall inception process is not affected right down to values of 0.4 x 10^5. Day, 1993b.)

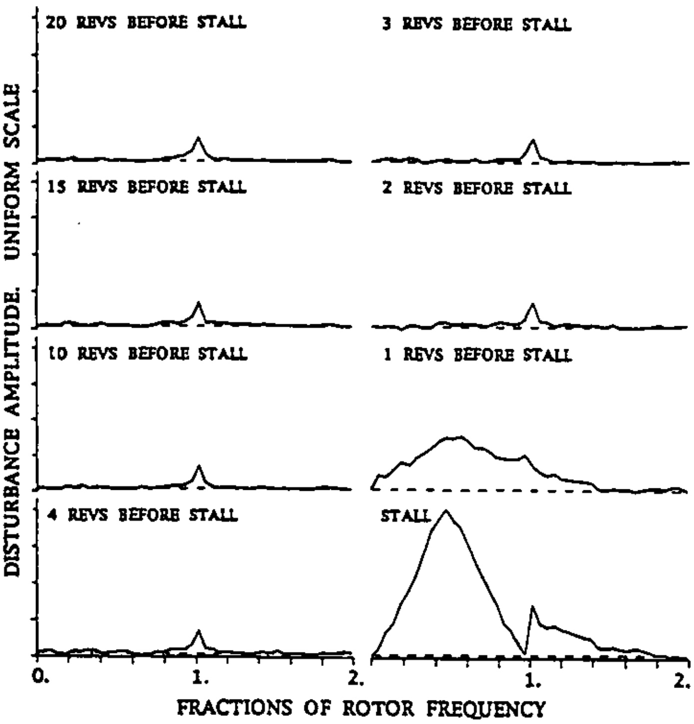


Figure 15. Travelling wave energy analysis of MTU first stage signals at full speed.

Inlet distortion

Both MTU and Rolls-Royce conducted inlet distortion tests. From the Rolls-Royce data it was hard to identify any specific changes in the stalling behaviour, especially at high speed where the stalling process is in any case very difficult to categorise. The MTU tests showed little effect on the stalling pattern at low speed but at full speed the presence of the distortion screen was noticeable.

Without the distortion screen at full speed the MTU time traces show that stall is initiated by a mixture of spikes and modes - the modes sometimes moving so slowly as to be indistinguishable from a fixed-location pressure ramp. When the screen is present, modal activity is greatly increased and the perturbations can be detected many revolutions before stall. The modal oscillations are most pronounced in the middle of the compressor, stage 2, and appear at a frequency of about 42% rotor speed. At the front of the compressor, disturbances of shorter lengthscale, not quite short enough to be described as spikes, appear at regular intervals at the edge of the screen just where the rotor blades leave the distortion, see Figure 16. These short lengthscale disturbances move at about 65% of rotor speed. What appears to happen is that each crest of the modal wave triggers a short lengthscale disturbance which rotates half way around the annulus and then dies out. The next modal crest then sets off another short lengthscale disturbance and so on. In the end, stalling occurs when one of these disturbances becomes big enough to survive a complete revolution around the annulus and then grows to become a fully developed stall cell. This pattern is similar to that described in the work of Longley (1990) and Hoying (1995). The interesting thing in this case is the noticeable increase in modal activity brought about by the distortion screen.

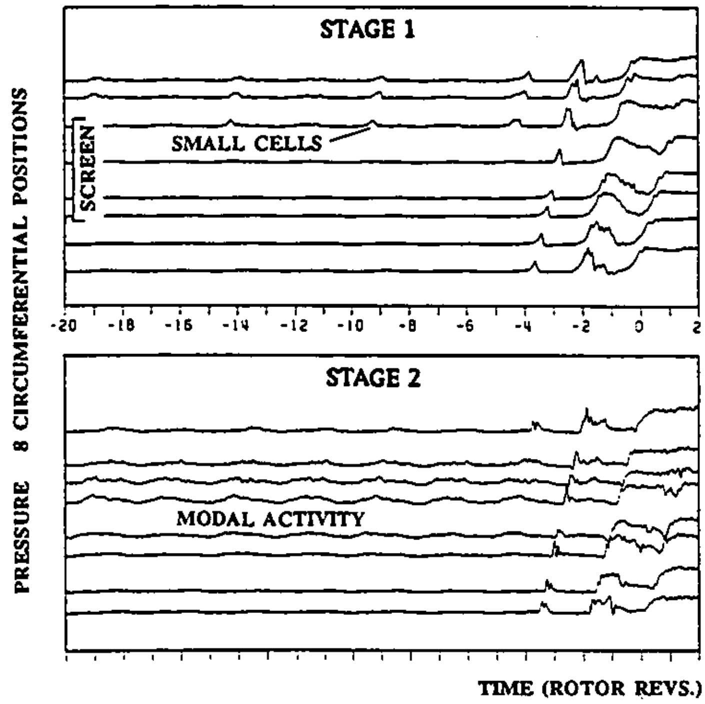


Figure 16. Inlet distortion experiment in the MTU compressor showing modal activity and localised disturbances.

DATA ANALYSIS STUDIES

The data analysis carried out during this project had two distinct objectives: the first to pick out features of the stall inception process which could not be identified by eye and, the second, to examine the possibility of obtaining early warning of the approach of stall. The latter objective, if it could be achieved reliably, would provide a means of initiating corrective action, eg opening bleed valves, before stall actually occurs. This would provide an alternative to active control and would reduce the need for super fast actuators.

In the main, two analysis techniques were used:

1) Spatial Fourier decomposition of modes. This technique was proposed by Longley and used by McDougall et al, (1990). The signals from a circumferential ring of transducers are spatially decompose at each instant of time to yield information on phase angle and amplitude of any circumferential disturbances.

2) Travelling wave energy analysis. This process was first implemented in a stall precursor context by Tryfonidis et al, (1995). It uses a windowing technique to monitor the build-up of the forward propagating wave energy of spatial modes. The objective here is to detect an increase in wave energy, of all forms, as far ahead of stall as possible.

The spatial Fourier technique proved very useful in identifying the physical details of modal activity prior to stall and provided a quantitative means of comparing the data from the respective compressors. The technique is, however, not well suited to the detection of short lengthscale disturbances and is therefore only useful over a limited range of compressor speeds. The travelling wave energy analysis worked very well, and, in the case of the DRA data, was able to point to the approach of stall some 50 to 100 revolutions before the actual event. For this machine, at compressor speeds where spike type stall inception was present, the analysis was still effective but the warning period was reduced. For the other three compressors, the travelling wave energy analysis proved less effective as a stall warning technique and in some cases gave no warning at all - as for example in Figure 15.

Some alternative methods of data analysis were also tried. SNECMA devised a scheme to deal with short lengthscale disturbances using a breadth-and-height filter, Escuret and Garnier (1996). The results are good and particularly useful for their compressor where spike type stall inception is predominant. MTU have developed a parameter identification technique aimed at picking out circumferential harmonics. The results are good, with first order modal perturbations being correctly identified well before stall. Also tried was a noise signal analysis suggested by Rolls-Royce. The technique concentrates on frequencies in the stall propagation range and is often, but not always, able to detect a rise in power level as stall is approached.

DISCUSSION: THE IMPLICATIONS FOR ACTIVE CONTROL

Active control, in the classical sense of the word, makes the use of real time measurements from within the compressor to detect stalling disturbances and to take action to suppress them. The response of the system is thus tailored to the particular disturbance in the compressor at the time. A control system of this type must be capable of identifying and responding to all forms of stall inception. Any mis-interpretation of the incoming signal will result in inappropriate instructions which could cause the compressor to stall prematurely. The purpose of the current work is to relate the physical disturbances in the compressor to the signals received by the control system - the idea being that improved understanding of what happens in the compressor will lead to improved control systems.

It has been shown that both modal and spike type stall inception occur in high speed compressors. The structures of these disturbances are fairly well understood and algorithms are available to detect their presence. Of the two types, modal perturbations are the easiest to detect because they affect the whole compressor at once and therefore all the measuring probes around the compressor contribute towards identifying the disturbance. The more probes there are, the more reliable the detection process will be. The work on the four compressors tested here shows that modal activity is most likely to occur in the mid-speed range.

By contrast, spike type disturbances are most often present at low rotational speeds. In principal these disturbances are more difficult to track than modes because of the localised nature of the disturbance. The disturbance appears to originate from flow breakdown in a particular blade passage and while the disturbance is small it has no measurable effect anywhere else around the circumference. Out of a circumferential array of probes, only one probe at a time will be affected by the disturbance, making early detection above the background noise very difficult. In cases where a limited number of probes is deployed, it is even possible for a small disturbance to remain undetected while it travels from one measuring position to the next. As above, this problem can be reduced by using more sensing probes.

Apart from the modes and spikes above, there are other forms of instability in high speed compressors which could have an influence on the reliable application of active control:

1) Front-end start-up stall. This usually consists of multiple stall cells which give rise to a high passing frequency - perhaps 1.5 times rotor frequency. The disturbance itself is benign and over a short period of time does not harm the compressor and does not hinder acceleration to higher speeds. From an active control point of view, no corrective action is required in this case, but the control system must remain active to prevent these small cells from merging into a single cell which could stall the compressor.

2) High frequency stall. This type of disturbance was detected in the DRA and SNECMA compressors, Figures 12 and 13. Here the high frequency signals appear just before stall at full speed and actually mask the start of the surge event. The disturbance produces frequencies well above shaft speed and these can be of sizeable amplitude. A successful control system will be expected to ignore these signals but will have to remain tuned to the underlying development of any surge event.

3) Abrupt stalling at full speed. In most of the tests reported here, some form of rotating stall is detected prior to surge. Active response to this type of disturbance is possible with modern actuators, Freeman et al 1997. In the case of the DRA compressor, however, the flow becomes unstable so quickly, in less than 1 rotor revolution, that there appears little hope of suppressing the event.

4) Fixed-location stall. When a localised pressure ramp precedes a surge event, as shown in Figures 10 and 11, the increased warning time may make it easier to sense the onset of instability - but whether or not this type of non-rotating pressure rise will respond to actuation is not known. Further work is necessary for a better understanding of this newly observed phenomenon.

5) Shaft order disturbances. This type of disturbance is present to a greater or lesser extent in all compressors and may or may not play a part in the stalling process. Because the frequency of the disturbance is precisely defined, it can, if necessary, be ignored by the control system.

6) Distorted inlet flow. Where inlet distortion is present, due either to nacelle separation, or to hot gas ingestion, the control system may receive signals of mixed type, i.e. spikes from one side of the compressor and modes from the other. To cope with this situation, the control system may require additional information from sensors external to the compressor to help identify the problem.

The list above gives some idea of the variety of signals which may be observed during the operation of high speed compressors. An individual compressor may not, of course, be susceptible to all of these problems, but it is useful to have foreknowledge of the possible range of signal which might be encountered - after all, changes in compressor behaviour do occur during the life-time of an engine and therefore all contingencies should be covered.

An alternative approach to the active control of stall is to disregard the physical details of the flow generating the incoming signals and to simply concentrate on deriving a stall risk index from the sum of all the inputs. The reliability of the index would be essential for a scheme like this, but as yet no system has been found which can provide a suitable signal under all operating conditions - the best scheme so far is the technique suggested by Tryfonidis et al, but this did not work well in all instances. Either way, active control or corrective control, the goal of building a flightworthy control system still seems to be some way off.

SUMMARY AND CONCLUSIONS.

At the start of this project some stall inception information from high speed machines was available in the literature, but gaps in the test ranges and the inadequacy of the instrumentation used, means that very little could be gained from studying this data. A test program covering a range of compressor designs, and using dedicated instrumentation, was therefore set up to obtain the type of detailed information required.

- Both short lengthscale (spikes) and long lengthscale (modes) disturbances were detected in three out of the four compressors. A clear trend was observed with spikes appearing at low rotational speeds changing to modes in the mid-speed range. (This trend is in line with the ideas of Camp and Day (1997) in terms of the influence of rotational speed on changes in stage matching.)

- A change of inlet guide vane setting will also change the stage matching in a compressor. The changes possible in the current series of experiments were not sufficient to cause a switch from short to long lengthscale disturbances, but in one compressor changes in the IGV setting certainly did affect the axial position at which short lengthscale disturbances first appeared.

- A new type of high frequency disturbance was observed for the first time in two of the compressors when operating at full speed. This disturbance is thought to be the result of multiple part-span stall cells. (The findings are supported by similar observations in a low speed compressor.)

- At full speed in three of the compressors, a non-rotating, fixed-location, disturbance was detected just prior to stall. The disturbance extends over three or four rotor revolutions and takes the form of a gradual upward ramping of the pressure signals over part or all of the compressor annulus. The disturbance is usually limited to a particular axial location. There are no known prior references to this type of stalling behaviour.

- In most cases rotating stall preceded surge at full speed, but the the origins of the stall cell were not clearly identifiable in terms of modes and spikes. The growth of the stall cells also took place over fewer rotor revolutions than at lower operating speeds.

- In one of the compressors, flow breakdown at full speed occurred so quickly that rotating stall could not be detected before the event. The symmetry of the flow throughout the compressor was disrupted in the space of one rotor revolution. Fixed-location pressure ramping usually preceded the event.

- In line with previous low speed experiments, tests on the effect of Reynolds Number on stall inception showed no noticeable effect.

- Shaft order perturbations were detected in all the compressors and the intensity usually increased with the speed of rotation. At any particular speed, no consistent increase in the intensity of the disturbance could be detected as stall is approached.

- The imposition of inlet distortion was found to have little effect on stall inception patterns at low and intermediate speeds. It did, however, amplify the presence of modal perturbations at full speed in one of the compressors. The modes themselves triggered short-lived spikes which meant that short lengthscale disturbances were detected on one side of the compressor and long lengthscale disturbances on the other.

- Various data analysis procedures were tried during the course of the project. Most were helpful in interpreting the data, but all failed in the search for a reliable stall warning signal.

- From an active control point of view, the results show that just being able to detect modes and spikes will not be sufficient in a high speed compressor. A whole range of other low and high frequency disturbances need to be taken into consideration.

ACKNOWLEDGEMENTS

Grateful acknowledgement is made of the funding for this work from the European Community's Brite Euram Civil Core Compressor Program. The financial participation of MTU, SNECMA, the Defence Research Agency and Roll-Royce is also gratefull acknowledge. Not to be forgotten are the efforts of the support groups in each of the above organisations.

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