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THE UNSTABLE BEHAVIOR OF LOW AND HIGH SPEED COMPRESSORS

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ABSTRACT

By far the greater part of our understanding about stall and surge in axial compressors comes from work on low-speed laboratory machines. As a general rule, these machines do not model the compressibility effects present in high-speed compressors and therefore doubt has always existed about the application of low-speed results to high-speed machines. In recent years interest in active control has led to a number of studies of compressor stability in engine type compressors. The instrumentation used in these experiments has been sufficiently detailed that, for the first time, adequate data is available to make direct comparisons between high-speed and low-speed compressors. This paper presents new data from an eight-stage fixed geometry engine compressor and compares this with low-speed laboratory data. The results show remarkable similarities in both the stalling and surging behaviour of the two machines, particularly when the engine compressor is run at intermediate speeds. The engine results also show that, as in the laboratory tests, *surge* is precipitated by the onset of *rotating stall.* This is true even at very high speeds where it had previously been thought that surge might be the result of a blast wave moving through the compressor. This paper therefore contains *new* information about high-speed compressors and confirms that low speed testing is an effective means of obtaining insight into the behaviour of high-speed machines.

INTRODUCTION

Although stall and surge have been one of the most persistent problems in the history of aero-engine design, the study of these disturbances in high-speed machines has received relatively little attention. There are two reasons for this, firstly, high-speed testing is expensive and, secondly, there is a natural reluctance to study a phenomenon which is not part of successful compressor design. Much of the work on stall and surge has thus been done in the laboratory at low speed.

By testing at low speed, the implicit assumption is made that compressibility is not an important factor and the results can be read across to high-speed machines. Until now there has not been a systematic study to prove that this assumption is correct. Some high-speed data does exist, e.g., Mazzawy (1980), Riess et al. (1987), Small and Lewis (1985) and Hosny and Steenken (1986), but, in general, the effects of compressibility have not been studied in any detail.

In historical terms, the experimental work on low-speed machines, and particularly the little which has been done on high-speed machines, has concentrated on the effects of the fully developed disturbance. Surge has been studied to assess the effects of reversed flow on structural loading, and stall, to examine recovery problems and fatigue stressing of the blades. In recent times, however, the emphasis has shifted away from the study of the fully developed disturbance to concentrate on how these disturbances came into being, i.e. the inception process.

Epstein, Ffowcs Williams and Greitzer (1986) published a paper suggesting that active control could be applied to stall and surge to extend the useful operating range of the compressor. The implementation of active control relies on the idea that if stall and surge can be shown to begin from small perturbations it should be possible, by the introduction of artificially induced "friendly" perturbations, to create conditions in which the original perturbation can be held in check. This concept has therefore prompted a whole new series of experiments on stall and surge inception. Laboratory tests show that the instabilities do indeed begin as small perturbations and that the inception process is ordered and well defined, Day (1991a). This work in now being followed up by detailed experiments on engine compressors to see if this is also true at high speed. It is the first of these experiments which is presented here to give a back-to-back comparison of stall inception processes in low and high-speed compressors.

STALL AND SURGE

In general terms, rotating stall is a disturbance which affects the flow in the region of the compressor blading. The disturbance, once initiated, rotates about the compressor causing severe blade vibration and, above all, a drop in overall pressure rise. In an engine, as opposed to a compressor rig, rotating stall, in all but it's minor forms, will seriously restrict the flow into the combustion area leading to overheating and deceleration of the engine. Restarting of the stalled engine is then not possible until the stall cell has been cleared and normal flow conditions restored. This usually requires an almost complete shut down of the engine. In high-speed rigs, where the compressor is driven by an electric motor, blade vibration and temperature rise are the most serious consequences of rotating stall.

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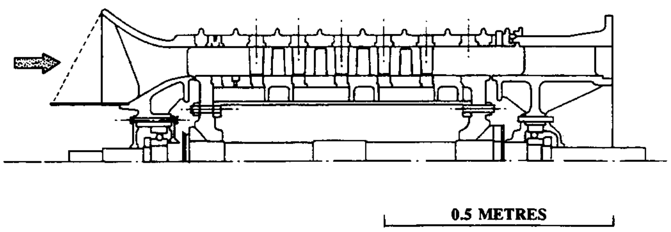
Surge, on the other hand, is an oscillation of the mass flow through the compressor and only occurs where a significant amount of pressure energy is stored downstream of the compressor. This is usually the case in an engine at high speed, where the combustion chambers act as an energy reservoir. In industrial situations this also happens where there is significant duct work at the compressor exit. Because surge only occurs where there is storage capacity downstream of the compressor, it is usually referred to as an "installation" or "system" instability. In the past, a lot of time and energy has been spent modelling surge as a one-dimensional disturbance. Much can be learned using this approach, however, it is becoming abundantly clear that in an axial compressor, surge is actually the consequence of a two-dimensional instability, i.e. rotating stall, Day(1991b). Thus, a compressor may go into stall or surge depending on the speed of rotation and the geometry of the installation, but both types of disturbance are the result of the same fundamental inception process.

In the work which follows, we will examine the general behaviour of a low-speed laboratory compressor and then compare this with the results from the high-speed tests. An engine compressor operates over a wide speed range and it is obviously not practical to consider all possible operating conditions. For convenience the results have therefore been divided into three categories; low, medium and high speed behaviour.

EXPERIMENTAL FACILITIES

The experimental results in this paper were derived from two sources; the low-speed compressor at the Whittle Laboratory, and a Viper compressor tested as part of a complete engine on a test bed at Rolls-Royce.

The Whittle Laboratory compressor, designated the C106, is a low-speed machine of incompressible design having four identical stages preceded by a lightly loaded set of inlet guide vanes. The hub-casing radius ratio is 0.75 and the speed of rotation is 3000 rev/min, giving a rotor Reynolds number of 1.7 x 10^5. The blading is of modern controlled diffusion design and is representative of current HP compressor practice. An outline drawing of the compressor is given in Figure 1.

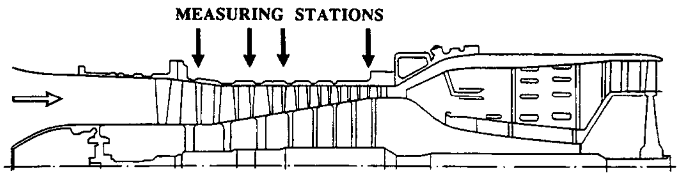


LOW SPEED LABORATORY C106 COMPRESSOR

NUMBER OF STAGES:	4
HUB/TIP RATIO:	0.75
REYNOLDS NUMBER:	1.7x10^5
SPEED OF ROTATION:	3000rpm

Figure 1. The Whittle Laboratory's C106 low speed compressor.

The high-speed compressor used in this study is part of a fully operational Rolls-Royce Viper engine. The engine has a single spool and the compressor is of fixed geometry design, i.e. there are no variable vanes. The speed of rotation from idle to full speed ranges from 4000 to 13850 rev/min, and the hub-casing ratio changes from 0.5 at inlet to 0.85 for the last blade row. The flow rate through the compressor is measured using a calibrated test-bed inlet and standard instrumentation. Details of the engine are shown in Figure 2.



ROLLS-ROYCE VIPER ENGINE Mk 522

NUMBER OF STAGES:	8
SLS THRUST:	3330 lbf
COMPRESSOR PRESSURE RATIO:	5:1
SHAFT SPEED:	13850 rpm

Figure 2. View of Viper engine showing compressor layout and pressure transducer positions.

In the case of the laboratory compressor, the operating point can be moved up and down a fixed speed characteristic simply by changing the setting of the downstream throttle valve. The pressure rise is low and the the flow is basically incompressible. Changing the compressor speed therefore has little effect other than to change the Reynolds Number. In the engine, however, special steps are necessary to change the compressor working line. The engine has neither a physical throttling device, nor the inclination to rotate at fixed speed while other conditions are being varied. The usual method of pushing an engine compressor beyond the stability limit is to use a technique known as fuel spiking. This involves the removal of the normal fuel management system so as to allow the sudden injection of excess fuel into the combustion chamber. The rapid over fueling raises the back pressure on the compressor and, before the shaft has time to accelerate fully, the compressor is driven into stall or surge. For most engines this technique is effective at low and medium speeds, but, at full speed the process leads to detrimental overheating and cannot be used.

In the current experiments, the problem of how to change the working line of the compressor at high speed was overcome by major modifications to the engine. Extra air from an outside source was injected into the combustion area via a ring manifold system with twelve inlet ports. In this way the compressor could be forced to go unstable at more or less fixed speed by suddenly increasing the amount of air being injected, i.e. ramp injection. At lower speeds instability could also be induced by injecting air at a steady rate while the engine is decelerated to a point where the back pressure is more than the compressor can support. Both the ramped injection and steady blowing techniques were used in the experiments reported here.

To study the processes by which the compressor becomes unstable, and the events subsequent to instability, numerous circumferential measuring positions are necessary to pick out the stalling disturbance as it rotates and grows in the circumferential direction. In the laboratory tests, circumferential arrays of six hot-wires were used; hot-wires having the advantage of being able to explore the full height of the annulus. In the engine tests hot-wires could not be used because of the harsh environment and pressure transducers were used instead, five per circumferential array. Unlike the low-speed compressor, flow conditions change along the length of the compressor in the high-speed case and therefore circumferential arrays were necessary at four or five axial stations.

STAGE MATCHING IN A HIGH-SPEED COMPRESSOR

The flow through the laboratory compressor is essentially incompressible and therefore all stages receive air at the same Cx/U and perform in the same way, all moving towards or away from stall at the same time. This is not true for the engine compressor, however, where the annulus is not parallel and compressibility effects mean that prior to instability the stages experience different loading

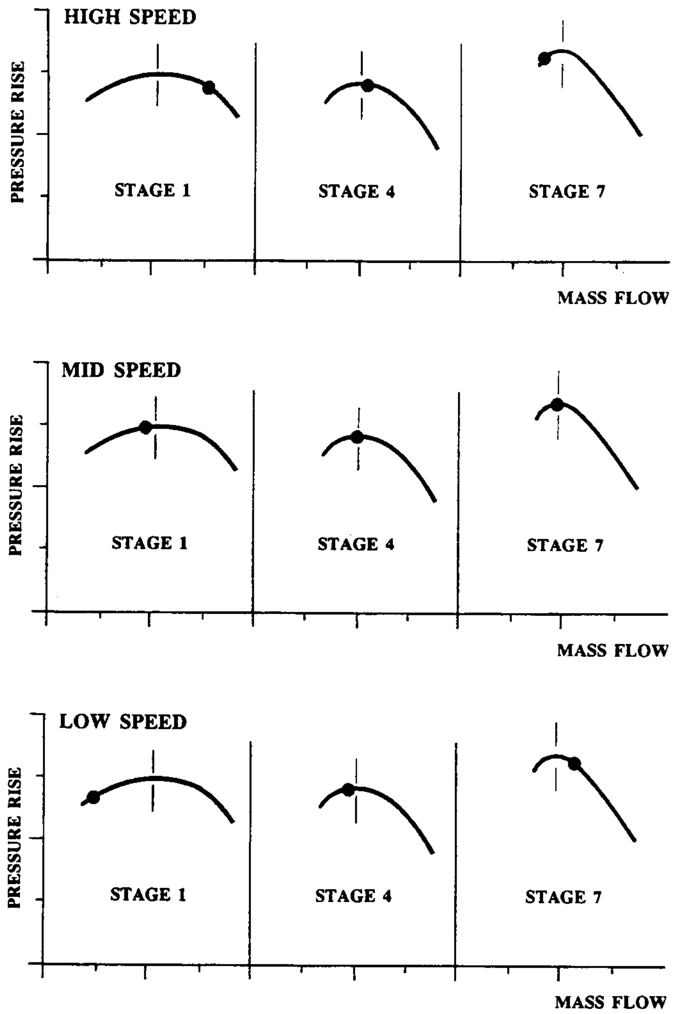


Figure 3. Engine compressor performance predictions showing the operating points for the front, middle and rear stages at low, medium and high speeds.

conditions depending on the speed of rotation of the machine. At low speeds, the air entering the compressor is incompletely compressed in the front stages with the result that the rear stages receive air at a higher than design Cx/U, thus pushing the operating point for these stages away from stall. At high speeds, on the other hand, the front stages compress the air so effectively that the rear stages receive air at a lower than usual Cx/U thus forcing them to operate close to stall. This shifting of peak stage loading as the compressor speed changes is illustrated in Figure 3.

In Figure 3, three groups of stage characteristics, representing conditions at the front, middle and rear of a compressor, are shown for low, medium and high speeds. These characteristics were derived by predictive means and on each one a dot is used to indicate the operating condition for that particular stage while the compressor as a whole is operating at the stability limit. It can be seen that in the low speed range, the front stage operates to the left of the peak of the characteristic while the rear stage operates furthest from the point of instability. In the mid-speed range, all three stages in this case are similarly loaded and the compressor may be considered to be well matched from front to back. At top speed it is the rear stage which operates closest to the the point of instability while the front stage is well away from stall.

The position of the stage which is most heavily loaded, and which is most likely to cause stall, thus shifts from the front to the rear of the compressor as the speed goes up. This phenomena is a well known in high-speed compressors, Cumpsty (1990), and in subsequent sections it will be shown how this shift in stage loading affects the stall and surge behaviour of the compressor.

OVERALL PERFORMANCE AFTER INSTABILITY

Before considering the process by which the flow through the compressor becomes unstable it will be useful to examine the overall behaviour of the high-speed compressor after instability has set in. To this end, Figure 4 shows a number of fixed speed pressure rise characteristics for an imaginary compressor at various operating conditions. Together with the efficiency curves, these lines constitute the overall performance map of the compressor. The map is bounded at the upper edge by the so called "surge line", this being a general term used to describe the locus of all points at which the flow becomes unstable either through surge or stall. At low speeds the compressor experiences a type of post-stability disturbance known as "front end stall". This is generally a disturbance of light intensity which does not hinder the start up process of the compressor - unless it degenerates into single cell rotating stall. At higher speeds, say between 65% and 85% of full speed, the compressor will go into rotating stall at the surge line. In this case the disturbance is severe and the abrupt loss of efficiency means that the engine must be shut down to prevent the turbine overheating. At still higher speeds, the combined compression and combustion systems hold sufficient stored energy for the engine to surge. This process results in repetitive cycles involving temporary flow reversal, unless, as can happen at maximum speed, combustor flame-out occurs and then only the first part of the first flow reversal cycle is observed.

The types of compressor behaviour described here can be illustrated in more detail by looking at the actual results from the Viper compressor, Figure 5. (It should be noted that the instrumentation used for these measurements is standard test cell equipment and is of limited dynamic range. Some transient features of the stalling behaviour are therefore lost, but the overall trends are well represented.) Figure 5 shows the experimentally determined working line of the compressor and the surge line. Also shown are the

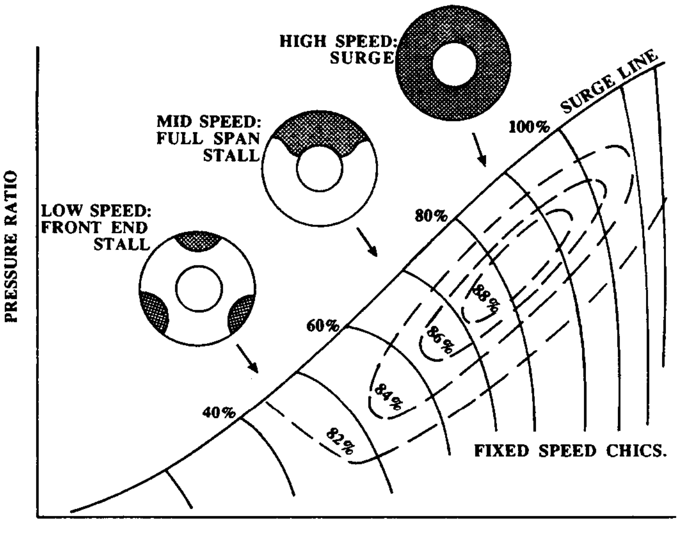


Figure 4. Idealised compressor performance map showing the types of instability encountered at various speeds.

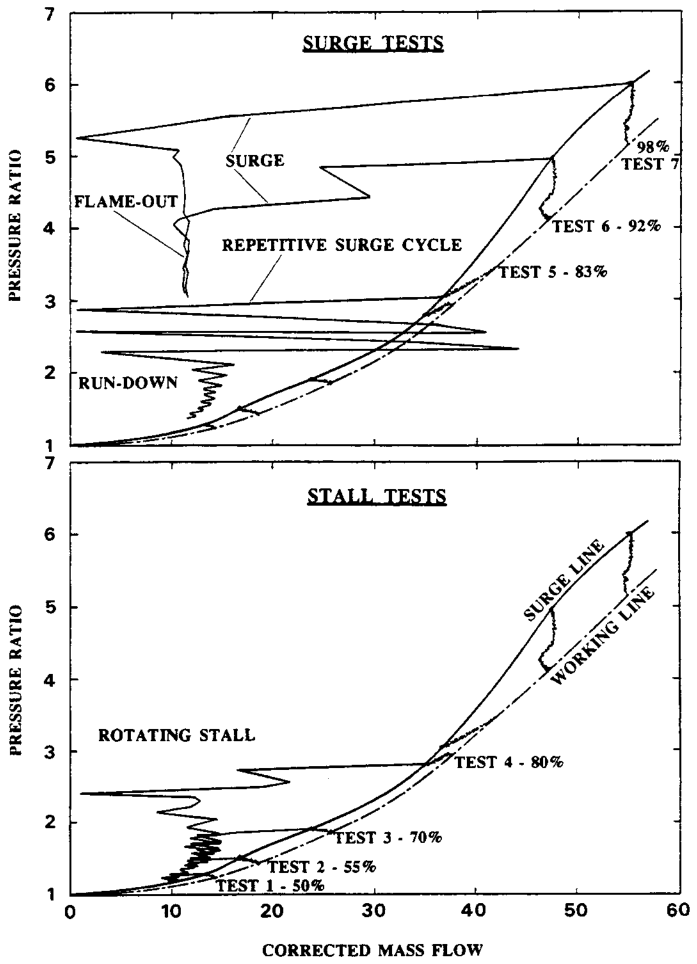


Figure 5. Viper performance map showing the post-stability trajectory of the compressor working point at various speeds. (Figure is in two parts to improve clarity)

trajectories of the compressor operating points immediately before and after instability. As previously explained, the operating point of the compressor does not move from the working line to the surge line along any particular path. The path depends on the technique used to induce instability.

Considering the various test runs in turn, we start with runs 1 and 2 where the working line and surge line are close together. At these low speeds the compressor operates in stall all the time; this type of stall being "front end stall" which is relatively light and is not severe enough to prevent the compressor from being accelerated to a higher speed. If over-fueling occurs while operating in this condition, however, a change to fully developed rotating stall, affecting the whole compressor, will occur with the result that the engine will have to be shut down. This change from "front end stall" to rotating stall will be illustrated in detail in a later section.

In the mid-speed range, as for tests 3 and 4, over-fueling the engine pushes the operating point over to the surge line where rotating stall is initiated. Once stalling occurs, the compressor mass flow drops suddenly with an accompanying drop in pressure rise and the engine is again forced to shut down. The type of stall which occurs at this speed is known as single cell full span stall, or "locked-in stall", and is responsible for an abrupt loss of efficiency. The stall cell extends as a single axial disturbance throughout the length of the compressor and rotates around the annulus at about 50% of rotor speed.

If the engine speed is increased slightly, but staying within the mid-speed range, for example test 5 in Figure 5, a disturbance of very different proportions occurs when the compressor is pushed beyond the point of instability. Here surge is encountered for the first time and the flow through the compressor undergoes a number of violent oscillations before the fuel is cut and the compressor slows down. In the low-speed laboratory compressor, this type of oscillation can be sustained indefinitely, but in an engine environment the risk of structural damage means that rapid shut-down is essential.

It is interesting to note that the change in speed between runs 4 and 5 in Figure 5 is relatively small, yet the change in compressor behaviour is remarkable. The same thing was observed by Greitzer (1978) in the testing of his low-speed compressor when it was noted that the speed range separating rotating stall and surge is very narrow. In the context of the Viper tests, Figure 6 illustrates both the narrowness of the speed divide and the dramatic difference in compressor behaviour. This figure shows the recorded pressure levels from circumferential arrays of probes at various stations along the length of the compressor. The left hand side of the figure, at 80% speed, shows the compressor going into rotating stall, while the right hand side, at 83% speed, shows the distinctive one-dimensional oscillations associated with surge. The speed range separating these two pictures is just 3%.

Returning to Figure 5, it can be seen that a further change in compressor behaviour occurs as we approach full speed, runs 6 and 7. Here, a repetitive surge cycle does not occur, as for test 5, because the initial mass flow fluctuation is so vigorous that the engine experiences a flame-out, followed by a rapid drop in rotational speed. Surge at this speed therefore consists of just the first part of a surge cycle, i.e. a sudden reversal of flow followed by rotating stall as the engine coasts to a standstill.

Having considered the question of stage loading prior to instability, and the overall behaviour of the compressor after instability, we now consider the instability process itself, i.e. the details of the inception process through which the flow becomes unstable.

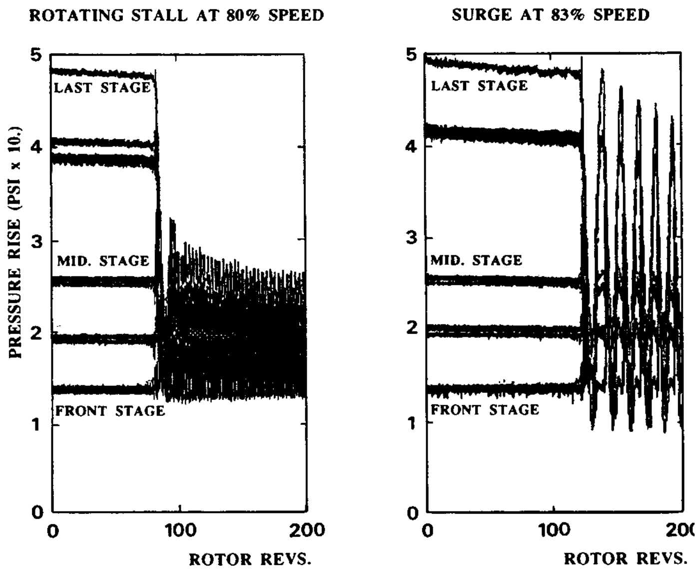


Figure 6. Examples of rotating stall and surge in the Viper compressor separated by a small change in speed. The plots show pressure measurements from 5 circumferential positions at various stations along the length of the compressor.

STALL INCEPTION IN THE LABORATORY COMPRESSOR.

A number of detailed studies of stall inception in low-speed compressors have been conducted in recent years. It has been found that there are two distinctly different mechanisms by which axisymmetric flow becomes unstable. The first mechanism to be identified was that associated with modal perturbations. The theoretical work behind the identification of these modes was done by Moore and Greitzer(1986) and the experimental validation by McDougall et al (1990), Garnier et al (1990), and Day (1991a). In this type of stall inception small velocity perturbations of circumferential length scale appear in the flow just prior to stall. The perturbations rotate around the annulus and grow over a number of revolutions until the perturbation itself becomes so intense as to be recognisable as a stall cell. The speed with which the modal perturbation rotates remains relatively steady from first detection right through to the fully developed state.

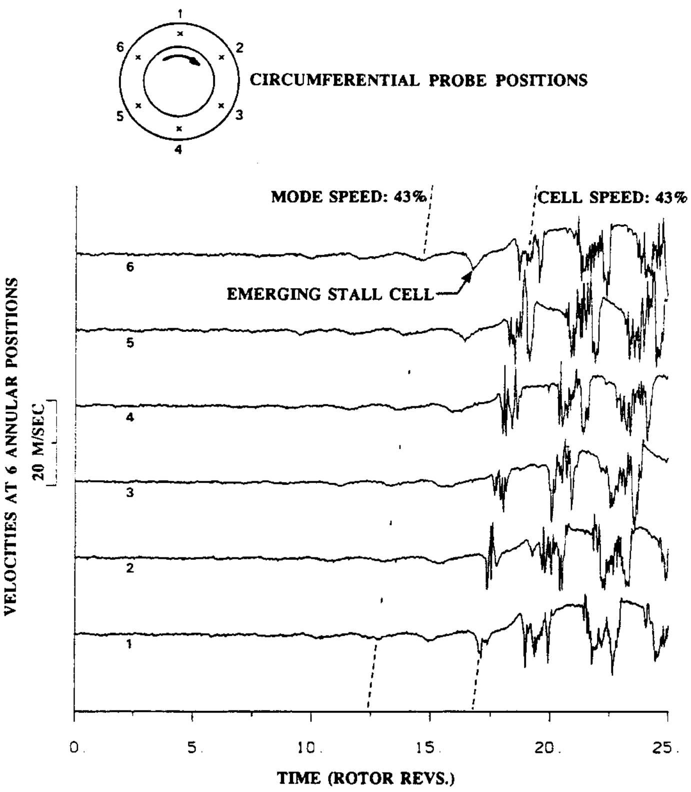


Figure 7. Example of rotating stall originating from a modal perturbation in the low speed compressor.

An example of stall occurring via a modal perturbation is illustrated in Figure 7 where six hot-wires are used around the circumference of the compressor to track the development of the mode. In this figure, the perturbation is first visible at about rotor revolution 10, and over the next twelve revolutions grows to maturity as a single, full-span, stall cell. The rotational speeds marked in the figure shows that there is no change in frequency during the development process. This type of stall cell thus originates from an infinitesimally small disturbance and can therefore be adequately modelled by linear theory.

The second mechanism by which the flow may become unstable is via the growth of a small localised disturbance which appears without precursive build-up and which grows quickly to engulf a large part of the annulus. This type of stall inception was first identified by Day (1991a) and can be visualised as a patch of locally separated flow initially affecting just the tips of one or two blades. An example is given in Figure 8 where the first sign of instability occurs on wire 2 at rotor revolution 15. At this point the instability is very small, but it grows to become a fully developed stall cell in just six rotor revolutions. It is important to note that the speed of rotation of the disturbance changes markedly as it grows in size, i.e. from 72% at outset to just 45% when mature. The change in speed provides a useful means of identifying this type of stall inception.

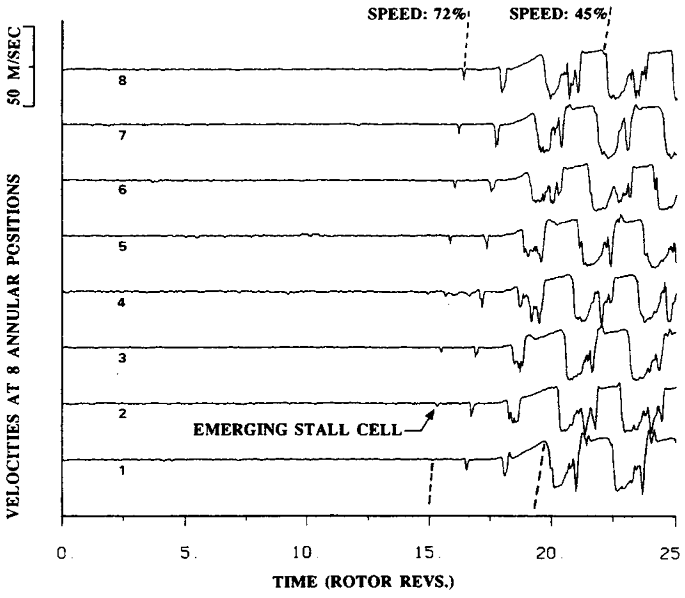


Figure 8. Example of rotating stall originating from a localised disturbance in the low speed compressor.

Of the two stall inception processes outlined above, the second is the most representative of what is observed in the Viper compressor. The measurements in Figure 8 will therefore be used as the starting point for assessing the similarities between the laboratory compressor and the Viper engine.

STALL INCEPTION IN THE ENGINE AT MID-SPEED.

It will be recalled that in the mid-speed range, 75 to 85% of full speed, all of the stages in the engine compressor are approximately matched near the surge line and all approach instability at the same time. This is much the same as what happens in the laboratory compressor where the flow is incompressible and all the stages are equally loaded. It is therefore in this speed range that we should expect to see the greatest similarity in the behaviour between the low-speed and high-speed compressors. Figure 9 shows measurements from the engine using five pressure transducers equally spaced about the annulus near the front of the compressor. Because wall mounted pressure transducers were used instead of hot-wires, the initial perturbations peak upwards instead of downwards, nevertheless, there is no mistaking the similarity of the stall development patterns in Figures 8 and 9.

In Figure 9 a small localised disturbance appears at revolution 24 and grows to become a single fully developed stall cell. As in Figure 8, the speed of rotation is high at the beginning and lower when the cell is fully developed. It may therefore be concluded that in this speed range the engine and the laboratory compressor behave in very similar ways, each stalling as the result of a small localized stall cell which rotates quickly at first, but slows down as it grows in size. Figure 9 is thus the first clear picture we have of stall inception in an

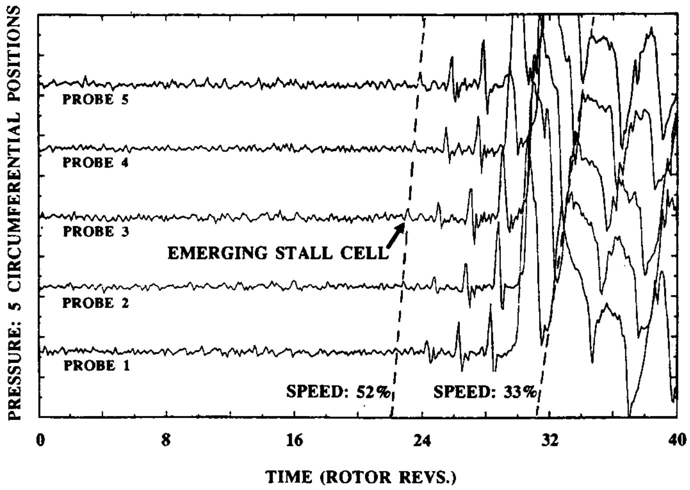


Figure 9. Stall inception in the engine compressor at mid-speed measured at 5 circumferential positions near the front of the compressor. The stalling pattern originates from a localised disturbance.

engine compressor and suggests that, in this instance at least, stall inception is via a localized disturbance rather than a modal perturbation.

The lead-in traces in Figure 9, from rotor revolutions 0 to 24, are more turbulent than those in Figure 8 (for the laboratory compressor) and therefore it is reasonable to ask if some form of modal perturbation is not perhaps present in the background noise. This is a possibility, however, Fourier analysis in one and two dimensions has failed to yield any sign of a coherent disturbance in this region. Other tests for modal coherence, based on radar identification techniques, have also found no evidence of precursive modal disturbances.

SURGE INCEPTION IN THE LABORATORY COMPRESSOR.

Even though it is a low-speed compressor, the C106 can be made to surge, rather than stall, if a large plenum is interposed between the compressor exit and the throttle. The additional energy stored in the plenum means that a surge cycle will be set up as soon as the compressor goes into rotating stall. Experiments demonstrating the part played by rotating stall in initiating surge have been reported by Day (1991b), and a set of representative measurements for the laboratory compressor is given in Figure 10.

Figure 10 shows the first, and part of the second, of a series of repeating surge cycles. The upper trace is from a hot-wire positioned upstream of the first rotor and records the axial velocity in the compressor, while the lower trace shows the corresponding pressure variation in the plenum chamber. For the first twenty-five revolutions of the traces shown, the compressor operates in a stable manner at peak pressure rise. At revolution 25 the flow becomes unstable and a burst of rotating stall is observed followed by a dramatic fall in plenum pressure. The rotating stall itself is short lived and is soon replaced by axisymmetric reversed flow, Day (1991b). About forty revolutions after the start of the cycle the compressor unstalls and the plenum pressure begins to rise. Provided the throttle has not been moved since the cycle began, a second surge cycle will be initiated when the compressor stalls again at peak pressure rise.

The patch of rotating stall which initiates the surge cycle is more clearly illustrated in Figure 11. Here six hot-wires around the annulus are used to monitor the circumferential movement and growth of the stall cell. It can be seen that a small localised stall cell appears, without modal build-up, opposite hot-wire 3 at revolution 4, and then rotates and grows to the point where the whole of the annulus is enveloped in reversed flow. This stall inception process is exactly like

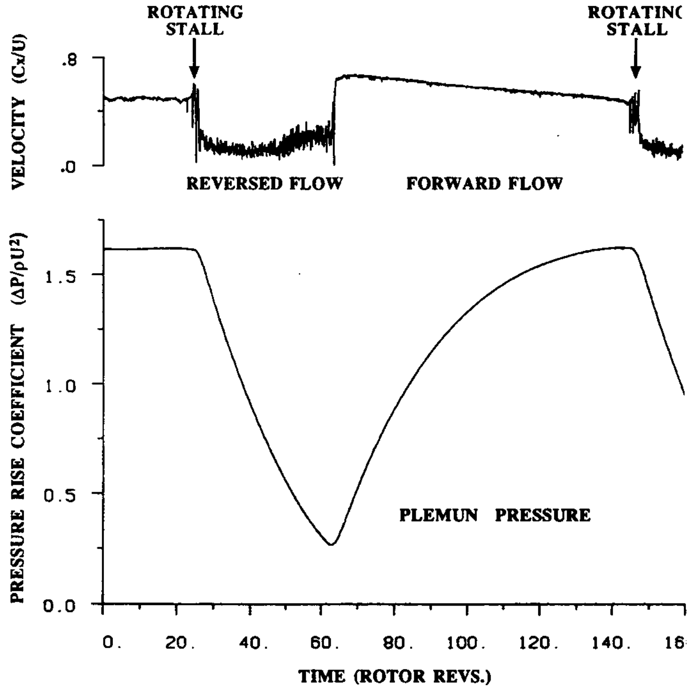


Figure 10. Velocity at the front of the compressor, and pressure in the plenum, as measured during surge in the low speed compressor. The surge cycle begins with rotating stall.

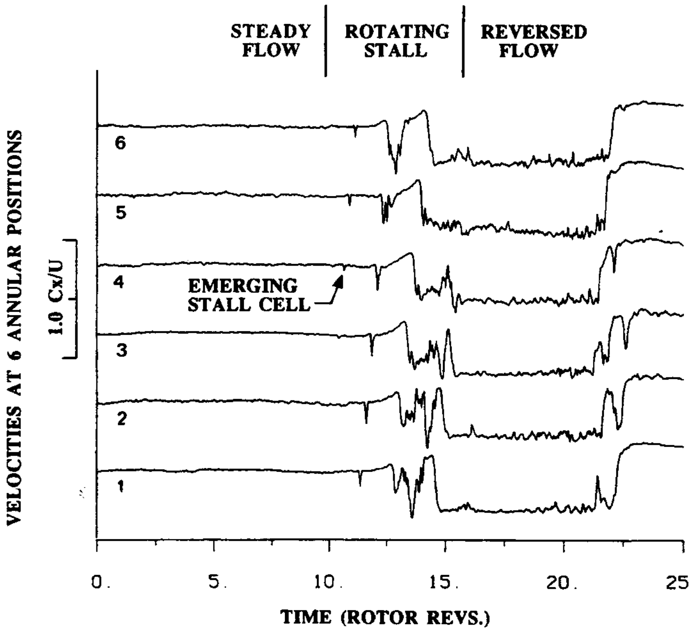


Figure 11. Details of the rotating stall at the start of the surge cycle in the low speed compressor, using six hot-wires ahead of the first rotor. (These measurements are for a similar, but not the same, event shown in Figure 10. A smaller plenum was used here.)

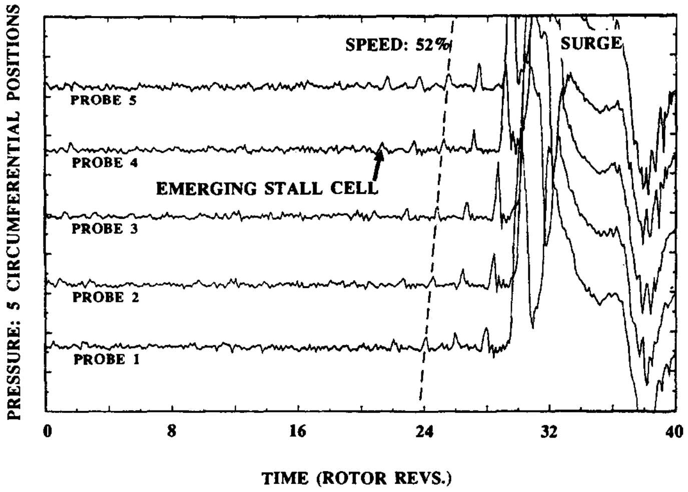


Figure 12. Engine pressure measurements at mid-speed showing rotating stall at the start of the surge cycle.

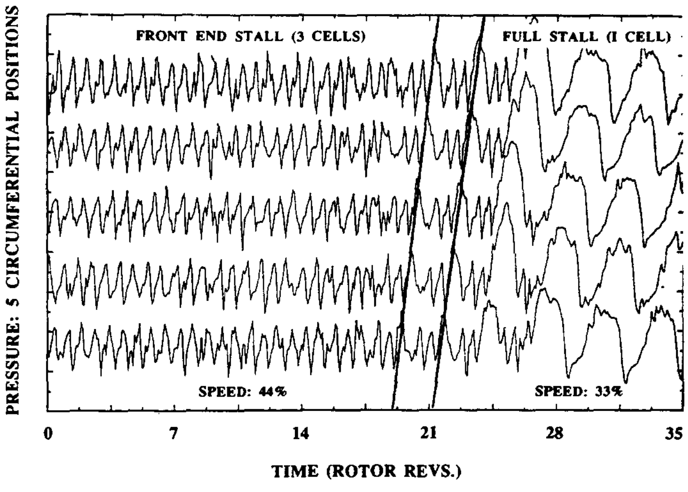


Figure 13. Engine measurements at low speed showing a three-cell "front end stall" pattern changing to a single cell pattern, This forces the engine to be shut-down.

that in Figure 8, even to the point where the initial speed of the stall cell is also 72%. Surge in the laboratory compressor thus begins with the same inception process associated with rotating stall.

SURGE IN THE ENGINE COMPRESSOR AT MEDIUM-SPEED.

In the Viper engine, a repetitive surge cycle is first observed when the speed reaches 83% of full speed, as can be seen in Figure 5, test 5. At this speed, the stages are well matched and all approach the instability line at the same time, as is the case in the low-speed laboratory compressor. We can therefore expect that the engine will again behave in the same way as the laboratory compressor. Detailed measurements of the start of a surge cycle in the Viper compressor at 83% speed are shown in Figure 12. As before, five equally spaced pressure transducers were used at the front of the compressor. It can be seen that the start of the instability process is fundamentally similar to that for rotating stall, as shown in Figure 9. This means that surge at mid-speed in the engine compressor begins with the formation of a localised stall cell and develops in the same way as in the laboratory compressor.

The preceding figures confirm that at certain speeds, where all the engine stages are well matched, useful information about stall and surge in the engine can be obtained directly from incompressible laboratory tests. It will be shown below that although conditions at low speed and high speed in the engine are somewhat different from those in the laboratory compressor, the fundamentals of stall inception learned from the laboratory tests provide useful insight into the engine results.

STALL IN THE ENGINE COMPRESSOR AT LOW SPEED

Having discussed the behaviour of the engine compressor at mid-speed where all the stages are well matched close to stall, it remains to show what happens in the engine at the lower and upper end of the speed range. We first consider conditions at low speed, i.e. as for test runs 1 and 2 in Figure 5. It will be recalled that the front stages are the first to run into trouble at this speed, see Figure 3, and that some rotating stall is often a permanent feature of operation in this range. In this particular case in the Viper compressor, three equally spaced stall cells were observed to be present right from start-up. The influence of these cells appeared to be restricted to the first three stages of the compressor. (The process by which these cells first form, i.e. their inception phase, was never observed because the cells were already present by the time the engine reached light-up speed.)

The type of stall cells seen here are not unlike the part-span cells sometimes found in lightly loaded laboratory compressors, Day (1976). How far the cells extend down the span of the blades is not known in this case, but the similarity with part-span cells in other compressors is supported by the fact that these cells do not stop the engine from operating normally, albeit slightly inefficiently. Part-span cells do not usually cause a large loss of pressure rise, and therefore their presence is not overwhelmingly detrimental to performance.

While operating with light stall at the front of the compressor, it may happen that the operating point is pushed still further into stall, for instance by sudden over fueling, in which case a larger full-span cell will develop out of the existing part-span pattern. An example of this happening is given in Figure 13 where five pressure transducers were used around the circumference at the front of the compressor. For the first 18 revolutions the compressor operates with three small stall cells restricted to the front end of the compressor. Events in the second half of the figure show how one of the three cells grows in amplitude until the other two are swallowed up leaving a single full-span cell rotating at a much slower speed. Parallel lines have been drawn in Figure 13 showing how the two smaller of the three original cells are squeezed out of existence as the larger cell grows to maturity. Other measurements from further back in the compressor confirm that the initial cells were limited to the first three stages and that the larger single cell, once formed, extended right through the length of the compressor.

SURGE AT HIGH SPEED IN THE ENGINE COMPRESSOR

At high speeds the engine surges in a very dramatic fashion, with a single bang and flames shooting out of the intake and exhaust. This happens for test runs 6 and 7 in Figure 5. A repeating surge cycle does not occur at this speed, because the initial flow reversal is so vigorous that combustion is extinguished, i.e. "flame-out" occurs. Without combustion, the engine decelerates very quickly passing through a period of rotating stall before coming to a standstill. In some engines, but fortunately not this one, structural bending as a result of the pressure loads can cause catastrophic failure.

In the past it has been suggested that this type of high speed surge consists of a blast wave which starts at the back of the compressor and sweeps through to the inlet at near sonic speed, the pressure difference across the shock front increasing as the wave approaches the front of the compressor, see Mazzawy (1980) and Cargill and Freeman (1990). In the current tests some evidence of a blast wave has been detected, but there are two other features of these tests which need to be considered first.

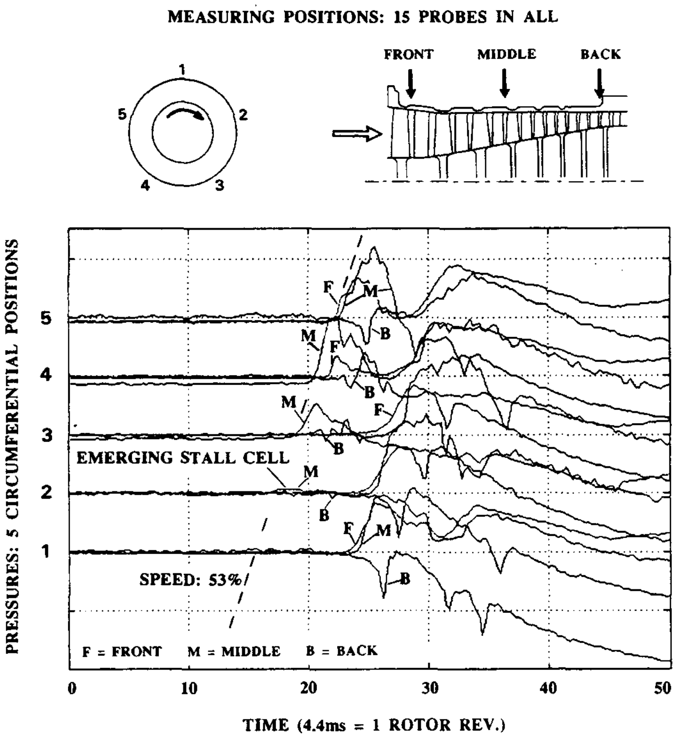


Figure 14. Engine measurements of surge inception at high speed. This figure shows front, middle and rear dynamic pressures at 5 circumferential positions. A rotating disturbance precedes the surge event.

To begin with, the physical location of the initiating disturbance in the Viper engine was not where it was expected to be, i.e at the rear of the compressor. The first sign of flow breakdown appears on the traces from the middle stages of the compressor. This can be seen in Figure 14 where three pressure traces are plotted at each of five circumferential position, one trace each for the front, middle and rear sections of the compressor. It might have been thought from Figure 3 that the rear stages would be the first to go unstable, but this does not appear to happen here. The first sign of instability occurs at time 18 at position 2 at the middle of the compressor. It is possible that the instability actually starts slightly further back than the plane of the measuring probes, however, it is nonetheless interesting that in this case flow breakdown does not start at the rear of the compressor.

The second interesting thing to come out of these full speed measurements concerns the rotational nature of the initial disturbance. Before these measurements were taken, ideas about the origins of the surge event were speculative rather than informed. The idea of a "blast wave " rushing forward from the rear of the compressor assumed some form of initiating disturbance, but just what form this disturbance would take was not known. From results like those in Figure 14 it is now clear that the initiating disturbance is once again rotating stall. The traces in Figure 14 show that the disturbance retains a coherent identity for about one and a half revolutions before being swamped by reversed flow. The speed of this disturbance is roughly 53% of rotor speed, similar to the stall cell speeds observed at lower power settings.

The preceding results have also been studied to ascertain if the emerging stall cell is a localised disturbance or has its origins in a modal perturbation. Modal analysis of the first 18 milliseconds of Figure 14 does not show any form of coherent precursive structure. In addition, the disturbance which starts the whole stalling sequence appears to be restricted to one sector of the annulus only. This suggests that, even at full speed, the onset of instability takes the form of a localised disturbance. This observation is not only significant in its own right, but it is also important from the point of view of future active control experiments. It determines the type of control strategy which will be required and highlights the need for very fast actuator systems.

In terms of a blast wave moving forward through the compressor, it can be seen from Figure 14 that the stall cell at the middle of the compressor has no effect on the front of the compressor until the the emerging stall cell reaches circumferential position 4. Starting from Time 0 and looking along the pressure traces at circumferential position 4, it can be seen that the flow at the front of the compressor becomes disturbed about 1 millisecond later than conditions at the middle, i.e. the ""M" trace begins to rise 1 millisecond before the "F" trace. The dimensions of the compressor are such that a disturbance taking this length of time to propagate from the middle of the compressor to the front will do so at a speed of 330 meters per second.This is near enough to sonic velocity to suggest that some sort of "blast wave" may be present. Evidence of this type from a single measurement is however inconclusive, especially as it does not explain why conditions at the front of the compressor are the first to be affected at circumferential position 5! Further measurements are therefore necessary to obtain a complete picture of what actually happens in the compressor at very high speeds.

The topic of surge at high speed is also discussed in a companion paper by Wilson and Freeman (1993).

CONCLUSIONS

Two compressors, one a low-speed laboratory machine operating incompressibly, the other a high-speed engine compressor, have been tested to examine the similarities in the stall inception processes. In the process of making the comparisons, interesting information about stall and surge in an aero-engine compressor is presented here for the first time. The conclusions to be drawn from this work are listed below.

1) This paper shows that useful information about stall and surge can be obtained from tests on low-speed compressors. Examples have been given where precise read-across from incompressible tests to the high-speed conditions is possible.

2) In the mid-speed range, all the stages in the engine compressor are evenly matched near the surge line and therefore it is reasonable to expect close similarity between the engine and laboratory tests. This has been shown to be correct for both stall and surge where the inception patterns in the two compressors are almost identical.

3) Previous work on laboratory compressors has identified two distinct mechanisms of stall inception: modal perturbations and localised stalling. Throughout the engine tests it was clear that a localised disturbance initiated stall and surge, with no modal disturbances being detected at any speed.

4) At low speeds, the engine compressor is seldom free of multi-cell stall in the front stages. This type of stall is similar in character to the part-span stall sometimes seen in laboratory compressors. If the engine is pushed further into stall when operating in this condition, the multi-cell pattern coalesces to form a single more severe cell pattern which extends throughout the whole compressor and forces the engine to be shut down.

5) Towards the upper end of the speed range the engine will surge rather than stall. Repetitive surge cycles were possible at medium to high speeds, but at top speed surging occurred with such violence that the flow reversal extinguished combustion. The results show that the repetitive type of surge is initiated by rotating stall in much the same way as in the laboratory compressor. Top speed surge has also been shown to be initiated by rotating stall, but the growth of the disturbance, and the rate at which it spreads throughout

the compressor, is much faster than at lower speeds. Some, though limited, evidence exists for a "blast wave" type of pressure front moving through the compressor soon after stall inception occurs.

6) It was observed that when surge was initiated at high speed, the initial pressure perturbation did not appear at the rear of the compressor, as might have been expected, but more towards the middle of the compressor. This finding may be a feature of the particular compressor tested, but it is mentioned for reference when measurements from other compressors become available.

7) Work in the laboratory has shown that the rotational speed at which the change from stall to surge occurs is well defined. Tests on the engine likewise show an almost abrupt change from stall to surge as the speed of rotation is increased.

8) Taken as a whole, this study confirms that laboratory testing of low-speed compressors is an effective and inexpensive way of gaining insight into many aspects of the behaviour of aero-engine compressors.

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