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STALL WARNING BY BLADE PRESSURE SIGNATURE ANALYSIS

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ABSTRACT

At low mass flow rates axial compressors suffer from flow instabilities leading to stall and surge. The inception process of these instabilities has been widely researched in the past – primarily with the aim of predicting or averting stall onset. In recent times, attention has shifted to conditions well before stall and has focussed on the level of irregularity in the blade passing signature in the rotor tip region. In general, this irregularity increases in intensity as the flow rate through the compressor is reduced. Attempts have been made to develop stall warning/avoidance procedures based on the level of the flow irregularity, but little effort has been made to characterise the irregularity, or to understand its underlying causes.

Work on this project has revealed for the first time that the increase in irregularity in the blade passing signature is highly dependent on both tip-clearance and eccentricity. In a compressor with small, uniform, tip-clearance, the increase in blade passing irregularity which accompanies a reduction in flow rate will be modest. If the tip-clearance is enlarged, however, there will be a sharp rise in irregularity at all circumferential locations. In a compressor with eccentric tip-clearance, the increase in irregularity will only occur in the part of the annulus where the tip-clearance is largest, regardless of the average clearance level.

In this paper, some attention is also given to the question of whether this irregularity observed in the pre-stall flow field is due to random turbulence, or to some form of coherent flow structure. Detailed flow measurements reveal that the latter is the case.

From these findings, it is clear that a stall warning system based on blade passing signature irregularity will not be viable in an aero-engine where tip-clearance size and eccentricity change during each flight cycle and over the life of the compressor.

INTRODUCTION

Rotating stall sets the low-flow limit of the operating range of an axial compressor. As a result of the damaging consequences of stall, a wide safety margin must be left between the operating flow rate of the compressor and the flow rate at which stall will occur. In an effort to reduce this unused safety margin, much work has been done in trying to detect when stall is likely to occur. This work has mostly concentrated on the small flow irregularities, modes and spikes, which are involved in the stall inception process; see for example the Moore and Greitzer [1, 2], Camp and Day [3] and the active control efforts of Strazisar et al. [4]. However, limited success has been achieved in obtaining reliable stall warning or effective stall suppression.

An alternative approach to the early warning problem is to step back from the actual stalling process and to look for signs of irregularity in the flow as the compressor is throttled towards stall. Some success has been claimed for this approach, but in general, efforts have concentrated on finding a stall proximity indicator rather than on understanding the nature of the disturbances in the flow. The current work is aimed at the latter problem.

In the past 15 years, work on pre-stall flow irregularities has been divided between tip-vortex studies [5], and stall warning work. An example of the latter is the work of Dhingra et al. [6], who measured the repeatability of the blade passing pressure signature using a correlation function. They found that this correlation function dropped as stall was approached. This trend was seen in both low- and high-speed compressors, suggesting some generality in the observed patterns. In a later paper, Liu et al. [7] defined this drop in repeatability in the blade passing signature as an 'event' and measured 'event rate' against stall margin. They found that the event rate ramped up rapidly as the flow rate through the compressor was reduced towards the stall point. Using a correlation function technique, Christensen et al. [8] describe a system that warns of imminent stall in time for the engine operating point to be adjusted away from danger.

Gannon et al. [9] carried out experiments in a transonic compressor at different speeds and measured the variation in blade passing pressure signal. They did not find a consistent ramp-up in irregularity as stall was approached. This result is contrary to the findings of Dhingra et al. [6] and Christensen et al. [8] and highlights the need to understand the basic physics behind the disturbances being measured.

Closely related to the topic of tip flow unsteadiness is the question of tip-clearance size. Studies have found that the stall margin deteriorates significantly if the tip-clearance gap is enlarged. In related work, Graf et al. [10] investigated the effect of non-axisymmetric tip-clearance on stall margin, and found that a compressor with non-axisymmetric tip-clearance would have a similar stall margin to one with a tip-clearance that is constant and equal to the largest tip-clearance in the non-axisymmetric case. They also showed that tip-clearance asymmetry causes a circumferential redistribution of the flow, such that the flow coefficient is reduced in the large tip-clearance part of the annulus and increased where the gap is smallest.

Bennington et al. [11] have studied the development of short-length scale disturbances (spikes) in a compressor with non-axisymmetric tip-clearance. They showed that, very close to the stall point, disturbances appear in the region with the largest tip-clearance, and then decay in the part of the annulus where the tip-clearance is smaller. Eventually, one of these disturbances completes an entire revolution of the annulus without fully decaying and then grows into a stall cell. However, Bennington et al. did not examine the subject of pre-stall blade passing irregularity.

The current work builds on the above findings and shows that tip-clearance asymmetry will lead to the formation of flow disturbances in the large tip-clearance sector of the annulus as stall is approached. These disturbances cause the signature of each blade passing to become less regular, but, because the compressor is less stable in the part of the annulus with the largest tip-clearance, the increase in irregularity is seen most clearly in this part of the annulus. In a concentric compressor with larger than normal tip-clearance, this rise in irregularity is seen at all circumferential positions.

The purpose of the current work is not to make better use of the observed flow irregularities to predict stall, but to understand when and why these irregularities occur. A better physical understanding of the whole topic of pre-stall flow disturbances will help explain the successes and contradictions of past work and will put the study of stall warning on a better footing. At the same time, the current work highlights the presence of structured disturbances in the pre-stall flow field and links these to the rise in blade passing irregularity as stall is approached.

EXPERIMENTS

The experiments described in this paper were carried out on a single-stage, low-speed, high hub-tip ratio compressor in the Whittle Laboratory. Details of the compressor are given in Table 1.

The compressor casing can be moved relative to the centreline of the rotor shaft to give a variation in tip-clearance around the annulus. In this paper, eccentricity is always quoted as a percentage of the average gap size:

$$\text{Eccentricity} = \frac{\varepsilon_{\max} - \varepsilon_{\min}}{\bar{\varepsilon}} \times 100\%$$

Where ε denotes tip-clearance. An eccentricity of 200% would lead to the blades rubbing the casing at one circumferential location. The average tip-clearance can be increased by cutting back the tips of the blades. In this paper, both tip-clearance size and eccentricity are varied systematically. While the tip-clearance variation can be measured to within 5 microns with a dial gauge, it is difficult to measure the absolute gap size. For this work, the gap size was

Tip Diameter	488 mm
Hub-to-tip ratio	0.75
Blade count (rotor, stator)	58, 56
Rotor stagger (hub, setting angle)	35°
Stator stagger (hub, setting angle)	22°
Rotor chord (true chord)	36 mm
Rotor tip-clearance (datum)	0.6 mm (1.7% chord)
Rotational Speed	2980 rpm
Design Flow Coefficient, $\phi = \frac{V_x}{U_{mid}}$	0.50
Design Stage Loading, $\psi = \frac{P_{exit} - P_{o,in}}{\frac{1}{2}\rho U_{mid}^2}$	0.57

Table 1: Datum parameters of the compressor

measured using feeler gauges, so is only accurate to 25 microns. As the blades are machined on the rotor, there is no significant variation in blade height.

In the datum configuration, the compressor showed spike-type stall inception. Both spike and modal stall have been observed in previous work on this compressor, depending on the stator stagger angle [12]. During the course of the present work, the tip-clearance gap size was also found to have an effect on the stall inception mechanism, with the compressor more likely to exhibit modal stall when operating with a large clearance.

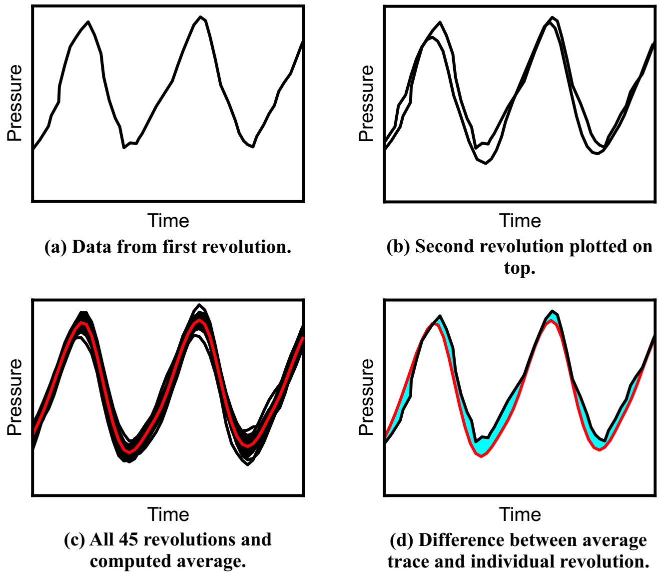
DATA ACQUISITION AND ANALYSIS

In the experiments, fast response pressure transducers were used in various configurations over the rotor tips. In all cases, data was sampled at 100 kHz, giving approximately 34 samples per blade pitch. Data was acquired for 45 revolutions at a constant flow coefficient, and was filtered at 50 kHz to avoid aliasing.

The standard probe configuration consisted of 14 transducers fitted at the same axial location as the rotor leading edge tips, evenly spaced around the annulus, i.e. one every 4 stator pitches. From a study in which the axial location of the transducers was varied, the leading edge location was found to be the most useful in terms of stall warning measurements, i.e. in terms of measuring the ramp-up in irregularity. The largest irregularity levels were, however, recorded at 20% chord.

The aim of the signal analysis process was to obtain a single number to represent the irregularity in the blade passing pressure signature. The signal analysis process is illustrated in Fig. 1, and consists of the following steps:

1. Data is acquired at a *fixed* flow coefficient for 45 revolutions. The signal from the first revolution is then plotted, as shown in part in Fig. 1(a). In reality, the whole revolution is analysed, i.e. 58 blade passings, of which just 2 are shown here. Data from each of the 14 pressure transducers is processed separately in the same manner.
2. The data from exactly one revolution later is then plotted on top of the first revolution, so that the blade passing signature of each individual blade can be compared with its own signal from one revolution later, as shown in Fig. 1(b).
3. This process is repeated for all 45 revolutions, and the average blade passing signal is computed, as denoted by the red line in Fig. 1(c).



(a) Data from first revolution. **(b) Second revolution plotted on top.**

(c) All 45 revolutions and computed average. **(d) Difference between average trace and individual revolution.**

Figure 1: Data analysis.

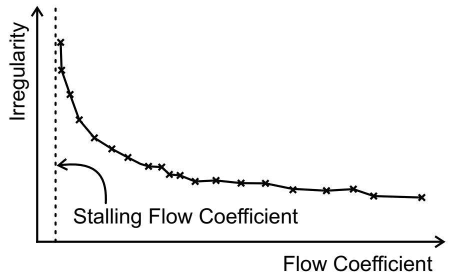


Figure 2: Hypothetical sketch of irregularity against flow coefficient (for one pressure transducer only).

4. The first revolution, as in Fig. 1(a), is then compared with the average trace, and the difference is found. This is represented by the blue shaded region in Fig. 1(d). The RMS of this difference across the entire revolution is then calculated. This RMS calculation is repeated for all 45 revolutions.
5. The mean of all the 45 RMS differences is then computed. This mean value is non-dimensionalised by $\frac{1}{2}\rho U_{mid}^2$, in order to make it directly comparable to the total-to-static pressure rise across the stage. This single number is taken to represent the irregularity of the blade passing signal at that flow coefficient. The higher the value of irregularity, the greater the variation in the blade passing signal from one revolution to the next.

The data from all the pressure transducers is analysed separately. In this way, separate values for irregularity are obtained for each circumferential location around the annulus, 14 in all.

The aim of the analysis is to build up an overall picture of how the repeatability of the blade passing signal changes as the compressor flow coefficient is reduced and stall is approached. The above process is therefore repeated for 20 to 30 flow coefficients from open throttle to stall. From this data, the variation of irregularity with flow coefficient is obtained. A sketch illustrating the possible dependence of irregularity on flow coefficient for one of the 14 pressure transducers is shown in Fig. 2. The crosses on the line represent the discrete flow coefficients at which measurements were obtained.

The effect of analysing a subset of the blades, or fewer revolutions' worth of data was studied by analysing a range of dataset sizes down to just 2 blades over 2 revolutions. It was found that the basic trends do not change, but the graphs are much clearer if the whole blade row is analysed over as many revolutions as possible. In this work, 45 revolutions' worth of data was chosen as a reasonable compromise between file size and repeatability.

STALL WARNING MEASUREMENTS

In this section, the results of the study into the change in irregularity of the blade passing signature as stall is approached will be discussed. The results obtained from the compressor in its datum configuration will be presented first, followed by the effects of tip-clearance size and eccentricity.

Datum Configuration

With the compressor in its datum, axisymmetric, configuration (i.e. a uniform tip-clearance of 1.7% chord), a small ramp-up in irregularity was observed as the compressor was throttled towards stall. This can be seen in Fig. 3, which is a plot of irregularity against flow coefficient (the location of the probes around the annulus is shown schematically in the top-right corner). This ramp-up is similar to that observed by Dhingra et al. [6] and Christensen et al. [8], though here the rise in irregularity is small, especially when compared to the stage pressure rise. The final level of irregularity (LHS) is between 5 and 10% of the near-stall total-to-static pressure rise of the compressor.

In Fig. 3, it can be seen that there is some variation in ramp-up between the different pressure transducers. Figure 4 is a plot of near-stall irregularity against circumferential position for the datum configuration, i.e. the final value of irregularity from each of the 14 pressure transducers at the last measuring condition before stall. It can be seen that there is no particular trend in irregularity level round the annulus.

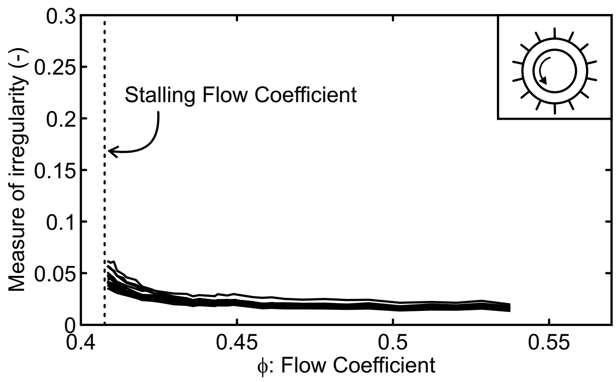


Figure 3: Irregularity against flow coefficient for the datum concentric compressor.

Effect of Tip-Clearance Size

The results for near-stall irregularity in a concentric compressor with increasing tip-clearance are shown in Fig. 5. It can be seen that in all cases the level of irregularity near stall is

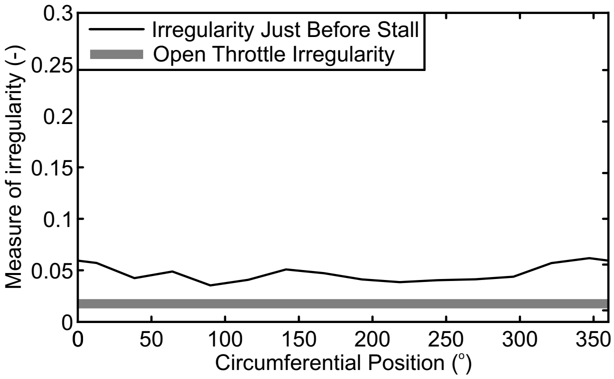


Figure 4: Near-stall irregularity against circumferential position for the datum concentric compressor (tip-clearance 1.7% span).

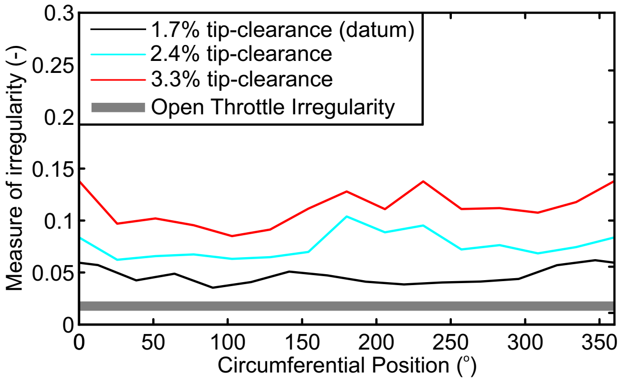


Figure 5: Near-stall irregularity against circumferential position with different average tip-clearances (concentric).

approximately constant round the annulus, and that the irregularity level increases as the tip-clearance of the compressor increases. Again, the lines in Fig. 5 are not perfectly constant, but there is no trend in irregularity with circumferential position.

It should be noted that there is a significant loss of stall margin associated with increasing the tip-clearance, so the near-stall flow coefficient changes as the tip-clearance is enlarged. This will be discussed in more detail below.

Effect of Eccentricity

To examine the effect of tip-clearance asymmetry, with the tip-clearance set to 1.7% chord (datum level), the casing was moved to give a variation in tip-clearance around the annulus of 75% of the mean value (i.e. the clearance varies from approximately 0.4 to 0.85 mm). The results from this test are shown in Fig. 6. As with enlarging the tip-clearance, introducing eccentricity leads to a loss of stall margin, so the near-stall flow coefficient increases. It can be seen that the variation in irregularity around the annulus is now more marked, i.e. the left hand end ends of the lines are spread out.

The pressure transducers in the small tip-clearance region of the compressor show no real rise in irregularity (3-5% of near-stall stage loading), while the probes in and just after the maximum tip-clearance show a significant ramp-up as stall is approached. The maximum level of irregularity (green trace) is seen to occur just after the maximum tip-clearance. The level is about 18% of near-stall stage loading. To see the connection between local tip-clearance and irregularity more clearly, it is useful to plot the level of irregularity near stall against circumferential position. This is shown in Fig. 7 for the datum symmetric and asymmetric configurations. In this graph, it is clear that the ramp-up in irregularity is confined to a sector starting just before the maximum tip-clearance and ending just before the minimum tip-clearance.

The compressor was also tested with lower levels of eccentricity. It was found that a peak in irregularity just after the maximum tip-clearance was the norm for all cases tested, even with as little as 10% eccentricity.

The results discussed here show that there will be a significant ramp-up in irregularity as stall is approached in an eccentric compressor. However, this ramp-up is confined to the large tip-clearance region of the annulus. No ramp-up at all is detectable in the small tip-clearance part of the annulus. It should be noted that there is also a phase difference between the maximum tip-clearance and the maximum level of irregularity of approximately 40°. This will be discussed in more detail below.

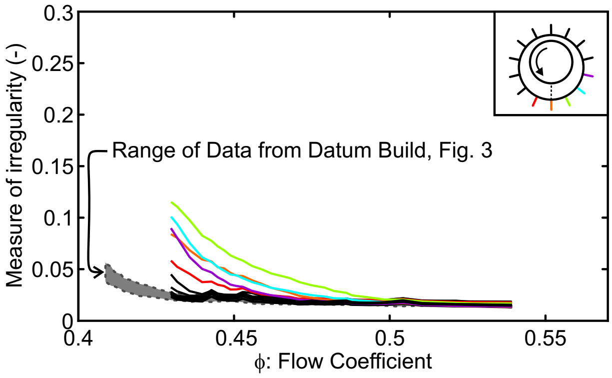


Figure 6: Irregularity against flow coefficient for the datum eccentric compressor.

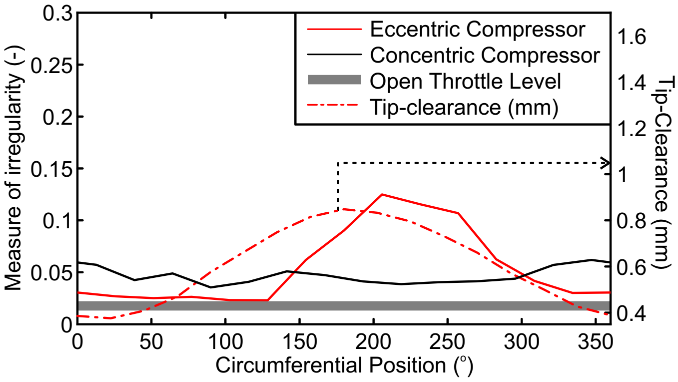
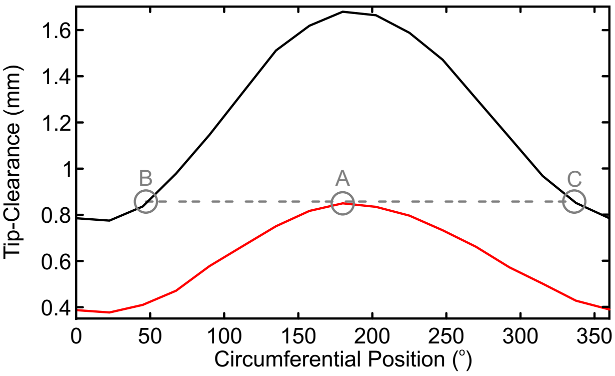


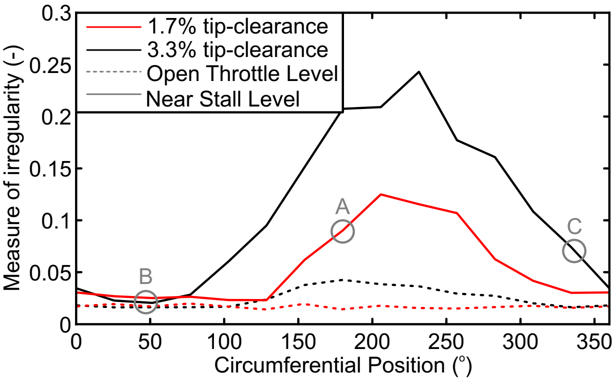
Figure 7: Near-stall irregularity against circumferential position for datum concentric and eccentric compressors.

Effect of Tip-Clearance Size with Eccentricity

The compressor was tested with eccentricity at three levels of average tip-clearance gap – 1.7, 2.4 and 3.3% chord (i.e. 0.6 mm, 0.85 mm and 1.2 mm). In each case the compressor was tested with the same absolute level of tip-clearance variation (0.45 mm) and also with a change of 75% of the average clearance. For the



(a) Tip-clearance distributions.



(b) Irregularity levels at open throttle and near stall operating points.

Figure 8: Comparison of near-stall irregularity levels with two levels of average tip-clearance (75% eccentricity).

sake of clarity and brevity, only the key results are shown here.

Figure 8 shows results from the compressor with eccentricity equal to 75% of the average clearance with two different levels of average clearance: 1.7% and 3.3% chord. The tip-clearance distributions are shown in Fig. 8(a), while the irregularity levels are compared in Fig. 8(b). It can be seen that the maximum level of irregularity is approximately doubled between the case with average tip-clearance of 1.7% of chord, and the case where the average tip-clearance is 3.3% chord. The region over which there is little or no pre-stall ramp-up, near point B in Fig. 8(b) has become smaller, but still covers a sector of approximately 75° from the minimum clearance.

The minimum tip-clearance in the 3.3% average clearance case is approximately equal to the maximum clearance in the 1.7% clearance case, as shown by the dotted line in Fig. 8(a). The irregularity levels at these locations can be compared in Fig. 8(b). It can be seen that, out of the three labelled positions, the irregularity is highest at circumferential position A. The level measured at position C is slightly lower, and there is no detectable irregularity at position B – remember that the local clearance is the same at all three locations. This shows that the level of irregularity is not only a function of the local clearance, but of casing eccentricity as well.

It should be noted that the stall inception mechanism changed from spike to modal when the tip-clearance size was increased. This does not seem to have affected the observed trends in near-stall irregularity.

From these results, it can be seen that the level of irregularity is not dependent on local tip-clearance alone, but on both tip-clearance and the level of eccentricity. There appears to be a dynamic effect associated with the passage of the blades around the eccentric annulus which affects the irregularity level.

ECCENTRICITY AND COMPRESSOR PERFORMANCE

The effect of tip-clearance eccentricity on compressor performance has previously been considered both experimentally and analytically. Hynes and Greitzer [13] developed a model for flow redistribution in a compressor operating under inlet distortion and Graf et al. [10] then adapted this model for a compressor with eccentric tip-clearance.

The Graf model is based on a parallel compressor argument and states that an eccentric compressor will have a higher flow coefficient in the small tip-clearance region and a lower than average flow coefficient in the large tip-clearance region.

The compressor was run at four different levels of (concentric) tip-clearance; the characteristics are shown in Fig. 9. From these results, the model of Graf et al. [10] can be applied to the test compressor.

The idealised tip-clearance variation used in the Graf model is shown in Fig. 10(a), and the computed local flow coefficient at near-stall conditions is shown by the black line in Fig. 10(b). It can be seen that there is a flow coefficient variation around the annulus of approximately 4%. The minimum flow coefficient in Fig. 10(b) is found to occur at about 10° after the maximum tip-clearance. Separate tests showed that the velocity distribution computed from the Graf model is in good agreement with measurements from around the circumference of the current test compressor. Looking at the black line only in Fig. 10(b), this variation in flow coefficient alone would mean that the compressor would first become unstable just after the maximum tip-clearance – because the flow coefficient is lowest in this region.

From Fig. 9, it can be seen that a compressor with large tip-clearance will stall at a higher flow coefficient than one with a smaller tip clearance. This means that, in an eccentric compressor, the flow in the small tip-clearance part of the annulus is expected to be more stable than that in the large clearance. From the relationship between tip-clearance and stalling flow coefficient demonstrated in Fig. 9, the equivalent axisymmetric stalling flow coefficient at all circumferential locations in the non-axisymmetric compressor can be estimated. The result is shown by the red line in Fig. 10(b). The variation in local stability round the annulus seen here also suggests that the compressor will be most unstable in the maximum tip-clearance region.

In the centre of Fig. 10(b), it can be seen that the region of lowest local flow coefficient overlaps with the region of high stalling flow coefficient, and so the compressor will be locally unstable in the area marked in grey.

In this simulation, the unstable sector would extend from 30° before the maximum tip-clearance to 40° after it. It is gratifying to note that the end of the unstable region in Fig. 10(b) is at approximately the same circumferential location as the maximum irregularity in blade passing signature recorded in the experiments.

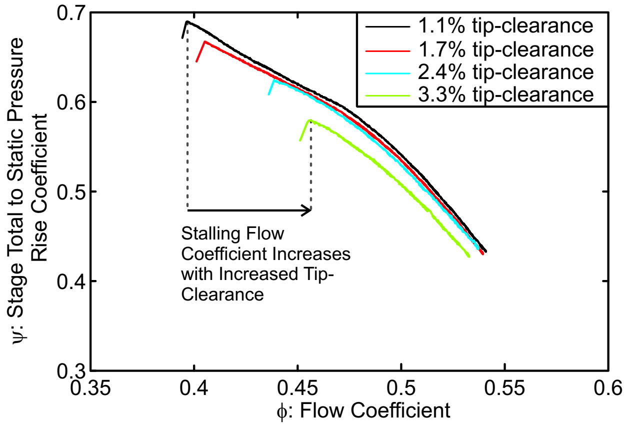
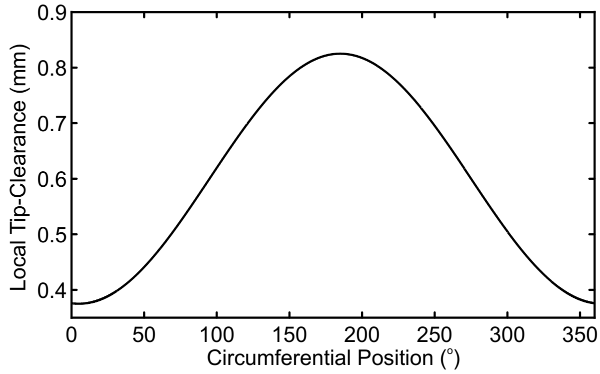
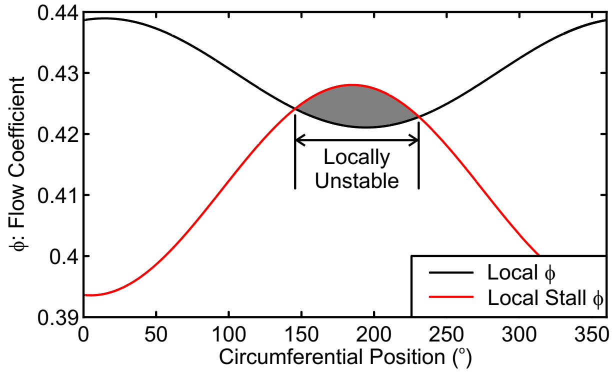


Figure 9: Total-to-static pressure rise characteristics for a compressor with four levels of concentric tip-clearance.



(a) Local Tip-Clearance.



(b) Comparison of Local Flow Coefficient and Stalling Flow Coefficient.

Figure 10: Effect of tip-clearance asymmetry on local stability.

The part of the annulus where the stall margin is smallest, due to the large tip-clearance, coincides with the region of low flow coefficient, causing this part of the annulus to be far less stable than the small tip-clearance sector. Taken together, the model of Graf et al. [10] for the flow distribution in a non-axisymmetric compressor and the variation in local stability around the annulus, go some way to explaining why the measured blade passing irregularity is highest in the large tip-clearance region of a non-axisymmetric compressor. The circumferential offset between tip-clearance and local flow coefficient also explains why the maximum level of irregularity occurs after the maximum tip-clearance.

THE NATURE OF THE DISTURBANCES

It has been shown that there will be an increase in irregularity in the blade passing signature if the tip-clearance is large and/or eccentric. The underlying cause of this irregularity has, however, not yet been established.

In order to examine the rise in irregularity more closely, three sets of pressure transducers were fitted to the casing one stator pitch apart as shown in Fig. 11. The probes were positioned from 40% axial chord upstream of the rotor leading edges to 60% downstream of the leading edge. These pressure transducers, as a group of three sets, were then moved round the annulus in 45° steps to observe the changes in disturbance level at different tip-clearance gaps. In these experiments, the compressor had a large average tip-clearance (3.3% chord) with 75% eccentricity.

Figure 12 shows the Fourier transforms of the signal from one of the pressure transducers at 20% chord at each of the eight locations around the annulus near stall (ϕ= 0.465, i.e. 0.6% above stall). It can be seen that the pressure transducers in the small tip-clearance sector (Fig. 12(a) and (b)) show very little frequency content apart from the blade passing frequency (around 2900 Hz). In and after the large tip-clearance, however, a broad peak centred on approximately 1000 Hz is present (plots (c) to (g)). This broad band peak reaches a maximum just after the maximum tip-clearance. This coincides with the location at which the highest level of irregularity is observed in the raw data. The peak of the broad band is at 30% of the blade passing frequency.

The question now arises as to what sort of flow irregularity would give rise to a signal with a seemingly coherent frequency content. To answer this question, it is necessary to broaden the spatial extent of the measurements.

Figure 13 is made up of a series of contour plots of the unsteady pressure signal. The contours are from the middle set of pressure transducers in each of the eight circumferential positions (see Fig. 11). The ensemble-averaged blade passing signal has been removed from the raw data so that only the irregularities in the pressure field remain. Again, the data was taken at a stable operating point near stall (ϕ= 0.465).

In Fig. 13 (a), which shows data from just after the minimum tip-clearance, no significant disturbances are visible in the flow field. Note: In this region of the compressor, there will be no ramp-up in irregularity as the compressor approaches stall; as shown earlier.

As the pressure transducers are stepped round the annulus, and the tip-clearance gap grows, small, finite disturbances appear, as patches of intense low pressure (blue holes) just aft of the rotor leading edges. As the probes are moved to the maximum tip-clearance (Fig. 13 (d) and (e)), these disturbances grow in both size and intensity, covering half a blade pitch and the forward part of the chord. Figure 13 (e) shows the disturbances at the location where the irregularity level is highest. After this point, the disturbances decay as the probes are moved back into the small tip-clearance region.

In Fig. 13 (h), some small disturbances are still visible. This data is from a part of the annulus with approximately the same tip-

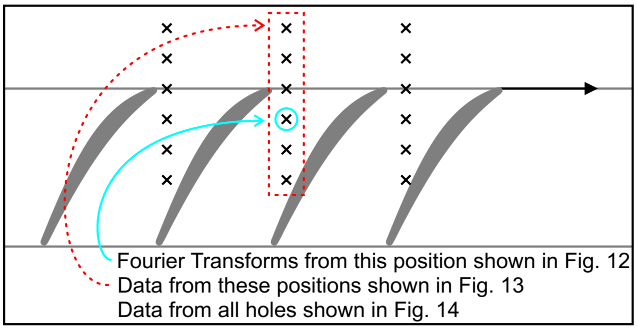


Figure 11: Measuring positions for detailed casing pressure measurements.

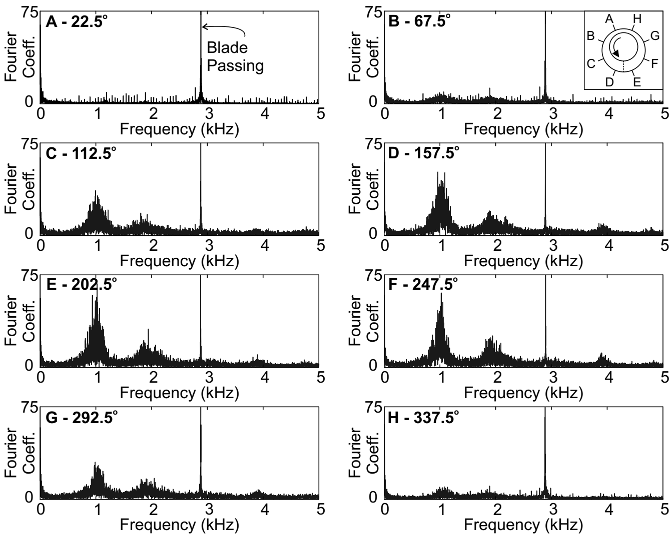


Figure 12: Fourier transforms from pressure transducers in different circumferential locations (3.3% tip-clearance, 75% eccentricity).

clearance level as Fig. 13 (a). The disturbance level here is, however, higher because the blades are moving towards the minimum tip-clearance not away from it. Again, this shows that the disturbances are linked to some form of dynamic effect, and not just dependent on the local tip-clearance.

The next point to consider is this: do the disturbances (the blue holes) form and die in quick succession, or are they coherent structures which propagate around the annulus?

To answer this question, data from the three closely spaced sets of pressure transducers (Fig. 11) were analysed for signs of propagation. Three contour plots are shown in Fig. 14, where the continued irregular spacing between the blue holes gives strong evidence of propagation. The propagation speed was found by measuring the time taken for each disturbance to pass from the first set of pressure transducers to the third set. From this, the average propagation speed was found to be 47% of blade tip speed (the speeds actually varied from 41 – 51%).

From the Fourier transforms, the dominant frequency of the disturbances can be seen to be 1022 Hz, which is just under a third of the blade passing frequency. The circumferential spacing of the disturbances can then be found by dividing the propagation speed by the frequency from the Fourier transform. This gives an average spacing of 1.3 rotor pitches. As the disturbances are propagating at just under 50% of rotor speed with a spacing of about 1.3 rotor pitches, they will appear at each measuring location every 2-3 rotor passings. This is consistent with the blue holes seen in Fig. 13 (d) to (f).

Similar disturbances were also seen in the concentric compressor with 3.3% tip-clearance, and are evident in data from the 1.7% tip-clearance tests with eccentric tip-clearance. In the tests at 1.7% clearance, the pressure troughs at the centres of the blue holes were approximately the same depth, but less frequent. They were also found to be of smaller circumferential and axial extent.

Some references to propagating pre-stall disturbances have been reported in the literature, but a clear picture has not really emerged. Mailach et al. [14] reported seeing "rotating instabilities" of similar frequency and propagation speed to those seen in the present work. However, similar disturbances were not seen in very similar experiments by Wisler et al. [15]. März et al. [5] also showed similar "rotating instabilities" in a fan with low hub-tip ratio and operating at part speed (again, see Wisler et al. [15]). Low pressure disturbances were also seen by Inoue et al. [16], though here the spacing between events was larger. Inoue et al. describe their disturbances as "mild stall". The characteristic given in the paper suggests that they were observing part-span stall.

It is believed that this is the first time that discrete pre-stall disturbances have been identified and shown to propagate in an isolated stage operating with a negatively-sloped characteristic.

APPLICATION TO A REAL ENGINE

In this section, data from a high-speed, multi-stage compressor will be examined for evidence of increasing blade passing irregularity as the compressor is driven towards stall. A discussion of engine life-cycle changes and the possibility of reliable stall warning will then follow.

High-speed Data

In order to examine the possibility of stall warning based on blade passing irregularity in a real engine, the data analysis technique was applied to measurements from a modern high-speed six-stage compressor with a design pressure ratio of 4.3. Data was acquired from 14 pressure transducers spaced over the six stages. Again, unsteady data was taken at stable operating points between the working line and the surge line, and the irregularity of the blade passing signature was measured. This was carried out at 79 – 90.5% compressor speed and with varying rates of bleed taken from between stages 3 and 4.

The irregularity level at a given operating point was found to be approximately constant through the compressor when the data was non-dimensionalised by the stage inlet total pressure. An example of this is shown in Fig. 15, which is a plot of irregularity against stage number at the working line and near stall for one of the test cases. The actual size of the fluctuations increases towards the rear of the machine, because of the rearward increase in density. The best stage for stall warning in terms of a consistent ramp-up in irregularity was found to be stage 5. This is not the stage that initiates stall in most cases; stall usually starts in stage 1.

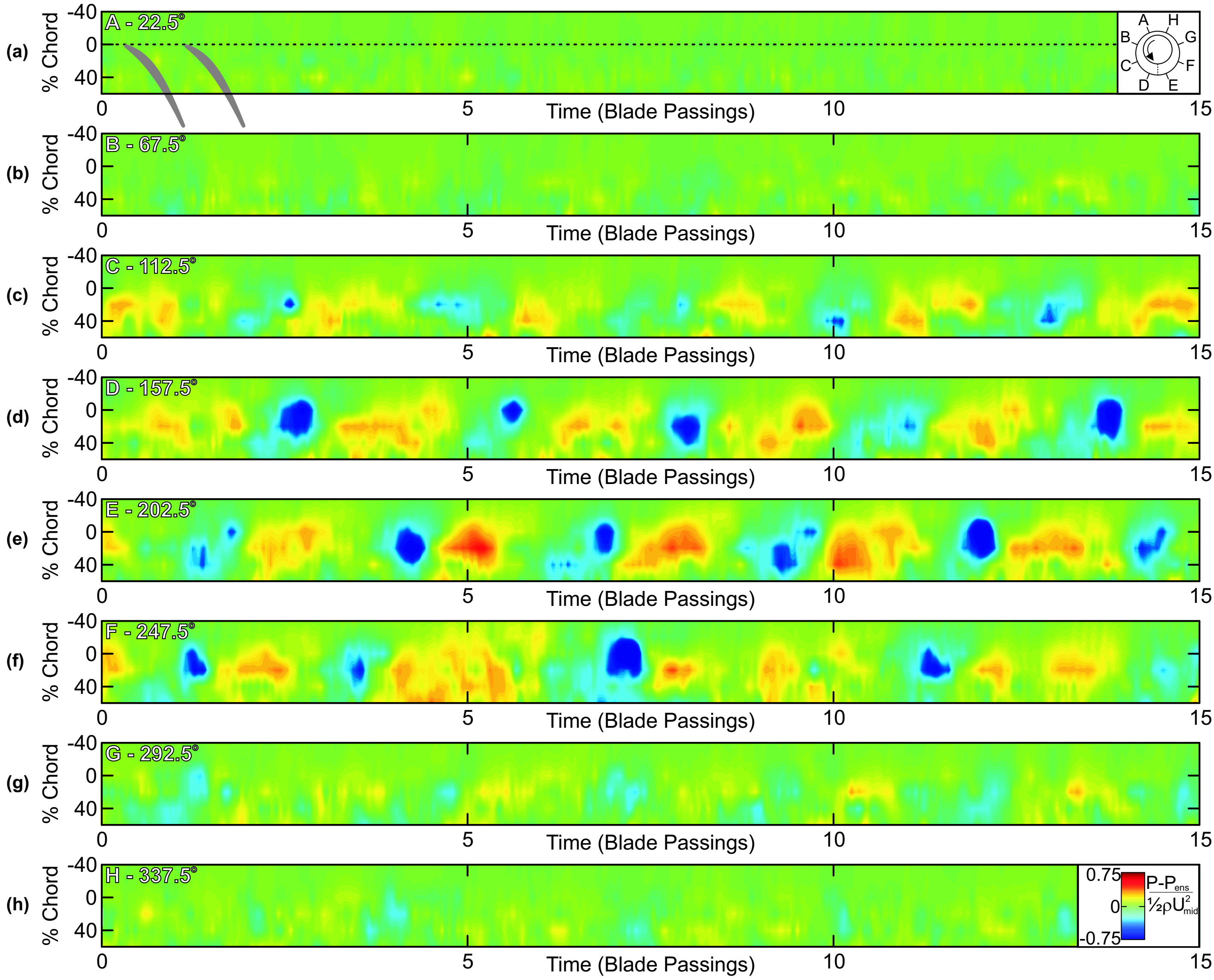


Figure 13: Irregularities in casing static pressure at different circumferential positions (3.3% tip-clearance, 75% eccentricity).
NB: Data sets were not recorded simultaneously, so disturbance tracking is not possible in this figure.

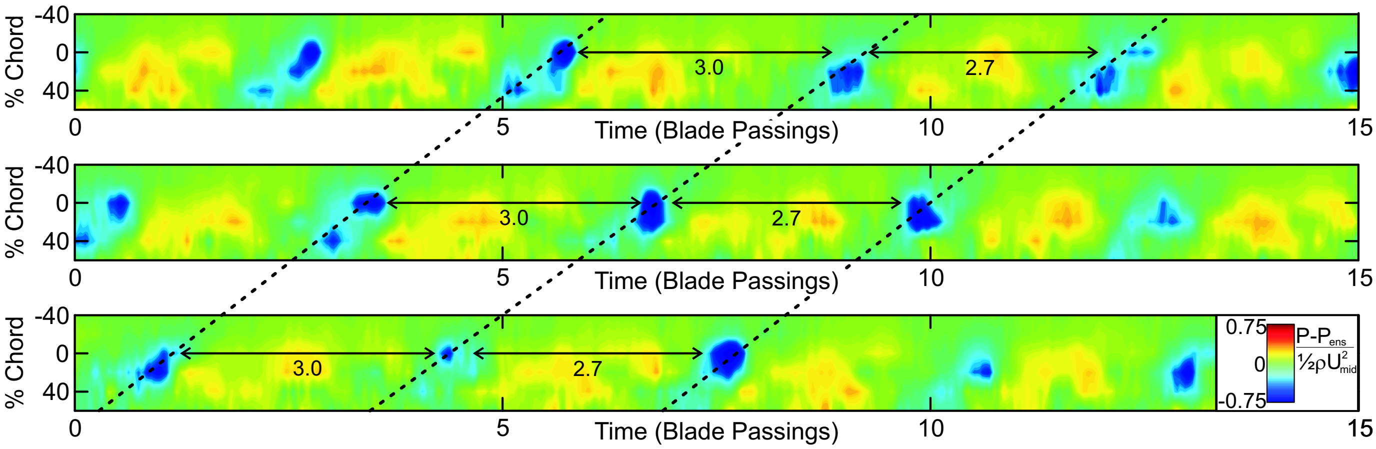


Figure 14: Propagation of "blue holes" in the large tip-clearance of the compressor (3.3% tip-clearance, 75% eccentricity).
Note: the variation in spacing between the blue holes remains the same as they propagate from one measuring position to the next.

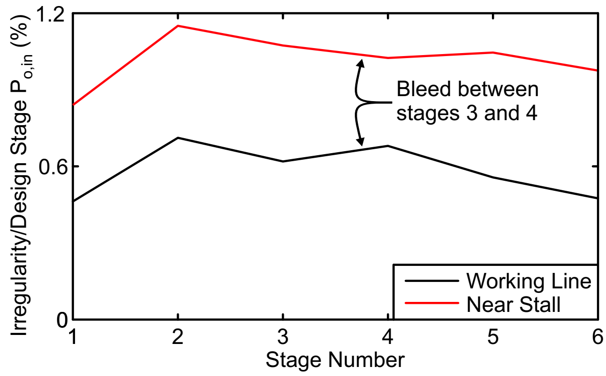


Figure 15: Irregularity against stage number for the high-speed compressor (84% speed, 3.5% bleed).

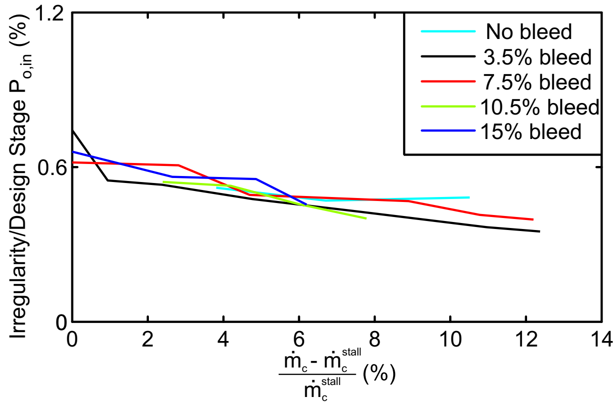


Figure 16: Irregularity against mass flow rate for a high-speed compressor at varying bleed rates (84%speed, stage 5 only).

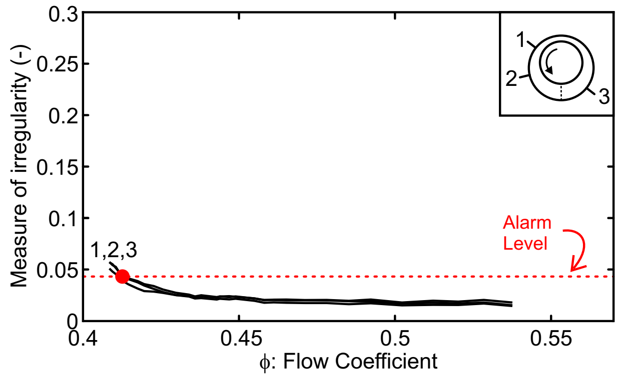
Figure 16 shows irregularity against corrected mass flow, $\dot{m}_c$, for one of the probes in stage 5. The data is from 84% speed with different bleed rates. A small ramp-up can be seen in each case, but the level varies between bleed rates. For effective stall warning, the alarm set-point would have to be adjusted each time the bleed rate changed.

From this data it can be seen that a multi-stage environment presents extra complications for stall warning as the irregularity level, and the ramp-up in irregularity, will vary from stage to stage, from speed to speed and from bleed rate to bleed rate.

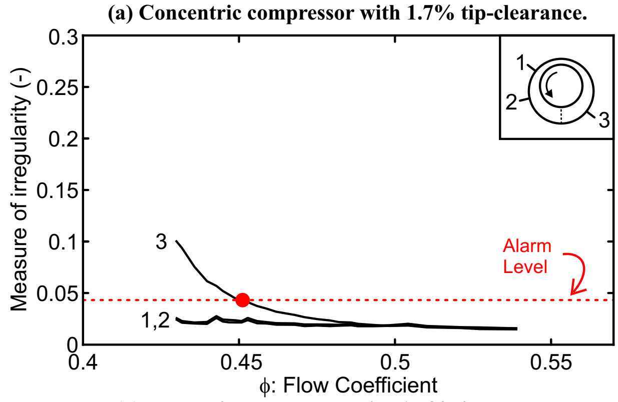
The Prospects for Stall Warning in a Real Engine

We now come to assess the usefulness of measuring the irregularity in the blade passing signature as part of a stall warning system in a real engine. To do this, we return to the data from the laboratory tests with different levels of tip-clearance and eccentricity and consider how these parameters will change over the life of an engine.

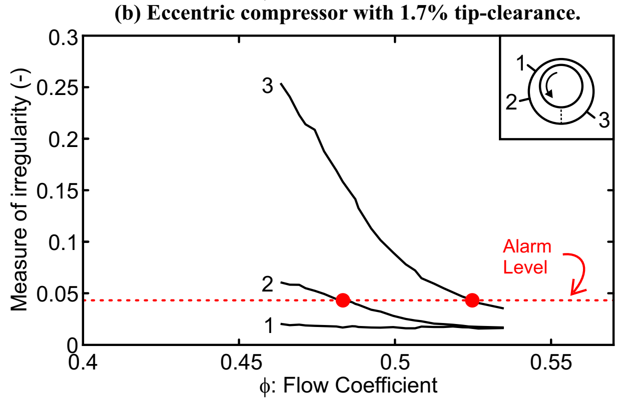
Initially, the engine will have a small tip-clearance, and it will be relatively concentric. The relationship between irregularity and flow coefficient for this case is illustrated in Fig. 17(a) (the results from three pressure transducers have been selected for clarity). The red dotted line shows an alarm level which will give warning when the flow coefficient is 1% above the stall point. At certain points during the flight cycle (e.g. at the point of rotation), the compressor is likely to become eccentric. Fig. 17(b) shows the



(a) Concentric compressor with 1.7% tip-clearance.



(b) Eccentric compressor with 1.7% tip-clearance.



(c) Eccentric compressor with 3.3% tip-clearance.

Figure 17: Changes in alarm point over the life of an engine.

same pressure transducers for a compressor with 75% tip-clearance asymmetry. This time, two of the pressure transducers will not trigger the alarm until the compressor has stalled, while the third, which is just after the maximum tip-clearance, will set off the alarm at a flow coefficient 5% before stall – an excessive margin.

Finally, towards the end of the engine's life, the blade tips will have rubbed and the tip-clearance will be larger. Fig. 17(c) shows irregularity against flow coefficient in a compressor with twice the normal tip-clearance and 75% eccentricity. In this case, one of the pressure transducers (in the tight tip-clearance) will fail to trigger the alarm, while the other two will trigger the alarm well before stall.

In a real engine, it would not be uncommon to find tip-clearance eccentricity of 50-75%. Indeed, the rotor blades often rub the casing at a particular circumferential location, implying

200% eccentricity. It can therefore be said that the eccentricity levels tested in this work are within the range seen in real engines. The tip-clearance levels are also representative of engine values.

From the measurements presented in this paper, it is clear that a stall warning system based on one pressure transducer at a fixed location would fail to give reliable results under different flight conditions. On the other hand, the use of more than one transducer could lead to any number of false alarms. Blade passing signature analysis is thus not the way to successful stall warning, but it has provided us with the interesting phenomenon of 'blue holes'.

CONCLUSIONS

A detailed study has been carried out of the seemingly erratic way in which the blade passing signature of each rotor blade in an axial compressor is different from its neighbour – and from itself in the previous revolution. In this paper, the relationship between the level of blade passing irregularity and proximity to stall has been linked to the important parameters of tip-clearance and eccentricity. At the same time, it has been shown that the rise in irregularity is attributable, at least in part, to the growth of pre-stall propagating disturbances. The following conclusions can be drawn:

1. For the first time, a systematic study of blade passing signature irregularity has been carried out in a compressor with different levels of tip-clearance and casing eccentricity. The key findings are:
   a. The level of blade passing signature irregularity in a concentric compressor, with small tip-clearance, is generally low and rises very little as the compressor approaches stall.
   b. In a concentric compressor with large tip-clearance, the level of irregularity is low at full flow, but ramps up noticeably as the flow rate is reduced.
   c. In an eccentric compressor, the level of blade passing irregularity in the large tip-clearance sector of the annulus will ramp up strongly as the compressor is throttled towards stall. By contrast, the irregularity will remain small and steady in the tight tip-clearance part of the annulus – surprisingly, this is true regardless of the actual size of the gap in this region.
   d. In an eccentric compressor, the blade passing irregularity will always be largest just after the blades pass the position of maximum clearance.
2. The above behavioural patterns appear to hold true in spite of the fact that, at large tip-clearances, the stall inception mechanism in the test compressor changed from spikes to modes.
3. It has been shown for the first time that the level of blade passing irregularity in an eccentric compressor is a function the average tip-clearance size and by the amount of casing eccentricity. There is a dynamic aspect to the level of blade passing irregularity which is not solely dependent on the size of the local tip-clearance gap.
4. An existing model for determining the circumferential variation in axial flow rate in an eccentric compressor (Graf et al. [10]) has been combined with measurements of stalling flow coefficient for different tip-clearances to explain why the maximum level of blade passing irregularity always occurs after the position of maximum tip-clearance.
5. Concentrated instrumentation and a new data analysis technique have lead to the identification of discrete pre-stall propagating disturbances which are partly responsible for the irregularity observed in the blade passing signature. The existence of such disturbances has been suggested in the past but now, for the first time, they have been individually identified and linked to the rise in blade passing irregularity seen before stall. (Note: the disturbances discussed here are a pre-stall phenomenon and are observable at flow coefficients well above the stall point.)
6. The pre-stall disturbances mentioned above are areas of intense low pressure, so called "blue holes," which are sub-pitch in size and occupy the forward part of the blade passage. The current measurements in a low speed compressor show that the blue holes form irregularly in quick succession, two or three blade pitches apart, and propagate at about 50% of blade speed. The blue holes are most easily observed in an eccentric compressor or in a compressor with large tip clearance.
7. The analysis of data from a high speed compressor has found a similar ramping up trend in the level of blade passing irregularity as was observed in the low speed compressor. This suggests that pre-stall propagating disturbances (blue holes) might also occur in a high speed machine, but clear identification was hampered by the lack of concentrated instrumentation. The behaviour of high speed compressors is made even more difficult by changes in speed and bleed rate.
8. The experimental results from this project show that the level of irregularity in the blade passing signature cannot be used to provide reliable stall warning in a real engine. Flight cycle and age related changes in tip-clearance and eccentricity will make the task impossibly complicated.

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