2012-01

Axial Transonic Rotor and Stage Behavior Near the Stability Limit

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http://hdl.handle.net/10945/44957

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1 Introduction

The trend in transonic axial compressors and their associated stages is toward higher pressure ratios and, by implication, higher speeds. This tends to result in machines with narrower operating ranges and so machines often run closer to their stability limits (i.e., near stall or surge lines) than in the past. This has made the accurate prediction of stall when designing new rotors even more critical. The current research focuses on the nonaxisymmetric behavior of transonic axial rotors and stages near to stall.

Previous work. A classical view of the operation of axial compressors holds that the flow field is steady or axisymmetric until operation at the stability limit. Detection of nonaxisymmetric instabilities of either spike or modal was described by Day et al. [1,2], and for the machine being studied, they indicated that unstable operation was imminent. Later research by Biela et al. [3] and Hah et al. [4] also encountered nonaxisymmetric flow phenomena near the stability limit of transonic machines but during sustained stable operation. Unlike the instabilities described by Day et al. [1,2], these nonaxisymmetric flow phenomena do not necessarily lead to stall or surge although behavior is similar to that of the modal instabilities described in Refs. [1,2]. Biela et al. [3] and Hah et al. [4] provided clear descriptions to the behavior of these nonaxisymmetric flow phenomena. They have a frequency in the region of 0.1–0.9 of rotor speed and are not measurable by traditional low-speed instrumentation such as static pressure ports and stagnation pressure probes. It requires high-speed (greater than 10 times the blade passing frequency) data acquisition systems and high-speed pressure probes. An earlier paper of Hah et al. [5] compares experimental and simulated results with the simulations also showing that prestall nonaxisymmetric flow phenomena can occur without leading to imminent stall. In most of the experimental work, high-speed pressure probes were embedded in the casing over the rotor blades [3,5,6]. Biela et al. [3] also had probes located downstream of the rotor in the stator leading edges. Their results show that these nonuniformities indeed penetrate the flow but they are greatest in the tip region of the rotor. Previous work by the current authors [6] has found similar behavior in a transonic axial rotor. The magnitude of these nonaxisymmetric flow phenomena is also found to be significant, in the order of a 2 deg variation of the rotor relative inlet flow angle.

Nonlinear characteristics. A comprehensive outline of the characteristics of the low-frequency phenomena is presented in Ref. [6] but a brief summary is given here. Measurements of these low-frequency signals at 70–100% of design speed show that they are normally dominated by one or two frequencies, and their overall trend is to increase in magnitude as the stall or surge line of a machine is approached. At various speeds, there are exceptions to this trend with the magnitude falling as stall is approached. The exact cause of this was unknown but it was usually associated with a change in the dominant frequencies that emerge between 0.1 and 0.9 of rotor speed. It is thought that the underlying causes of these low-frequency phenomena may change as stall is approached thus causing the observed frequency changes.

Significance. It was thought that these low-frequency signals may be useful in detecting stall but the blade passing frequencies are probably more suitable for this, as shown by Tahara et al. [7], in a low-speed machine. There are two main implications of these nonaxisymmetric phenomena. If safe operation of compressors near to the stability limit is desired, then a good understanding of their effects and influence on stall is required. The second significance is during the design phase where, traditionally, the prediction of stall has been difficult. The exclusion of these low-frequency nonaxisymmetric phenomena through the use of the axisymmetric assumption in single blade passage simulations may result in the overprediction of the stability limit of a machine. The use of full-rotor time accurate simulations would eliminate
this uncertainty but is computationally expensive especially during the design phase when many geometry iterations may be needed. A better description of these low-frequency phenomena may lead to less computationally expensive methods by approximating their effect in single passage flow simulations.

Rotor only versus stage. Much of the work in this area has tended to focus on a single type of machine. The previous work by Gannon and Hobson [6] was for rotor-only, while the work by Biela et al. [3] and Hah et al. [4] was for a rotor-stator stage. Thus, this study aims to investigate the difference in characteristics of the low-frequency nonaxisymmetric flows between a rotor-only and rotor-stator stage where the rotor is common to both. Obviously, the inclusion of a stator relatively close to the rotor will affect the flow through a nonuniform change of the downstream blockage. This study shows that there is a change in the behavior of the low-frequency phenomena between a rotor-only and rotor-stator stage both operating near the stability limit. The results show that the stator nonaxisymmetric phenomena tend to be of a lower magnitude than for the rotor-only.

Higher order analysis. In this study, longer data samples were taken (2 s versus 0.5 s) that showed higher frequencies between the first and second orders of the rotor speed were present. This would seem to indicate that these low-frequency phenomena can occur in pairs as it is commonly accepted and physically reasonable that they travel at less than rotor speed. The increase in sample length equated to 900 revolutions of rotor data compared with 225 revolutions in previous work [6].

2 Test Facility

The turbomachinery used was the Sanger designed stage [8] shown in Fig. 1. It is well documented with the steady-state performance, test rig, and instrumentation presented in Gannon et al. [9]. To summarize, the rotor and the stator consist of 22 and 27 blades, respectively. The design tip speed is 396.2 m/s with rotor-only and rotor-stator total-to-total design pressure ratios of 1.61 and 1.56, respectively. The design mass flow rate is 7.75 kg/s. An abradable strip is placed over the rotor to ensure a minimal tip gap details of which are given in Ref. [6]. For the rotor-only case, a ring with the same hub profile replaced the stator. It should be noted that for the rotor-only case, the machine stalled but in the case of the rotor-stator stage, it surged.

Performance. The steady-state stagnation pressure ratio for the rotor-only and stage is shown in Fig. 2. The machine performance is fairly typical for a transonic rotor with the speed-lines becoming steeper at higher speeds. Comparison of the total-to-total pressure ratios shows that the stator causes a noticeable stagnation pressure drop once the rotor is operating beyond its peak efficiency. The stage stalled at a lower pressure ratio than the rotor at all speeds. At 100% speed, the stage stalled at a noticeably lower mass flow than the rotor-only case when compared with the other speeds.

Figure 3 show the isentropic efficiencies of the rotor-only and stage cases. The stator results in a lowering of the isentropic efficiency of between 2% and 4%. The rotor-only peak isentropic efficiencies lie between 85% and 88%, while the stage efficiencies lie between 83% and 85%.

Figure 4 shows the relative inlet Mach numbers for each speed-line. At 80% speed, the relative inlet flow at the blade tips is nearly sonic at the full open throttle condition, while 90%, 95%, and 100% speeds are in the transonic region.

Instrumentation. The instrumentation was largely unchanged from the previous test program that was documented in Ref. [6] and summarized here. The steady-state data acquisition system consisted of two upstream and 10 downstream stagnation temperature-pressure probes and an additional 10 downstream stagnation pressure probes. Mass flow was measured by an upstream flow nozzle. The transient data acquisition system consisted of high-speed Kulite pressure probes positioned over the rotor. These were sampled at 196 kHz, giving 19.8 samples per
blade passage. For this study, an additional probe (no. 10) was added to the trailing edge of the rotor as this gave a more complete view of the pressure field. The Kulite probe positions relative to the rotor and local hub-tip ratios are shown in Fig. 5. The Kulite probes were calibrated through the use of static pressure ports located at the same axial location but on the opposite side of the rotor. An additional impact Kulite was placed 5.4 axial chords upstream of the rotor. This was at the same axial location as the inlet stagnation pressure probes and was calibrated against these.

3 Experimental Method

The investigation of the low-frequency phenomena was performed through the capture of data over the complete range from 70% to 100% speeds for the rotor-only and stage cases. Data were taken at all the points shown in Figs. 2–4. After post-processing of the previous data [5], it was found that it was better to take longer high-speed pressure samples from the Kulite probes as this allowed for better resolution in capturing the frequencies of interest. The sample time was increased from the previous 0.5 s to 2 s, which at 100% speed resulted in a sample covering 902 revolutions. The 11 channels typically generated 100 Mb data sets per point and meant that the fast-Fourier-transforms (FFTs) required more time to complete than the previous samples, 2 min versus 30 s. For the detection of the nonaxisymmetric flow phenomena in this rotor, it was found that sampling of more than 900 revolutions (2 s) was sufficient vice previous analyses over 225 revolutions (0.5 s), which were inadequate.

Before taking data, the machine was allowed to settle to ensure that the overall conditions were as near to steady-state as possible. The stall line was approached in small increments with the last stable operating point shown from Fig. 2 to Fig. 3 at the end of each speed-line.

4 Results

It is useful to investigate the data in both the time and frequency domains as each offers insights into the complex tip region flow field. In the time domain, it is possible to investigate the bulk changes in axial inlet Mach number and relative inlet angle. In the frequency domain, the relative signal magnitudes and their distribution over the rotor can be investigated. Previous studies have focused on viewing the pressure structure over the rotor blade and it is instructive to include such a figure here. Figure 6 shows the wall pressure contours with the Kulite tracks superimposed. Kulite no. 2 is positioned at the leading edge of the rotor blade, and no. 10 is just past the trailing edge. As this is the 100% speed of the rotor-only case, the rotor is highly loaded and contains a strong tip vortex that distorts the detached normal-shock wave as shown in Fig. 6. Successive images of each passage can be put together to make an animation file. From these, it can clearly be seen that the flow, while maintaining the basic structure shown below, is different from passage to passage. It is this passage-to-passage difference that we wish to investigate.

Post-processing. All data were reduced using the programming language of MATLAB. The Kulite transducers measure the near instantaneous wall static pressure as the rotor blades travel past, while a traditional static pressure port with a long tube to the pressure transducer will measure the mean static pressure. Using this mean value and the stagnation pressure upstream, the bulk Mach number can be found. Using the high-speed data but only looking at its low-frequency components allows the variations in the bulk Mach number to be calculated. Digital signal processing techniques are used (c.f. Gannon and Hobson [6]) for two reasons. First, narrow band-pass filters with very steep roll-offs are required. Second, the band-pass frequencies are very low, \( \approx 1/22 \) blade passing frequency relative to the sampling rate of 19.7 times the blade passing frequency.

As with any signal-based data, it is useful to first perform a spectral analysis of the entire set. Figure 7 shows the spectral analysis from all the high-speed pressure probes superimposed onto a single axis for the 100% speed near stall rotor-only case. As is typical, the blade passing frequency and its harmonics are predominant. Imperfections in the rotor blisk show up as a clear once-per-revolution signal with associated harmonics. With the increase in sample length from 0.5 s to 2 s, the nonaxisymmetric...
frequency regions between 0.1–0.9 and 1.1–1.9 of rotor speed can be more clearly seen. In fact, with the previous 0.5 s samples, the existence of the second higher frequency non-axisymmetric flow regions from 1.1 to 1.9 was in some doubt.

While the frequency domain analysis allows one to easily find regions of interest and also view the distribution of the frequencies through the rotor [6], it does not always lend itself to easy interpretation of its physical significance. Once frequency regions of interest have been identified, they can be isolated using the filter shown in Fig. 8 with its associated performance and band-pass regions. The extremely strong roll-off at the edges of the band-pass regions can be seen, which is aided through the use of double precision in the analysis programs. As mentioned, only the single pass region from 0.1 to 0.9 was used previously but Fig. 8 shows the second pass region from 1.1 to 1.9. Higher order signals of the kind being sought here were not evident or were of such a small magnitude as to have little effect and so additional band-passes were not required.

After filtering a particular signal, this filtered signal can then be transformed back into the time domain using an inverse FFT. Again these are not a computationally expensive, each taking around 2 min to complete on a personal computer.

Figure 9 shows an unfiltered and filtered pressure signal. The signal is taken from Kulite no.1 upstream of the rotor, as shown in Fig. 5, for the rotor-only case at 100% speed. The data are non-dimensionalized to the inlet stagnation pressure. Just two revolutions of data are shown as this allows the difference between the filtered and unfiltered signals to be observed. There are 44 cycles in the unfiltered data, with each peak corresponding to a blade passing. As expected, the filtered signal’s amplitude is small in comparison to its parent signal and its frequency much lower. The filtered signal’s trend can be seen in the unfiltered data.

We now plot just the filtered signal over a longer range of data in Fig. 10. It is not a pure sinusoid but rather composed of a number of overlapping frequencies, and as mentioned previously, if the flow were purely axisymmetric the filtered signal would be constant. We may assume that the inlet stagnation pressure remains constant as no work is done to the flow before it enters the rotor. Using the constant inlet stagnation pressure and the filtered static signal, it is then possible to calculate the low-frequency changes to the bulk inlet axial Mach number, as shown in Fig. 11. As can be seen, it is a near inverse function of Fig. 10, which is to be expected. This method is summarized here for completeness but is outlined in more detail in Ref. [6]. As mentioned, the mean values of the relative inlet Mach numbers are shown in Fig. 4.

The corresponding peak-to-peak variations in relative inlet Mach number are shown in Fig. 12 for the rotor-only and stage cases. This peak-to-peak variation is the maximum found over 900 revolutions of the rotor at 100% speed. Figure 11 only shows 10 revolutions of data, hence its lower apparent peak-to-peak value. All later results are related to these two figures, so it is useful to pause and observe their characteristics. In general, the rotor-only case behaves in a much more predictable manner with

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**Fig. 7** Rotor-only 100% speed spectral analysis

**Fig. 8** Filter performance and band-pass regions

**Fig. 9** Unfiltered and filtered pressure signal

**Fig. 10** Filtered pressure signal
a peak-to-peak variation of between 0.08 and 0.1 Mach number when operating near to stall. The stage case behaves in a similar way but there is much greater variation in the magnitude of the changes as stall is approached, most notably at 80% speed.

Figure 13 presents data from multiple high-speed probes and shows the frequency and spatial distribution of the low-frequency phenomena over the rotor-only and stage at 100% speed, near stall. It must be noted that the vertical scale of the rotor-only case is an order of magnitude larger than for the stage. The propagation region appears to be at the same axial location and frequency (30% of rotor speed) in both cases. In the rotor-only case, the signals propagate downstream of the rotor, while in the stage, they attenuate rapidly. This seems to indicate that the stator damps out the low-frequency nonaxisymmetric flow phenomena. This premise would support the behavior of the observed signal powers in Fig. 14.

Signal powers. Some useful insights into the difference in behavior between the rotor-only and stage cases can be gained by looking at the root mean square (rms) powers of the components of the measured signals. We have one raw unfiltered signal and a second signal from the dual band-pass filter. In both cases, the constant or, to use electrical terminology, the direct current parts of the signal are removed. In calculating the rms power of a signal about its mean, we measure the power of the unsteadiness of the signal. The results shown here contain the sum of the rms signals from all the Kulite probes. As the low-frequency signals are detected over the entire rotor tip region it was thought simplest to investigate the rms signals in this way. Figure 14(a) shows the rms power of the band-pass filtered low-frequency nonaxisymmetric signals for the rotor-only case. It can be seen that there is a trend upward for all speed-lines as the rotor moves closer to its stall line. Figure 14(b) shows the same data for the stage case, and it can be seen that the magnitude is approximately 30% of the rotor-only case. While there is some upward trend in the values for the 70% and 80% speed-lines as the stall point is reached, the transonic speed-lines (90% speed and above) do not behave in the same way as the rotor-only case. It is thought that the stator acts as a damping mechanism to the low-frequency flow phenomena.

Figure 15 shows the total rms powers for the two cases. The stage tends to have a higher total rms power than the rotor for the same mass flow rate at a particular speed. This is probably due to the rotor being more highly loaded in the stage due to the presence of the stator. The overall trend is for the rms power to increase as
stall is approached but there are clear exceptions to this. The most notable being at design speed where the power in both cases drops as operation moves toward the stall line.

Figure 16 shows the nondimensionalizing of the power in the low-frequency instabilities for the rotor-only and stage cases. Their power is plotted as a fraction of the total power along each speed-line. No obvious trend emerges with each line behaving quite differently.

Relative inlet angle. It is also useful to investigate the variation in relative inlet angle into the rotor blade. To calculate this, we first make the assumption that the rotor’s angular speed remains constant, which based on the measured once-per-revolution signal is reasonable. With the rotor tip Mach number constant and the variation in axial Mach number known, a basic velocity triangle analysis will give the transient rotor blade relative inlet angle.

Figure 17 is a velocity triangle diagram showing the actual variation in inlet flow angle for the 100% rotor-only case at the blade tip shown. In red are the mean, maximum, and minimum relative inlet flow angles for the near stall case, while the blue is for the choked flow case. As mentioned for this rotor-only case, near stall a 1.9 deg variation in the relative inlet flow angle compared with a total variation of 5.6 deg. This figure illustrates the very narrow operating range of the tip flow of a typical transonic compressor. The total range for the mean relative inlet angle at the blade tip only covers 11 deg from choke to near stall over the entire speed range (67–78 deg).

The mean relative inlet angle is the same as that calculated if standard wall static and stagnation pressure probes to calculate the mean inlet Mach number were used. Figure 18 shows the mean-to-maximum inlet angle variations over the entire data sample for each data point for the rotor-only and stage cases. In general, the variation in angle is greater for the rotor-only case. As mentioned previously, this could be due to the stator damping the system by reducing the volume available behind the rotor for the transient signals to amplify in. The exception is for the 80% stage speed-line that has a near 2.5 deg variation in the inlet angle when operating near to stall. While these angle variations do not appear large, around 2 deg, they are significant when compared with the entire 11 deg mean operating range.

Figure 19 shows the maximum inlet flow angles measured over an entire 2 s sample. They are simply the mean inlet angles added to the mean-to-maximum variations in Fig. 18. When comparing these two figures, it is interesting to note that the maximum inlet flow angles for the rotor-only and stage cases agree along each speed-line, except for, once again, the 80% stage speed-line. When positioned in the stage, the rotor proves more resistant to stall for this speed-line. For the other cases, the rotor-only and stage stall occurs for almost the same maximum inlet flow angle to within 1 deg when operating near stall. Note that from Fig. 11, the variation in flow angles is clearly not symmetric about the mean. This is significant as it is assumed that the maximum relative inlet angle will be the one to cause stall.

5 Discussion

The exact reasons for the low-frequency signals are not known but they clearly exist in axial turbomachinery type flows. The flow in one passage seems to affect the flow in the following passage even in transonic machines. Due to the geometry of a turbomachine, the signal is allowed to amplify as it travels around the rotor. A linear cascade or single passage simulation would not allow the disturbance to amplify.

Signal powers. It was thought that by measuring the rms power of the low-frequency nonaxisymmetric signals, it might be possible to detect how far a rotor was operating from stall in some useful way. Figure 14(a) confirmed the possibility of using these signals as there was a reasonable trend of increasing low-frequency signal rms power toward the stall line as noted.
earlier. However, once the stator is included in the machine, the trend disappears and the powers become fairly flat for each speedline. While it is thought that the low-frequency signals are important in the engine behavior near to stall, it is thought that their detection and use in an engine monitoring system may be difficult. It may be simpler to observe the high frequency part or the total power of the signal. First, the trends in total power have a more predictable pattern as is shown in Fig. 15. The second thing to be considered is that while a high sampling rate will always be needed for any transient casing pressure measurement, the detection of higher frequency signals does not require the long samples that the detection of low-frequency signals requires, 0.25 s versus 2 s for the current machine. It may be more effective to simply compare a measured pressure trace with the expected one [7], which would detect spike-type instabilities that are typical just prior to stall [1]. In the current research compressor, a control system response would have to be fast enough to detect an instability within one rotor rotation, a time of 2.2 ms at 100% speed. It is noted though that on very large engines, for the same tip Mach number, the engine rotational speed is lower than for the smaller
test compressor presented here. The required response time would be lower, and this would lessen the data processing requirements.

Initial observations made by nondimensionalizing the rotor-only data at 100% speed as shown in Fig. 16(a) were promising. The fraction of signal power due to the low-frequency instabilities increased as stall was approached. However, looking at both Figs. 16(a) and 16(b) shows the importance of reviewing the entire data set. Each speed-line behaves in a different way, and the authors were not able to suggest any useful pattern that could be used to detect the proximity to stall for either the rotor or stage cases.

**Improved stability limit prediction.** By observing the frequency of the nonaxisymmetric characteristics of the flow field relative to the speed of the machine, we can gain some useful insights. A typical peak-to-trough of one cycle shown in Fig. 11 will occur over 0.75 of a revolution of the rotor. Knowing the relative inlet flow speed allows us to make a rough calculation and observe that this would be near 14 chord lengths of flow for a single rotor blade [6]. This is at too low a rate to be considered a type of oscillatory flow such as that found in airfoils with rapidly changing angles of attack. Each blade and passage will experience a gradually changing inlet flow field. Considering the classical “parallel compressor theory” [10], we assume that each individual passage of the rotor is operating along a speed-line. In Fig. 2 and Fig. 11, two points (a) and (b) have been highlighted at axial Mach numbers 0.394 and 0.444, respectively (compared with a mean of 0.423). As the change in flow field is relatively slow, it is likely that a steady-state single passage simulation would reasonably predict the conditions at (a) and (b) for their respective axial Mach numbers. If we accept that a steady-state single passage simulation will predict the flow field reasonably accurate for a given inlet Mach number, it may be possible to make better estimates of the stall and surge lines based on this computationally inexpensive simulation. A possible method is outlined here.

Table 1 shows the experimentally measured difference between the maximum and mean inlet flow angles at near stall condition. The average values are 1.77 deg for the rotor-only case, 1.31 deg for the stage case, and 1.54 deg for all values. Let us suppose a steady-state single passage computer model with no additional stall correction model was used to simulate a compressor’s performance along a particular speed-line. We would usually expect the predicted stall margin to be too large. It may be possible to reduce this error by using the above variations in inlet flow angles. A single passage steady-state simulation forces the inlet flow angle to be constant and may optimistically predict the stall point as being near the maximum transient relative inlet flow angle observed in Fig. 19. It is suggested that a correction be applied to this type of prediction based on the measured variation in inlet flow angle. The value of this correction would vary based on the machine but it is suggested that it is in the range of 1.1–1.9 deg, which is one standard deviation about the mean of the above data in Table 1. This may yield a slightly more accurate prediction of the stability limit without resorting to computationally expensive full rotor transient simulations.

### 6 Conclusions

A full data set showing the behavior of low-frequency nonaxisymmetric flow phenomena for a rotor-only and rotor-stator stage sharing a common rotor is presented. A spectral analysis showed these nonaxisymmetric characteristics to be present between 0.1–0.9 and 1.1–1.9 of rotor speed in both rotor-only and stage cases. Above this range, only once-per-revolution or blade passing frequencies is significant. The magnitude of the low-frequency signals was much greater for the rotor-only case and generally increased as the stall line was approached. In the case of the stage, the power of the signals remained constant as stall was approached. The effect of the low-frequency instabilities was investigated in the time domain by calculating the magnitude of the change in the relative inlet flow angle. It was found that the variation in angle was generally greatest in the rotor-only case, which agreed with the power of the low-frequency signals. The only exception to this was at 80% speed for the stage where the compressor is sonic near the rotor tip. Of significance was the fact that the peak relative inlet angle measured before stall agreed well between the rotor-only and stage cases. The low-frequency nonaxisymmetric phenomena are present in both the rotor-only and stage cases and should be taken into account when trying to obtain accurate simulations near the stall points. Stability limit predictions based on steady-state single passage simulations may be improved by assuming a 1.1–1.9 deg reduction to the rotor relative inlet angle.

### Acknowledgment

This study was part of the compressor research program sponsored by the Propulsion and Power Department of the Naval Air Warfare Centre, Patuxent River, MD, with Ravi Ravindranath as the technical monitor.

### References


Table 1 Largest difference between mean and maximum inlet flow angle at near stall condition

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<tr>
<th>Speed</th>
<th>70%</th>
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<th>90%</th>
<th>95%</th>
<th>100%</th>
<th>Mean</th>
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<td>1.8 deg</td>
<td>1.8 deg</td>
<td>2.0 deg</td>
<td>1.3 deg</td>
<td>1.9 deg</td>
<td>1.77 deg</td>
</tr>
<tr>
<td>Stage</td>
<td>1.1 deg</td>
<td>2.1 deg</td>
<td>1.3 deg</td>
<td>0.8 deg</td>
<td>1.2 deg</td>
<td>1.31 deg</td>
</tr>
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