AN EXPERIMENTAL INVESTIGATION OF SYSTEM EFFECTS
IN AXIAL FLOW COMPRESSORS

Peter D. Silkowski
Pratt & Whitney Aircraft Engines
400 Main Street M/S 163-17
East Hartford, CT 06108

Hyoun-Woo Shin
GE Aircraft Engines
One Neumann Way, Mail Drop A-411
Cincinnati, OH 45215

ABSTRACT

In this paper, experimental results are presented which demonstrate the potential importance of component coupling and system effects on the pumping stability and performance of a rotor in an axial flow compressor. Three different configurations are presented: (1) series coupling, (2) parallel coupling, and (3) combined series/parallel coupling. In all three configurations the more stable component or region successfully stabilizes the entire compression system beyond the stability boundary of the less stable component or region. A novel method for assessing the relative stability of the different regions is employed. This method utilizes the small scale pre-stall disturbances of the compressor to probe the stability of the different regions. The role of length scales or domains of dependence in determining stability is consistently demonstrated. Specifically, the stability of small length scale disturbances is shown to be governed by the local environment while being relatively insensitive to other components. However, the stability of larger length scale phenomena is affected by the global environment created by considering all of the parts of the overall system. Consideration and knowledge of these results is important when modeling, analyzing, or designing a compression system. Furthermore, these results serve as a cautionary note when interpreting solutions from a simplified analytical/computational analysis.

INTRODUCTION

In an effort to simplify compression system models, various assumptions are often made. Typical examples include the assumptions of an isolated blade row or an idealized compressor of uniform tip clearance and inlet conditions operating in an infinite duct. While these assumptions do simplify the model, as well as the subsequent analysis, they do not truly reflect the operating conditions of an axial flow compression system. In fact, an axial flow compression system is composed of several parts which may interact and thereby affect one another's performance. For example, consider a single rotor row. In reality, this component does not function in isolation. The rotor may be affected by inlet or exit conditions created by upstream or downstream components, or by tip clearance variations created by a rubbed out or imperfect casing.

Recent experimental and analytical work has demonstrated that neglecting these system effects may lead to incorrect predictions of the compressor's behavior. Greitzer et al. (1978) have shown how the presence of different downstream components (nozzle or diffuser) can influence the compression system's response to an inlet distortion. Similarly, in the field of aeroacoustics, Hanson (1993), (1992), Meyer (1993), and Topol et al. (1987) have demonstrated how the presence of multiple blade rows and inlets can affect the radiated noise. Finally, Hall and Silkowski (1997) and Silkowski and Hall (1997) have examined the impact of neighboring blade rows on the aeroelastic stability of a given rotor. These examples demonstrate how system effects can alter the pumping, aeroacoustic, and aeroelastic stability as well as affect the overall performance of a compression system.

To further examine this role of system interaction on compressor stability and performance, a series of experiments were performed on a four stage, low speed, large scale, axial flow compressor rig. The next section details the experimental facilities and the three configurations that were...
examined. The performance and stability characteristics of the baseline configuration are then presented. With this information as a benchmark, results from the three coupling experiments are discussed. All three experiments highlight the importance of considering the entire compression system and guarding against incorrect conclusions based on idealized analysis of just one portion of the system.

EXPERIMENTAL FACILITIES

Compressor and Instrumentation

These experiments were conducted at the GE Aerodynamics Research Laboratory on the Low Speed Research Compressor (LSRC) rig. This facility utilizes aerodynamic scaling to examine compressor blading at low speed but on a larger scale (five foot diameter). This larger scale makes instrumentation easier and relatively less intrusive. Four stages of the GE/NASA E^3 blading were used to represent a modern multistage high performance axial flow compressor. More detailed information on the facilities and the blading can be found in the report by Wisler (1981).

An aggressive array of instrumentation was employed in these experiments. Beyond steady state flow and pressure rise instrumentation, an array of forty high response probes was also used. This array was composed of twenty hot wires for unsteady velocity measurements and twenty pressure transducers for unsteady pressure measurements. The pressure transducers were located in the casing while the hot wires were attached to probes that were then inserted through the casing into the gas path. This allowed the hot wires to be put at a variety of radial immersions, such as near the hub at 80% immersion, or near the tip at 20% immersion. Ten hot wires and ten pressure transducers were fixed at the inlet guide vane exit for all of the experiments as a set of reference instrumentation. These reference wires were at 50% immersion, midspan. The remaining ten hot wires and ten pressure transducers were used at several axial locations, but focus was given to rotor 1 inlet and exit. Each group of ten probes was circumferentially distributed as follows. Eight of the probes were equally spaced around the annulus, at locations: 0°, 45°, 90°, 135°, 180°, 225°, 270°, 315°. The remaining two hot wires were located at 15° and 25°, while the remaining two pressure transducers were located at 200° and 210°. This arrangement allowed for tracking circumferentially propagating disturbances. The eight equally spaced probes provided full annulus resolution and allowed circumferential Fourier transformation as a post-processing tool. The closely spaced probes provided a concentrated region of higher circumferential resolution and allowed cross-correlations to be computed over a wider range of circumferential offsets. More information on the details of the instrumentation can be found in the report by Silkowski (1995).

Multiple Configurations

First, the baseline configuration was investigated to provide a benchmark of this particular compression system's pumping and stability characteristics. Then, three variant configurations, each exhibiting a different form of coupling, were investigated in turn. The three forms of coupling created were coupling in series, in parallel, and combined series/parallel coupling. These parallel and series definitions are defined with respect to the gas path and are analogous to their meanings in electric circuits. Coupling in series was achieved by restaggering the rear three rotors closed by 10°. In this manner the rear three stages represent a different compressor with a different design point operating in series with the first stage. Coupling in parallel was achieved by opening the tip clearance over a fraction of the circumference. The nominal system had a tip clearance gap-to-span ratio of 1.3% and this was opened to a gap-to-span clearance of 3.1% for a 120° section over the first rotor. Finally, a combined series/parallel coupling was achieved by using a fixed inlet distortion screen over the same fraction of circumference that had been altered in the tip clearance experiments. Specifically, the distortion screen was fixed 1.5 radii upstream of rotor inlet and covered the entire span over a 120° circumferential region. For a more complete discussion on the use of the distortion screen, see the papers by Longley et al. (1994) and Plumley (1990). The three variant configurations are also referred to as the mismatch, notch tip clearance, and fixed distortion screen builds, respectively. Figure 1 shows the steady state speedlines for the different configurations, where ψ is the total-to-static pressure rise coefficient and φ is the flow coefficient. Each of these configurations will now be examined in detail.

RESULTS

Baseline Results

At low flows for a given speedline this particular compression system experiences what is often referred to, for example by Day (1991), as "pip" type rotating stall inception. Specifically, detailed measurements by Silkowski (1995) reveal that for the nominal configuration the pre-stall pip is a three dimensional disturbance traveling in the rotor rotation direction at roughly 70% of the shaft speed. The circumferential extent of the pip is less than three rotor blade passages. Within the pip there is a radial redistribution of flow with a flow deficit in the tip region and a compensating flow increase in the hub region.

The pip's radial redistribution and circumferential
The circumferential character of the pip can be discerned from Fig. 3, which shows a time trace on an expanded scale. This measurement was taken at the tip immersion of 20% at the rotor inlet. The pip’s flow deficit is clearly visible, and by comparing it’s extent with the blade passing signal, it is seen that the pip spans less than three rotor blade passages. This pre-stall pip and its behavior prove to be useful metrics in comparing the relative stability of different compression system regions and configurations.

Series Coupling

To demonstrate the effect of series coupling, consider the mismatch configuration relative to the nominal configuration. The stalling flow coefficient of the steady state

propagation are illustrated in Fig. 2 which shows time histories of the unsteady velocity at the rotor inlet. Each time trace has been normalized, and the mean value has been replaced by the respective probe number. Only the eight equally spaced probes are shown. Note that the radial positioning of the probes alternates between the tip at 20% immersion and the hub at 80% immersion. Rotor rotation and pip propagation are in the direction of increasing probe numbers. From this diagram, the circumferential propagation of the pip may be detected as it passes by each successive probe. A guideline has been superimposed on the figure to highlight this propagation. The slope of the guideline is just one way in which the pip’s circumferential speed may be calculated. The pip’s speed of propagation was also confirmed by both Fourier transform and crosscorrelation calculations. Note that the pip is first visible as a tip flow deficit (valley) on the first probe at a nondimensional time of about 77.25. However, the pip has not sufficiently grown, and the radial redistribution is not sufficient enough for the pip to be seen at the hub until it reaches the eighth probe at a nondimensional time of roughly 78.5. At this location, the pip appears as a peak, or flow increase, compensating for the tip flow deficit. Note that this pre-stall pip quickly grows, in under five rotor revolutions, into a mature rotating stall cell traveling at 45% of the shaft speed.

Figure 1. Summary of stage 1 characteristics for various builds.

Figure 2. Alternating hub/tip velocity at rotor inlet vs. time as the compressor is throttled down into stall. Eight equally circumferentially spaced hot wires, normalized data, baseline build. Note disturbance is first visible as a deficit in the tip region at t = 77.25

Figure 3. Velocity at rotor inlet vs. time for fixed operating point. 20% immersion, mismatch build, baseline stall point. The pip is roughly 3 pitches wide.

Series Coupling

To demonstrate the effect of series coupling, consider the mismatch configuration relative to the nominal configuration. The stalling flow coefficient of the steady state
speedline is decreased from the baseline value of 0.345 to the mismatch value of 0.308. The presence of the rear three stages has artificially stabilized the first rotor. This point is illustrated by Figs. 4 and 5 in which the first stage and overall steady pumping characteristics are shown. In the artificially stabilized region the first rotor characteristic is positively sloped (unstable) while the overall characteristic is negatively sloped (stable). In other words, locally the first stage is unstable, but the overall system is stable. These results are similar to observations made by Longley and Hynes (1989).

This example demonstrates how the pips can be used to examine the relative stability of either a specific region or the overall system. Because of their relatively small length scale, the pips give detailed information about the local stability of the first stage. Closer examination of the first rotor in the artificially stabilized operating region reveals the presence of the pre-stall pip. Specifically one pip is visible at the flow coefficient of the nominal stall point, and as the flow is decreased, more pips appear around the annulus. Twelve pips are present prior to stall. The pip’s presence confirms the relative level of instability of the first rotor over this flow range. Furthermore, at any fixed operating point in the artificially stabilized region, the pips neither grow nor decay; they maintain their steady propagation about the annulus. This behavior is in contrast to the pip’s quick growth rate in the nominal machine. The presence of the pips indicates that the first stage is relatively unstable in the artificially stabilized flow region. However, the fact that for a fixed operating point the pips do not grow into the larger length scale system instability of rotating stall suggests that the overall system is stable.

This example clearly demonstrates how other components in series can affect the system stability. A stability analysis performed on the first rotor alone would have inaccurately predicted the system to be unstable for flow coefficients between 0.308 and 0.345. One must also consider the environment in which the rotor operates, i.e. the downstream bladerows. It is evident that series effects should be taken into consideration if the stability is to be properly predicted.
Parallel Coupling

The parallel coupling problem may be demonstrated by considering the asymmetric or notch tip clearance configuration. The critical point in this problem is that the pip's dimension is small compared to 120°, i.e., the length scale of the problem is significantly larger than that of the pip phenomenon. Consequently, the annulus appears to consist of two different circumferential regions. As described by Mazzawy (1976), these two regions may be analyzed as two parallel compressors. Following this idea, consider two separate compressors both of axisymmetric tip clearance. The first compressor has a tip clearance (gap-to-span) equal to the baseline build clearance (1.3%) and the second compressor has a larger tip clearance equal to the notch clearance (3.1%).

Experiments with these two axisymmetric compressors reveal some key points. First, the stalling flow coefficient for the larger clearance machine is 0.352 which is greater than the nominal machine's stalling flow coefficient of 0.345. This change is evident from Fig. 6 which shows the first stage characteristics for the two axisymmetric builds. Also, it was demonstrated by Silkowski (1995) that over the outer 20% of span the larger tip clearance build has a reduction in axial flow compared to the baseline build. For these reasons the larger clearance sector is considered the less stable region. Finally, the speed of propagation of the pip is different in the two different axisymmetric tip clearance machines. The pip speed is 70% of the shaft speed in the nominal clearance machine, and 62% of the shaft speed in the larger clearance machine.

With these facts in mind the relative stability of the two sectors can be examined and the role of coupling between the two sectors can be highlighted. One way to assess the relative stability of the two sectors is to examine the flow coefficient, \( \phi \), and RMS pressure profiles around the annulus. Figures 7 and 8 show the lowest \( \phi \) and highest RMS pressure are at the exit of the large clearance notch. These facts also suggest that the notch is the less stable region. Similar behavior was seen in subsequent work by Graf et al. (1997) on a machine which exhibited "modal" stall when all four stages were configured with asymmetric tip clearance.

The relative stability of the two sectors can also be examined by studying the behavior of the pipes. The pipes give a good indication of local conditions due to their small size relative to the geometric variations. The observations of the pip's behavior are consistent with the previous comments regarding the relative stability of the two sectors and the parallel compressor ideology. Specifically, pips first form and grow in the relatively less stable region, the notch, and then are damped in the relatively more stable region of the
tighter clearance. This is in agreement with the flow coefficient and RMS pressure trends and can be clearly seen in the time traces from probes distributed about the annulus, see Fig. 9. Note that disturbances which have started in the notch tip clearance may not reach an observable amplitude until after they have grown during their propagation through the unstable region. Furthermore, these downstream probes see some circumferential offset due to the pip's circumferential propagation. In this figure there are ten time histories. This is because the histories from the closely spaced hot wires at 15° and 25°, designated by probe numbers 1a and 1b, have also been included.

Pips first appear in the unstable region of the larger tip clearance. However, in this region there is a high concentration of pips (higher level of unsteadiness) and it is difficult to identify any single pip. This point is made more clear by examining raw data traces, such as those in Fig. 13 which are from a similar experiment. Wires 1, 1a, 1b, and 2 in Fig. 13 are in the least stable region and are much noisier than the wires in the stable region. Due to this higher general level of unsteadiness (higher concentration of pips), individual pips do not stand out clearly in the unstable region. As shown in Fig. 9, normalization does not help isolate an individual pip since the entire signal, including the noise, is simply scaled. The high concentration of pips in the unstable region at the edge of the notch is indicated in Fig. 9 by the fact that in these normalized time traces, as compared to those in the stable flow region, the blade passing signal is not easily distinguishable. This is because the blade passing signal is masked by the stronger unsteady signal created by the high concentration of pips in this unstable region. Additionally, the higher unsteadiness caused by the presence of a large number of pips leads to a larger normalization factor, and therefore even when blade passing is visible on these wires, it appears as a much smaller signal. Therefore, individual pips are first clearly visible only after they have migrated from the unstable region to the stable region. Examples of this are given at points "A", "B", and "C" in Fig. 9, which show pips at the border between the two clearance regions, where the RMS is highest and the flow coefficient is the lowest. (Recall that the rotor rotation and pip propagation are in the direction of increasing probe numbers.) Each of these points show a clearly visible pip, after it has migrated into the stable region, away from the region dominated by high unsteadiness and multiple pips. The changing level of normalization (due to the changing level of unsteadiness) between the wires also explains why disturbances such as "B" appear to grow in the stabilizing region. This point is illustrated by noting that the blade passing signal also appears to grow from wires two through six. However, these isolated pips are quickly damped in the more stable tight clearance region. This damping is seen by following the guidelines which have been drawn with a slope representing the pip's tight clearance propagation speed of 70% rotor speed. None of these disturbances propagate through to the end of the stabilizing region, wire 7.
The relevance of the parallel compressor analogy, and the dominance of the local environment in determining the behavior of small scale phenomena is shown in Fig. 10. This figure shows the circumferential propagation speed of the pip at different locations around the annulus. These speeds were calculated by crosscorrelating adjacent pressure transducers and plotting the results at a circumferential location halfway between the probes. Note that in the notch the pip speed is similar to that of a pip in an axisymmetric machine of that clearance. The same can be said for the nominal clearance region.

This notch tip clearance configuration has demonstrated that parallel coupling can affect system performance and stability. Near stall, the nominal clearance region stabilized the large clearance notch region. Cumpsty (1989) made a similar observation for parallel coupling created by asymmetric casing treatment. Furthermore, the relevance of the parallel compressor approach for small length scale phenomena was demonstrated, i.e. the pip behaved in each region as if in an axisymmetric machine of the same character.

Combined Series/Parallel Coupling

A hybrid series/parallel coupling may be achieved by using a fixed inlet distortion screen configuration. Similar conclusions to those reached for the 120° notch clearance configuration can also be made for the fixed distortion screen configuration. Specifically, consider the circumferential profiles of flow coefficient, ϕ, and the RMS velocity at the rotor inlet as shown in Figs. 11 and 12. As in the notch tip clearance experiment the lowest flow and highest RMS both occur at the end of the distortion screen region. These observations are consistent with time histories at the rotor inlet (Fig. 13) which show that the pip disturbances are first visible behind the distortion screen and that individual pips are first identifiable at the end of the distortion screen region. These individual pips then decay through the clean flow region. This figure shows that the region behind the distortion screen is relatively unstable as indicated by the noisy signals due to the presence and formation of pips. Pips are created and grow throughout this distortion screen region. (As discussed in the notch tip clearance section, individual pips are not clearly visible until one migrates into the cleaner, stable region.) However, damping in the more stable, clean flow sector prevents the pips from causing a larger global system instability (rotating stall). This damping is evident by the lower noise levels on the wires of the clean flow region. Only occasionally does a pip migrate any substantial distance into the clean flow sector before eventually decaying away. Hence, the clean flow sector has artificially stabilized the distortion screen sector. In fact, even after rotating stall has formed the relative stability of the two sectors is confirmed by the fact that the stall cell attenuates in the clean flow region, see Fig. 14.
Once again, the small scale pips have been used to study the relative stability of different regions of the machine. The presence or absence of the pips, as well as their growth rates were used to identify regions of relative stability and instability. Similar to the previous examples, a relatively stable region stabilized a relatively unstable region.

CONCLUSIONS
Comparing and contrasting these different variations of a single rig has provided useful information and data to assist modeling development and CFD analysis. Specifically, experimental investigation of a baseline compression system configuration and three variations has precipitated the following observations:

- "Pip" type rotating stall inception is initiated in the tip region (3-D).
- The initial pip's are only a few blade pitches in circumferential extent and travel at roughly 70% of rotor speed before decelerating to roughly 45% of rotor speed.
- Myopic examination of a compression system component or region may lead to incorrect results.
- Coupling between different parts of the compression system may affect overall stability and performance.
- Have given examples of coupling in series, parallel, and combined series/parallel. Three examples where an un-stable element or portion of the compression system was prevented from leading to system instability by coupling to other more stable portions of the compression system.
- In parallel coupling, the lowest \( \phi \) and highest unsteadiness occur at the end of the least stable region.
- Component coupling's role in stability is moderated by length scale considerations. In all the examples, formation and behavior of the small length scale pips was governed by the local conditions. However, the ability or inability of the pips to grow into the larger length scale phenomena of rotating stall and hence the overall system stability was controlled by global coupling.
- The small length scale, pre-stall disturbances (pips) proved to be a useful tool for examining the relative stability of different regions.

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