Origins and Structure of Spike-Type Rotating Stall

The MIT Faculty has made this article openly available. Please share how this access benefits you. Your story matters.

<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>As Published</td>
<td><a href="http://dx.doi.org/10.1115/1.4028494">http://dx.doi.org/10.1115/1.4028494</a></td>
</tr>
<tr>
<td>Publisher</td>
<td>ASME International</td>
</tr>
<tr>
<td>Version</td>
<td>Final published version</td>
</tr>
<tr>
<td>Accessed</td>
<td>Wed Oct 31 04:59:30 EDT 2018</td>
</tr>
<tr>
<td>Citable Link</td>
<td><a href="http://hdl.handle.net/1721.1/114778">http://hdl.handle.net/1721.1/114778</a></td>
</tr>
<tr>
<td>Terms of Use</td>
<td>Article is made available in accordance with the publisher's policy and may be subject to US copyright law. Please refer to the publisher's site for terms of use.</td>
</tr>
<tr>
<td>Detailed Terms</td>
<td></td>
</tr>
</tbody>
</table>
Introduction

It has been known for more than a decade that there are two distinct routes to compressor rotating stall with different onset criteria for each [1]. One is through the growth of small amplitude (compared to the mean velocity) disturbances with length scale of the compressor circumference. The initial stages of this process are in accord with the linear-to-nonlinear wave evolution described by established theories of compressor instability [2]. A second route, however, is through transient disturbances with much shorter length scale (several blade pitches), which, as detected by probes upstream of a blade row, are of large amplitude compared to the mean velocity and are not described by existing analyses. The disturbances in this second route have been called “spikes,” in reference to the sharp waveform they exhibit compared to the first type of disturbance. The phenomena responsible for the formation of spikes are not well understood.

In this paper, we define the fluid dynamic processes responsible for the occurrence and formation of spikes as a route to compressor rotating stall. It will be shown that the spike is linked to separation from the leading edge in the tip region. Further, spike stall has been found in axial compressors with tip clearance, axial compressors with tip shrouds (hence, no tip clearance or relatively rotating endwall), and in centrifugal compressor vaned diffusers. The implication we draw from these experimental results, and from the simulations, is that the proposed mechanism appears generic and applicable to a broad range of turbomachines.

We emphasize the generic aspects in what follows, although there is an important caveat in that the circumstances that promote the leading edge separation in a specific situation do depend on the details of the turbomachinery. In this context, a rough analogy can be made with the first route to instability. It is well-known that the conditions under which long length scale, small amplitude, disturbances grow into rotating stall are associated with operation on a positively sloped compressor pressure rise characteristic; that is the generic finding concerning long length scale disturbances. There are, however, multiple fluid dynamic processes by which this criterion can be achieved: increased blade loss, increased tip clearance flow blockage, increased deviation, or a combination of two or more of these, which occurs depending on design specifics, and any or all can lead to a situation in which the pressure rise characteristic peaks. We will show that an analogous set of circumstances is present in the description of spike stall, in that there are many possible causes for the increase in incidence required to trigger a leading edge separation.

The organization and scope of the paper is as follows. We first show experimental measurements from axial compressors with tip clearance and with a shrouded tip, and from a centrifugal compressor with vaned diffuser, as examples of spike stall. Following that we introduce the proposed vortex behavior and the structure of the spike to address, in qualitative terms, the question of “what is a spike?” A series of unsteady simulations of the NASA E3 Rotor B are interrogated to provide insight into the quantitative nature of spike stall, with the understanding gained then applied to interpret further computations and detailed experimental information from an additional compressor.

Experimental Information on Spike Stall

Spike-type rotating stall inception is known to occur in axial flow compressors near the tip of the rotor blade leading edge. The stall onset is apparent in time-resolved casing static pressure measurements from probes at different circumferential locations.
upstream of a rotor. Figure 1 is an example, which shows data from five pressure transducers in an MHI single-stage axial compressor with hub-to-tip ratio of 0.7 and rotor blade tip clearance of 1% span. The spike disrupts the blade passing waveforms by a sudden rise in pressure, followed by a rapid pressure drop. Similar traces have been reported in Refs. [3–5] and elsewhere. The pattern grows in amplitude and length scale as it travels around the circumference at approximately 80% of rotor speed before evolving into rotating stall.

Although it has been conjectured that tip clearance flow is necessary for spike formation, there are at least two sets of experiments that show this not to be the case, the first on a tip shrouded axial compressor and the second on a centrifugal compressor with vaned diffuser. In the first of these, a single-stage axial compressor at MIT was run with a sheet metal shroud to prevent tip leakage [6] and data were taken on the type of stall onset. Figure 2 shows the resulting time-resolved casing static pressures during throttle ramps into stall. Stall inception in the shrouded compressor is seen to consist of spike-like disturbances that rotate around the annulus at about 70% of rotor speed and evolve into a full-span stall cell.

In the second set of experiments, measurements carried out in a centrifugal compressor vaned diffuser suggest that spike-type stall precursors occur near the leading edge of shrouded diffuser vanes [7]. In this case, the precursors are larger in spatial extent and slower in rotation rate than in axial machines, consistent with the larger blade pitch. Figure 3 shows the time-resolved shroud static pressures, at eight locations around the circumference, at a radial station between the impeller and the diffuser. The disturbance propagates at about 20% of rotor speed. The sudden rise in pressure followed by the rapid pressure drop can again be seen. Unsteady calculations reveal that flow separation at the diffuser leading edge, caused by high incidence near the shroud, leads to reversed radial flow, allowing vorticity shed from the leading edge to move inward and recirculate around the circumference. As blockage accumulates, the spikes grow, leading to flow breakdown into rotating stall [8].

In support of the comments in the Introduction, therefore, the inference we draw is that spike-like prestall patterns occur in centrifugal as well as axial turbomachinery and that blade tip leakage flow is not a necessary condition for their formation.

A Conceptual Picture of Spike Flow Structure

While the above two sections give information on the flows in which spikes form, and while there are descriptions of the conditions that accompany this formation [1,4,9], our perspective is that there exists no description of the fluid dynamic events which are causally linked with the creation of the spike. Providing this link, based on interrogations of numerical simulations and experimental measurements, is the main contribution of this paper.

Most of the calculation results to be presented are instantaneous snapshots of pressure and radial vorticity near the compressor casing. Anticipating those results, we state that they allow a conceptual picture to be built up of the key features of the vortical spike structure as depicted in Fig. 4: (i) High incidence results in a separation from the leading edge in the outer part of the blade. (ii) The vorticity shed by this separation forms a vortex tube that spans from blade to casing as was suggested by Inoue et al. [10]. (iii) The upper “end” of the vortex is on the casing and the lower end is on the blade. The convection of the vortex, as seen in the rotor

---

**Fig. 1** Spike stall inception in MHI single-stage axial compressor with rotor tip clearance of 1% span. (Data courtesy of Takasago R&D Center, MHI.)

**Fig. 2** Spike stall inception in the MIT single-stage axial compressor with tip shrouded rotor [6]

**Fig. 3** Spike stall inception in the vaned diffuser of a centrifugal compressor [7]

---

2This speed is measured in the same (absolute) frame of reference as the stalling blade row. For stall inception occurring in a rotor, a spike travelling at 80% of wheel speed in the absolute frame has a speed of 20% in the relative frame.

3Vortex lines cannot end in a fluid, and what we have drawn is a section of a vortex tube. The vortex lines within the tube continue in thin layers on the casing and on the rotor (at these surfaces they must be tangential to the surface) rather than in the form of a discrete vortex. For simplicity, we have not drawn the vortex lines outside of the section of the tube that is shown.
frame of reference, is indicated by the arrows. (iv) A new separation then forms on the next blade and the spike propagates. Simulations both with and without clearance show this structure, and we thus infer that the formation of, and vorticity source for, the vortex shed from the leading edge is not associated with the tip clearance vortex.

A pressure transducer just upstream of the leading edge at the casing will respond to the proposed spike structure in the manner shown in Fig. 5. The stationary sensor first encounters the potential field of the blockage caused by the leading edge separation (a pressure rise) then a sharp pressure drop as the vortex tube crosses the transducer. The resultant up–down waveform is a characteristic of a spike.

Figure 6 is a sketch of the time evolution of the “vortex skeleton” associated with this behavior. It shows vortex filaments near the blade leading edge region, looking upstream, roughly along
the blade stagger angle. The blades are drawn as heavy vertical lines. The sketch on the left side of Fig. 6 shows the passage at three time instants. At time $t_0$, there is no separation, and the vortex filaments are close to the suction surface. At time $t_1$, the upper part of the filament (and the surrounding vortex tube) has detached from the suction side, and at time $t_2$ the upper part has moved further away from the suction surface. In terms of image vorticity, as in the right side sketch of Fig. 6, the configuration at time $t_2$ appears as a vortex ring, with sense of vorticity implying upstream propagation, given an appropriate combination of low enough onset velocity and strong enough circulation.

The conceptual picture of the spike that we put forward appears to apply to the different geometries examined; axial compressors with tip clearance, tip shrouded axial compressors, and centrifugal compressors. It is also consistent with the simulations, with the experimental findings both overall (the view of spikes as an incidence-related phenomenon localized to the tip region), and detailed (the characteristic wave form of pressure transducers upstream of the leading edge, and the casing static pressure field measured by arrays of transducers).

Summary of the Computational Approach

Information on the computational method is provided in the Appendix and only a brief summary is given here. The computations were performed using Turbostream [11], a structured multiblock URANS code based on the algorithms developed by Denton [12]. A single equation Spalart–Allmaras turbulence model was used. All computations were performed on graphics processing units (GPU), allowing an order of magnitude reduction in runtime compared to running the same calculations on conventional processors.

E$^3$ Rotor B Simulations

Introduction. The NASA E$^3$ Rotor B is used to investigate the fluid structures at spike onset. This rotor is chosen because it has been the subject of a previous study of spike-type stall inception [9], the information on the geometry is public, and it is representative of a range of machines of technological interest. The rotor is characterized by the parameters listed in Table 1 and detailed information is provided in Ref. [13].

The computations are presented in order of increasing complexity: a linear cascade of translating E$^3$ Rotor B tip profiles; E$^3$ Rotor B without tip clearance; and E$^3$ Rotor B with tip clearance. This progression has two advantages. First, it allows the details of the spike to be diagnosed first in situations with fewer interacting phenomena than when tip clearance is present and, second, it facilitates the identification of mechanisms common to all cases.

Figure 7 shows computed total-to-static characteristics for the three E$^3$ Rotor B geometries. The circles denote unsteady single passage computations, and the squares denote computations carried out using a domain of a quarter of the circumference (for blade periodicity, 93 deg). Only stable operating points are indicated. Two observations can be made. First, in each case, the characteristic has a negative slope at the stall point and we therefore expect the simulations to take the spike route to rotating stall.

Two-Dimensional Tip Section Computations. Based on the evidence that tip clearance is not required for spike formation, the first set of simulations we explore are for a two-dimensional linear cascade of E$^3$ tip section profiles. Following the procedure described in the Appendix, an unsteady calculation of a single passage domain was used to identify the stability limit. Then, a “full wheel” (periodic over 54 passages) domain was used, with the flow coefficient gradually reduced until stall onset. To initiate stall at a known location for detailed analysis, four adjacent blades were restaggered (permanently) by 1 deg to increase their incidence.

Figure 8 shows a set of computed nondimensional static pressure traces at six equispaced locations across the 54 passages, 10% of axial chord upstream of the leading edge. A perturbation in the upstream potential field of the blades is first detected, at the restaggered blades, at time $t_1$. Within half a rotor revolution, this disturbance has grown and the pressure trace now shows the characteristic up–down waveform of a spike.
The flow field accompanying the initial disturbance (time $t_1$, location "X") and spike (time $t_2$, location "Y") are now examined to determine the origins, structure, and propagation mechanism of the spike. Figure 9 is representative of a type of plot that will be used frequently during the paper. The solid and dashed line traces at the top of the figure show the static pressure at 10% of axial chord upstream of the leading edge (at the location indicated by the dotted line). The solid line is the instantaneous trace; the dashed line is the time-averaged trace (averaged in the relative frame over one rotor revolution immediately prior to stall inception). A line representing $0.5pU^2$ is drawn on the right of the plot to provide a scale. Below these traces are contour plots of non-dimensional static pressure coefficient, $\Delta p$, and radial component of vorticity, $\omega_r$. Each blade is assigned a number shown at the bottom of the figure and time is indicated with reference to the accompanying pressure trace plot (Fig. 8) with $\tau$ indicating the blade passing period.

Figure 9 shows the flow field associated with the initial disturbance at two time instants. The pressure trace deviates from the time-averaged value due to a perturbation in leading edge loading. In Fig. 9(a), the suction-surface boundary layers of blades 1–3 are separated at approximately midchord; the consequent blockage means that the pressure upstream of the blades rises. The separation on blade 3 causes an increase in incidence on to blade 4; the resultant increase in leading edge loading causes a local drop in the upstream pressure trace. The suction-surface boundary layer of blade 4 cannot sustain the required deceleration and separates, increasing the incidence on to blade 5, Fig. 9(b). In this way, the disturbance has propagated, by one blade pitch, in the way proposed by Emmons et al. [14].

The suction surface separations continue to propagate from blade to blade, moving away from the restaggered blades where they first occurred. The separations move further forward with each blade because of the following positive feedback mechanism: blockage from the separation on one blade increases the incidence on to the next, causing the next blade to separate closer to the leading edge and hence create a larger blockage, and so on. At the time shown in Fig. 10(a), the separation occurs close to the leading edge of blade 7 and at the leading edge of blade 8. The passage between blades 7 and 8 is almost completely blocked (causing the pressure rise on the upstream trace). The vorticity that is shed by the leading edge of blade 8 rolls up into a vortex (labeled “A”) that convects toward the leading edge of blade 9 in Fig. 10(b)—the low pressure associated with this vortex is responsible for the sharp pressure drop seen on the upstream trace. Once it arrives at blade 9, vortex “A” triggers a new leading edge separation, causing a new shedding of vorticity. The propagation of the spike therefore includes the convection of shed vorticity from one blade to the next. In the relative frame, this means that the pressure drop (the low pressure zone associated with the shed vortex) leads the pressure rise (high pressure caused by the blocked passages). In the absolute frame, stationary pressure probes measure a positive pulse followed by a negative pulse, see Fig. 5.

The two-dimensional calculations thus show that leading edge separation is responsible for the spike. The spike propagates by the convection of the vortex that is shed from the leading edge. This will be shown to be the case in all the configurations presented in the paper. For this two-dimensional geometry, the high
The incidence responsible for the leading edge separation was caused by blockage from separated blade surface boundary layers.

**Computations of 3D Blade With Zero Tip Clearance.** We extend the preceding two-dimensional description to three dimensions and examine the full E3 Rotor B blade. The first set of three-dimensional calculations to be shown is with zero tip clearance. While the setup is artificial because the casing is stationary (rather than rotating as in a shrouded rotor), it is a limiting case for the unshrouded rotor, providing a step upward in complexity toward the full situation. A similar procedure to that followed for the two-dimensional computations was adopted: single passage unsteady simulations (1.4 × 10^6 nodes, 12 operating points) to obtain an approximate stability limit; quarter annulus (14 blades, 93 deg) simulations (19 × 10^6 nodes, 2 operating points) with four restaggered blades to obtain the operating point at spike onset; full annulus (74 × 10^6 nodes, 2 operating points) simulations to be certain that the spike development and propagation is unaffected by the imposition of a 4-per-rev repeating sector of 1 deg increased incidence.

Figure 11 shows simulated time traces from six static pressure probes arranged around the casing circumference at 10% of axial chord upstream of the rotor leading edge. The similarities between this plot and that of the two-dimensional linear cascade of tip section profiles (Fig. 8) can be seen. The initial perturbation to the regular upstream potential field of the blades is seen at \( t_1 \). Within half of a rotor revolution, this disturbance has formed a spike (time \( t_2 \), location \( "Y" \)).

In the same manner as the two-dimensional case, the perturbation to the blade loadings in zone \( "X" \) of Fig. 11 is caused by separation from the blade suction surface that occurs first on the restaggered blades, Fig. 12. In common with all similar plots from three-dimensional simulations shown in this paper, the flow field and pressure traces in Fig. 12 are from a cut at 95% span so the separation shown is actually a slice through a three-dimensional corner separation. Blockage from the corner separations on blades 1 and 2 causes the pressure rise in the upstream trace. The high incidence on blade 3 results in high leading edge loading and a small pressure drop in the trace.

The first separation from the leading edge of a blade is seen in Fig. 13(a) and occurs at blade 6. The shed vorticity rolls up into vortex \( "B" \) that convects toward, and in front of, the leading edge of blade 7. Here, as was the case in the two-dimensional simulations, the vortex triggers a new leading edge separation (from blade 7) and vorticity will be shed.

The key difference, as regards the spike structure, between the two-dimensional cascade and the full blade row is the spanwise extent of the leading edge separation. To illustrate this, a sequence of snapshots of blade 6 showing an isosurface of the \( \lambda_2 \) vortex criterion [15] is presented in Fig. 14. The \( \lambda_2 \) isosurface can be regarded as a marker for vorticity associated with discrete swirling flow structures (i.e., vortices) rather than vorticity in a sheet or shear layer (a boundary layer, for example). Thus, even though
vortex lines cannot end in a fluid, the \( \lambda_2 \) surfaces can do so. The final snapshot in the series of three, Fig. 14(c), is at the same time instant as Fig. 13(a); two preceding time instances are also shown so that the development of the structure can be observed.

Figure 14(a) shows that the spanwise nonuniform incidence, greatest at the tip, has caused a leading edge separation (circled in the figure) to form over the outer 25% of span of blade 6. In Fig. 14(b), the vorticity shed from the blade by this separation has one end at the casing—this convects to the leading edge of the next blade in the same way as the vortex in the two-dimensional simulations—and one on the suction surface which travels downstream. For this three-dimensional, zero clearance, geometry, the high incidence responsible for the leading edge separation was caused by blockage from a corner separation.

Computations of 3D Blade With Tip Clearance. The final set of computations of the E\(^3\) Rotor B was performed with a tip clearance of 1.8% of chord. The computational procedure and run times for this set of simulations were similar to those described above for the zero clearance case.

Approaching Stall. As the E\(^3\) rotor is throttled toward stall, the interface line that demarcates the oncoming flow from the over-tip leakage flow rotates toward the circumferential direction until, just before stall, it is almost coincident with the leading edge plane. This is illustrated by the time-averaged entropy contours, at 95% span, shown in Fig. 15. The operating point shown is stable.

It is also found that the tip clearance flow is highly unsteady, as has been observed by others [20,21]. Figure 15 shows instantaneous contours of radial vorticity at the same operating point. The vortex filaments of the tip leakage vortex, which have no radial component at the exit of the tip gap, deform to produce a series of islands of positive and negative radial vorticity which convects across the passage. Low pressure regions associated with these structures were measured by Young et al. [22] on the casing of the large tip gap portion of their axial compressor with eccentric clearance.

At Stall. Figure 16 shows simulated casing static pressure traces as the E\(^3\) rotor with tip clearance is brought into stall. When compared with the pressure traces from the two-dimensional and
leading edge separation results. However, because the next, the incidence is very high (the flow is essentially tangential) clearance flow from one blade reaches the leading edge of the incidence necessary for leading edge separation. When the tip surfaces. Instead, it is the tip clearance flow that provides the clearance cases, no separations are visible on the rear suction (zone “0” in Fig. 16). In contrast to the two-dimensional and zero clearance cases, the structure and propagation of the spike, Fig. 17(b), are the same. Leading edge separation causes the shedding of vorticity which convects to, and in front of, the adjacent blade. Prior to stall, the unsteadiness of this tip leakage flow causes multiple transient leading edge separations that do not survive (i.e., do not convect in front of the adjacent blades) to form a spike.

Summary of E3 Calculations. All three cases of the E3 geometry exhibit the same mechanism of spike formation and propagation. In each case, flow separation from the leading edge results in shed radial vorticity which convects to the leading edge of the adjacent blade, triggering a new separation. The principal difference between the simulations is the cause of the incidence that precipitates the leading edge separation. For the two-dimensional and zero clearance cases, the incidence is caused by the blockage from rear suction surface or corner separations. For the case with tip clearance, the incidence is caused by the impingement of the tip leakage flow from the adjacent blade. It is plausible that a rotor with a weaker tip clearance flow (e.g., due to a smaller tip gap) may exhibit corner separations and behave in a similar manner to the zero clearance case. Whatever the particular circumstances that promote high incidence in a given compressor, separation from the leading edge is seen to be the common factor in the structure and propagation of the spike.

Cambridge Low Speed Compressor

To provide experimental verification of the structures and mechanisms discerned from the E3 computations, a series of tests were performed on a low speed single-stage compressor at the University of Cambridge. Details of the machine are given by Young et al. [22] and are summarized in Table 2.

The compressor was run with a tip clearance of 1.4% chord. An axial array of five pressure transducers, covering a region from \(-0.4 \leq x/L_c \leq +0.4\) was installed at one circumferential location. Eight additional sensors were placed around the annulus to track stall development. To compare with the experimental data, computations were run on the stage (132 \(\times\) 10\(^6\) nodes for the full annulus domain) using the procedure described in the Appendix.

Table 2  Design parameters for the Cambridge compressor

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design flow coefficient (V_r/U_i)</td>
<td>0.44</td>
</tr>
<tr>
<td>Design stage pressure rise coefficient (\Delta p_0/(0.5pU_i^2))</td>
<td>0.44</td>
</tr>
<tr>
<td>Hub to tip radius ratio (r_h/r_t)</td>
<td>0.75</td>
</tr>
<tr>
<td>Tip diameter</td>
<td>488 mm</td>
</tr>
<tr>
<td>Number of rotor blades</td>
<td>58</td>
</tr>
<tr>
<td>Rotational speed</td>
<td>2980 rpm</td>
</tr>
</tbody>
</table>
Figure 18 shows measured and calculated total-to-static pressure rise characteristics. As for the $E^3$ characteristics, Fig. 7, the machine stalls on a negative slope and the quarter-circumference (87 deg in this case) sector computations stall at a higher flow coefficient than those for the single passage. The overall shape of the computed characteristic agrees well with the experiment with the flow coefficient at stall from the simulations 0.01 higher than the measured value.

Traces recorded by eight equally spaced casing pressure transducers at the rotor leading edge plane during a stall inception event are shown in Fig. 19(a); equivalent results from a full annulus computation are presented in Fig. 19(b). The close qualitative agreement between the measurement and simulation is encouraging with the spike having a propagation speed, growth rate, and structure (as indicated by the waveform within the stall cell) that are all reproduced well in the computation.

Figure 20 depicts the early stages of spike development ($t = 0.39$ revs in the computation). Note that the experiments show data from ten blades as they move past the stationary pressure transducers, whereas the CFD is an instantaneous view of ten blades. The computations show that the same mechanism observed in the $E^3$ simulations is responsible for the formation of the spike: leading edge separation initiated by high incidence caused by the tip leakage flow. The intermittent leading edge separation seen in the $E^3$ simulations with tip clearance is present in the Cambridge compressor, but the small spatial extent of the vortices (labeled "D") make them hard to discern in the measurements. The reduced blade loading caused by the flow separating from the leading edges of blades labeled "E" is apparent.

At the later time depicted in Fig. 21 ($t = 0.55$ revs in the computation), the experiments and computations show a spike. A large low pressure region associated with a leading edge separation is seen at "F." The location of this region, including the low pressure zone further aft in the preceding blade passage, is well-reproduced by the computations. The measurements and simulations also show a series of vortices, labeled "G," in the preceding passages.
Specific Findings

(1) Spike-type stall inception involves localized disturbances, of blade passage scale, which propagate from blade to blade around the annulus. The origins of a spike involve radial vorticity, which is shed when flow separates from the leading edge of a blade and then rolls up into a discrete vortex of subpassage proportions. The vortex convects circumferentially toward, and then in front of, the leading edge of the next blade, which experiences increased incidence and then separation and the formation of a new vortex. The pattern of separation, vortex formation, translation, new blade flow separation and renewed vortex formation, explains the structure and propagation mechanism of the spike.

(2) During stall onset, flow separation causes blockage and a rise in static pressure upstream of the blade row with circumferential extent of a few blade pitches. This localized increase in pressure is observed as a sharp upward peak in measurement data and is responsible for the classification of this type of disturbance as a “spike.”

(3) The shed vortex that accompanies the leading edge separation gives rise to a region of low static pressure on the compressor casing. The resulting pattern of a region of high pressure adjacent to a region of low pressure gives rise to the typical up–down signature of a maturing spike seen in pressure signal recorded upstream of the rotor.

(4) The experimental and computational results agree and they suggest that spike formation is not dependent on tip leakage flow. More precisely, the vortex described above is not associated with the tip leakage flow in that the same general features are observed in blade rows with and without tip clearance. In cases where tip clearance is present, as in a conventional axial compressor, the leakage flow contributes to high incidence at the rotor tips, but the leakage flow itself is not a requirement for flow separation and spike formation.

(5) The fluid dynamic processes that give rise to the incidence that creates the initial leading edge separation depend on the details of the blades being used. For a two-dimensional cascade, the separation process begins at the trailing edge of the blade and works its way toward the leading edge. In a three-dimensional case without tip clearance, corner separation can play a part in triggering separation at the leading edge. In a conventional blade row with tip clearance, leakage flow can sometimes exacerbate leading edge incidence. In all cases, however, regardless of the precise cause of increased incidence, spike development is due to leading edge flow separation and vortex formation.

Summary and Conclusions

Overall Findings

(1) Numerical simulations and experimental measurements have been used to explain the origins, structure, and propagation mechanism of spike-type disturbances in rotating stall inception. The general features of spike formation are found to be similar in two-dimensional cascades, axial compressor stages with rotor tip clearance and with rotor tip shrouds, and vaned diffusers in centrifugal compressors. In all these geometries, spikes originate from flow separation near the blade leading edge on the suction side, which results in vortex formation and propagation around the annulus.

(2) The description of spike behavior given here aligns with early ideas put forward by Emmons et al. [14], adding depth and quantitative information to his incidence based mechanism, and by Inoue et al. [10], who initially suggested the involvement of vortex structures in cell formation. The results in the paper show that the description proposed is consistent with both the numerical simulations and the experiments for compressors with a range of differing blade parameters.

Acknowledgment

The authors gratefully acknowledge the contributions made by several individuals during many discussions on the topics described in this paper, including those provided by C. Freeman and S. A. Weichert of the Whittle Laboratory, N. A. Cumpsty of Imperial College, and J. J. Adamczyk. The authors would also like to thank S. Aoki, S. Uchida and E. Ito of MHI for their support during the project; J. J. Bolger of Rolls-Royce for supplying the geometry of the Cambridge Low Speed Compressor and for supporting the associated experimental program; and M. Brand and A. Kottapalli for taking the data on the MIT shrouded compressor.

Nomenclature

\[
\begin{align*}
  c_s & = \text{axial chord} \\
  p & = \text{pressure} \\
  p_{01} & = \text{rotor inlet stagnation pressure} \\
  r_t & = \text{tip radius}
\end{align*}
\]
Appendix: Details of the Computational Approach

All the computations presented in this paper were performed using Turbostream [11]. The code is a structured multiblock RANS solver that was developed using the method employed in the Denton code, TBLOCK [23]. The approach is finite-volume time-marching, second order in space, with three levels of multigrid and a single block explicit time integration scheme. All simulations presented here used the Spalart–Allmaras turbulence model with adaptive wall functions and had fully turbulent boundary layers. All the runs were second order accurate in time using Jameson’s dual timestepping technique, with 72 physical time steps per blade passing period.

The principal feature of Turbostream is its speed. As well as running on conventional central processing units (CPU), the code is also optimized to run on clusters of GPUs. Turbostream runs 10–20 times faster on one GPU as compared to all cores of a contemporaneous CPU. This allowed an operating point of the quarter annulus (≈20×10^2 node for the E^7 domain) calculations to be completed in 1 day on 32 GPUs.

The grids were generated using a Mitsubishi Heavy Industries in-house code that produced elliptically smoothed blade row meshes with H–O–H topology. Tip gaps were solved with two additional blocks (O–H topology) with 11 points in the gap. All blades were meshed with 293 × 37 × 74 points in the O-mesh, with approximately 30 points around the leading edge, and typical y⁺ values were below 5 on the blade surfaces. The computational domain extended 1.5 tip radii upstream and downstream of the blade rows. This gave an axial distance of approximately one quarter of a circumference between the compressor and the planes at which inlet or outlet boundary conditions were imposed. To reduce the mesh count required, the pitchwise node count per passage was reduced from 57 at the blade row, to 15 at the domain boundaries in three stages.

At the inlet of the domain, the total pressure and total temperature were specified as uniform. However, a portion of the inlet duct was set to have zero skin friction in order to control the inlet endwall boundary layer thickness. The boundary layer thickness was matched to measurement data for the Cambridge compressor. At the exit of the domain, a convergent nozzle was employed to provide the compressor with a back-pressure that changes with mass flow rather than a fixed pressure boundary condition [24]. The nozzle was not choked, so that altering the pressure downstream of the nozzle (specified to be uniform) caused the compressor operating point to change and the pressure upstream of the nozzle to vary according to the matching of the compressor characteristic and the characteristic of the nozzle.

Each operating point was run for five flow-through times (from domain inlet to exit); this was of the order of 4 rotor revolutions. Over the first rotor revolution, the back pressure at the exit of the nozzle was increased linearly with time and the flow coefficient typically reduced by 0.01. This procedure was done first with a single passage domain to obtain the approximate stall point, then with a quarter annulus domain, and finally with a full annulus.

Postprocessing is a formidable task with unsteady datasets of order 10^9 nodes. The approach taken was similar to that of the experimentalist. Several measurement planes were computed 2 times per rotor passing period, e.g., a selection of blade-to-blade planes, rotor blade surface, and rotor exit planes. From these, a particular range of times was identified for deeper analysis and the operating point was then rerun to obtain this data.

References


Journal of Turbomachinery
MAY 2015, Vol. 137 / 051007-11