CHARACTERIZATION OF TIP LEAKAGE FLOW TRAJECTORIES IN A MULTISTAGE COMPRESSOR USING TIME-RESOLVED OVER-ROTOR STATIC PRESSURES
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Abstract
Fast-response pressure measurements collected in a three-stage axial compressor highlight the development of the rotor tip leakage flow for three tip clearance heights. Data collected from an array of high-frequency-response pressure transducers measure time-resolved static pressure over the rotors. These data are presented at several compressor loading conditions for three rotor tip clearances to investigate changes of the tip leakage flow. Circumferential traverses of the stationary vane rows with respect to the fixed sensor positions show a modulation of the leakage flow trajectory as a result of interactions with upstream stator wakes. Multistage effects are also revealed by comparing measurements from each of the three rotors, and rotor-rotor interactions are observed in blade-to-blade analyses of the tip leakage flows. In some cases, variations of the leakage flow trajectory angle due to these blade row interactions are greater than the differences due to a doubling of rotor tip clearance height, an important finding not previously reported in the literature. Clearance derivatives for leakage flow angle are calculated and compared with trajectory angle measurements from a similar study to develop a reliable method for evaluating tip leakage flow trajectories at several loading conditions, which is independent of compressor geometry.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>BPP</td>
<td>Blade passing period</td>
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<tr>
<td>$c_x$</td>
<td>Axial chord</td>
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<tr>
<td>$\delta_\xi$</td>
<td>Clearance derivative for leakage flow angle</td>
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<tr>
<td>EA</td>
<td>Ensemble average</td>
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<tr>
<td>$H$</td>
<td>Annulus height</td>
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<tr>
<td>$\bar{m}_c$</td>
<td>Corrected mass flow rate</td>
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<tr>
<td>$P$</td>
<td>Pressure</td>
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<tr>
<td>RMS</td>
<td>Root mean square</td>
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<tr>
<td>$s$</td>
<td>Blade pitch</td>
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<tr>
<td>$t$</td>
<td>Time</td>
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<tr>
<td>TPR</td>
<td>Total pressure ratio</td>
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<tr>
<td>TLF</td>
<td>Tip leakage flow</td>
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<td>$\tau$</td>
<td>Tip clearance height</td>
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<tr>
<td>$\xi$</td>
<td>Leakage flow trajectory angle</td>
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Subscripts

- $o$-in,AA Area-averaged inlet stagnation pressure

Accents

$\langle \rangle$ Ensemble average

Introduction
In axial compressors, the tip leakage flow (TLF) develops as a result of the pressure difference between the pressure surface and suction surface of the rotor blade in the tip region. Past research has shown that increased rotor tip clearances leads to a growth of the blockage associated with the recirculating tip leakage flow, and this leakage flow is a significant contributor to overall compressor loss.\(^1\) As a result, the overall pressure rise capability of a compressor generally decreases with increasing rotor tip clearance.

Although the growth and development of the tip leakage flow is important for all stages of the compressor, its effect can be most significant in the rear stages of high-pressure compressors. In these rear stages, the rotor tip clearances are often large relative to the overall blade height, and the leakage flow affects a more significant fraction of the overall flow area. Next-generation engine designs will feature small flow areas and thus shorter blading in the rear stages as a result of increased bypass ratio and overall pressure ratio. In these cases leading to large relative tip clearance heights, an opportunity exists to build upon a limited understanding of the tip leakage flow and facilitate the development of loss-reduction design mechanisms.

For this reason, the detailed flow features associated with the rotor tip leakage flow are important for the advancement of computational modeling tools. However, the challenges associated with adequately modeling tip leakage flows in turbomachinery are substantial, as discussed by Denton.\(^2\) Therefore, detailed measurements of the tip leakage flow trajectory provide valuable information which will help develop and validate reliable modeling techniques.

Over the years, a variety of techniques have been incorporated to identify rotor tip leakage flow...
phenomena. One of these techniques utilizes high-frequency-response pressure transducers in a flush-mounted configuration in the casing over the rotor tips to monitor time-resolved casing static pressures.\textsuperscript{3,6} Using this method, the low static pressure region associated with the tip leakage flow can be used to identify the leakage flow trajectory through the blade passage.

Previous examples of this technique in single-stage or isolated rotor machines\textsuperscript{4,5} have provided valuable information about the development of the tip leakage flow, and data collected in a low-speed repeating stage machine\textsuperscript{3} have yielded valuable comparisons at two clearance heights. However, these studies lack important multistage effects.

Measurements collected in a two-stage high-speed compressor by Ernst et al.\textsuperscript{5} have provided valuable insight into the stator-rotor interactions and rotor-rotor interactions associated with the tip leakage flow. However, the study was only carried out for one tip clearance. Moyle et al.\textsuperscript{7} used casing static pressure measurements with two tip clearances in a two-stage compressor, but the clearances evaluated were both less than one percent span. Nonetheless, the observations documented by Moyle et al. showed an important dependence of the measured wall static pressure traces on the relative position of the adjacent stator vanes.

In support of research investigating tip leakage flows, other authors have also identified an interaction between the rotor tip leakage flow and wakes from upstream stator vanes.\textsuperscript{6,11} As the rotor passes through the velocity deficit region associated with the stator wake, an interaction occurs between the wake and the formation of the rotor tip leakage flow. This periodic interference pattern with the upstream stator wake introduces corresponding periodic fluctuations of the tip leakage flow.

In the present study, three rotor tip clearance heights are investigated using a series of high-frequency-response static pressure sensors mounted in a removable block over the rotors for several loading conditions on the 100% corrected speedline. These measurements provide valuable insight into the development of the tip leakage flow through quantification of the leakage flow trajectory angle and tip leakage flow variations due to blade row interactions.

**Experimental Approach**

The measurements for this study were collected from the three-stage axial compressor research facility at Purdue University. The facility features an inlet guide vane (IGV) followed by three stages designed for flow conditions representative of the rear stages in modern high-pressure compressors, including engine-representative Reynolds numbers and Mach numbers. The design speed for the compressor is 5000 rpm, which is sufficient to achieve appreciable compressibility effects. Furthermore, the four vane rows are circumferentially indexable to capture flowfield variations due to wakes and potential fields. Additional information about the facility is available in Ref. 12.

An extensive study of tip clearance flow effects has been conducted using three separate rotor tip clearance heights. The design intent for these clearances represents 1.5%, 3.0%, and 4.0% as a fraction of overall annulus height, $H$. These tip clearance configurations will be identified as TC1, TC2, and TC3, respectively.

The time-resolved pressure measurements presented herein were collected using a custom-designed array implementing 25 flush-mounted fast-response pressure transducers (Kulite XCS-062-5G). The XCS-series transducers feature high-sensitivity piezoresistive sensing elements to maximize the resolution of the measurements in a small diameter unit (0.066 in. diameter, in this case). This small diameter of the selected sensors provides the ability to incorporate as many sensors as possible in one axial row. Previous authors have utilized a method of offsetting sensors in two or more axial rows, separated by some angle in the pitchwise direction, to accommodate increased axial resolution.\textsuperscript{3,13-14} However, such sensor installations can introduce complications in the measurements for multistage machines from the influence of wakes and potential fields from adjacent vane rows, which must be appropriately considered in both the data acquisition and processing procedures.

The fast-response pressure sensors are permanently installed in a removable block which

![Figure 1. Rotor over-tip instrumentation block.](image-url)
can be inserted into any one of nine frames (one for each rotor and each tip clearance). The removable sensing block and one of the nine frames are shown in Fig. 1. Because the sensor locations are fixed in the removable block but the three rotor rows in the facility differ slightly in their chord length, the sensor locations are listed as a percentage of axial chord for each of the three rotors in Table 1.

<table>
<thead>
<tr>
<th>Blade</th>
<th>Min. Sensor Location [%]</th>
<th>Max. Sensor Location [%]</th>
<th>Separation [%]</th>
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<tr>
<td>Rotor 1</td>
<td>-14.0</td>
<td>114.0</td>
<td>5.33</td>
</tr>
<tr>
<td>Rotor 2</td>
<td>-12.8</td>
<td>113.6</td>
<td>5.26</td>
</tr>
<tr>
<td>Rotor 3</td>
<td>-11.9</td>
<td>113.2</td>
<td>5.21</td>
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The excitation and amplification for these fast-response pressure sensors were supplied by a Precision Filters 28000-series system. Immediately prior to installation and operation in the compressor, the sensors were calibrated in a custom-designed calibration chamber over the full sensing range of 0 to 5 psi, and a least-squares linear fit was applied to the calibration data for each sensor. The overall uncertainty for the measured time-resolved pressures represents approximately 0.030 psi. The data presented herein represent statistical results calculated using 500 revolutions (identified by a once-per-revolution signal from a laser tachometer) of data collected with a sampling frequency of 100 kHz and a low-pass filter applied at a cut-off frequency of 40 kHz. The utilization of a protective screen for the sensors decreases the sensor frequency response to approximately 30 kHz.

The data collected in this study are compared for four loading conditions on the 100% corrected speedline, as identified by the compressor map in Fig. 2: a low loading (LL) condition, a nominal loading (NL) condition with a flow rate slightly higher than the peak efficiency point, a peak efficiency (PE) point, and a high loading (HL) condition at high incidence. In this study, the same loading condition is compared for different tip clearance configurations using equivalent corrected mass flow rates.

Results

Tip Leakage Flow Trajectories

Data collected from a large number of revolutions allow a representation of the pressure measurements using statistical methods. The time-resolved pressure measurements presented herein are evaluated in terms of the root-mean-square (RMS) unsteadiness with respect to the ensemble average:

\[
P_{\text{RMS}}(t_i) = \sqrt{\frac{1}{N} \sum_{k=1}^{N} [P(t_i) - \langle P(t_i) \rangle_k]^2},
\]

at the \(i^{\text{th}}\) time position in a given revolution for \(N\) revolutions of data. In this equation, the ensemble average (EA) is defined by:

\[
\langle P(t_i) \rangle = \frac{1}{N} \sum_{k=1}^{N} [P(t_i)]_k.
\]

This definition of RMS provides the ability to identify regions of pressure unsteadiness associated with the tip leakage flow further into the blade passage compared to the EA static pressure field.

Figure 3 shows the normalized EA static pressures measured over Rotor 1 at the nominal loading condition for each of the rotor tip clearances, and Fig. 4 presents the same data in terms of normalized RMS pressure unsteadiness. The data in these figures represent a mean rotor tip flow, calculated by dividing the ensemble averaged revolution into 36 equal segments (one segment for each blade of Rotor 1) and averaging across these 36 passages. This average result is shown twice, assuming periodicity, to more easily discern the applicable flow features. Also in these figures, dots on the abscissa identify the sensor positions which were used to create the contours. Data collected from malfunctioning sensors are not presented, and the corresponding sensors markers (if any) are absent for such cases.
Figure 4 shows that the unsteadiness in the leakage flow increases as the rotor tip clearance is increased from TC1 to TC3. In particular, the TLF can be identified more clearly past 60% axial chord for the larger tip clearances than for TC1. Furthermore, all three tip clearances show a change of trajectory angle near 50% axial chord which is most significant for TC3. This change in direction of the leakage flow trajectory was also identified by Yoon et al., but the results shown here represent a less severe turn of the leakage flow than was observed by those authors. Chen theoretically predicts this non-linear trajectory only after the leakage flow has exited the rear of the blade passage, where the image vortices required to satisfy kinematic constraints change with the absence of the blade as an effective wall.

Yoon et al. used these RMS pressures to identify the trajectory of the TLF by tracing the locus of peak unsteadiness positions. Using results similar to Fig. 4 for the other loading conditions, the leakage flow trajectories have been identified for the LL, NL, and HL conditions. These trajectories are shown in Fig. 5 (the PE trajectory is not shown due to its relative similarity to NL). For the two high flow rate conditions, NL and LL, Fig. 5 shows that an increase in rotor tip clearance has the effect of moving the leakage flow trajectory closer to the blade suction surface – the same effect as unloading the blade row. The same non-linear leakage flow trajectory noted for Fig. 4 can also be identified at the LL condition. As the loading is increased, the position of peak pressure difference across the blade tip moves toward the leading edge, and the location of leakage flow inception moves upstream. Figure 5 also depicts a noticeable change of the leakage flow trajectory at the high loading condition as the tip clearance increases. Specifically, the two larger clearances,
TC2 and TC3, turn more noticeably away from the axial direction and toward the adjacent blade, whereas the TC1 trajectory portrays a more linear path. This observed change in behavior for the HL condition can be attributed to the relative proximity of the HL points to the stall point for each tip clearance, as outlined in Table 2 (see also Fig. 2) using the following equation for stall margin (SM):

$$SM = \left( \frac{T_{PR}}{m_c} \right)_{stall} - \left( \frac{T_{PR}}{m_c} \right)_{peak} \times 100\%.$$  \hspace{1cm} (3)

The results presented thus far for Rotor 1 have shown that the loading condition of the compressor can affect the trajectory of the leakage flow through the rotor passage, an observation which agrees with findings from previous studies. However, analysis of the Rotor 2 and Rotor 3 results requires an understanding of other interactions which affect the leakage flow in those blade rows. As a result, these interactions will be analyzed before quantifying the leakage flow trajectories for Rotor 2 and Rotor 3.

**Table 2. Stall margin calculated for HL condition.**

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<th>TC1</th>
<th>TC2</th>
<th>TC3</th>
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<tbody>
<tr>
<td>SM</td>
<td>17.5</td>
<td>13.3</td>
<td>9.3</td>
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**Influence of Stator Wakes on Leakage Flow**

In the wake of a stator vane (i.e., Stator 1), the absolute velocity deficit creates a corresponding increase of incidence into the downstream rotor row (e.g., Rotor 2). When the rotor passes through this wake region, the local increase of incidence angle momentarily increases the rotor loading, thereby affecting the strength and trajectory of the tip leakage flow. As a result, the location of the sensors with respect to the stator vanes in the stationary frame of reference will dictate whether or not the pressure measured by those sensors will be affected by the rotor passing through the upstream vane wake.

To better understand this relationship, the same over-rotor static pressure measurements were repeated for several loading conditions with the baseline tip clearance, TC1. A cartoon schematic, Fig. 6, outlines this measurement process. The sensors are in fixed positions in the compressor casing, but the upstream and downstream stator vanes can be moved (either simultaneously or independently) through the use of a series of linear actuators. Thus, Fig. 6 shows representative measurements that may be collected at two vane positions with respect to the fixed measurement.
locations: (a) the first few sensors are located between the upstream stator wakes and measure the freestream flow, or (b) the first few sensors are approximately in the upstream stator wake.

In each case, the time-resolved measurements are phase-locked with the rotor rotation, so the shaded measurement region identifies one blade passage of data. The cartoons in Fig. 6 show measurements over Rotor 2 for which the upstream and downstream stator vanes were moved together, but data were also collected over Rotor 2 when Stator 1 and Stator 2 (the upstream and downstream vane rows) were moved independently from one another.

To assess this effect, the IGV and Stator 1 were moved to 25 unique equally-spaced pitchwise positions with respect to the fixed pressure sensors over Rotor 2. A smooth modulation of static pressures over Rotor 2 can be identified by moving cyclically through the 25 pitchwise positions, but only two out-of-phase positions are shown here for comparison, identified in Fig. 7 as 0% vp and 52% vp. For these results in Fig. 7, the vanes downstream of Rotor 2 (Stator 2 and Stator 3) were maintained in fixed positions to separate effects due to the upstream stator wake from the potential field associated with the downstream vanes.

The pressure RMS unsteadiness results for these two discrete vane positions shown in Fig. 7 have been used to determine the leakage flow trajectories, as explained in the discussion accompanying Fig. 5. This comparison of trajectories, Fig. 8(a), shows an identifiable difference of nearly three degrees between 0% vp and 52% vp. Although this difference may seem small, it is significant compared to the effects associated with changes of tip clearance or loading condition, as will be shown later in this paper.

A procedure similar to the one outlined for Fig. 7 was followed to identify whether the potential field associated with the downstream vanes may also affect the trajectory of the tip leakage flow. In this case, the IGV and Stator 1 were maintained in fixed positions while Stator 2 and Stator 3 were moved simultaneously to 25 pitchwise positions. The time-averaged static pressure downstream of the Rotor 2 trailing edge was used to identify the vane positions associated with the minimum and maximum static pressure field related to Stator 2. The leakage flow trajectories from these vane positions are presented in Fig. 8(b). These results confirm that there is no discernable effect on the trajectory of the tip leakage flow associated with the downstream potential field compared to the effect from the upstream vane wakes.

Although previous authors have identified a stator-rotor interaction governing the development of the rotor tip leakage flow, these measurements help to specifically discern between upstream and downstream vane effects.

**Blade-to-Blade Leakage Flow Variability**

In general, compressor measurements associated with the rotor are often considered in an average form, neglecting variations which may exist from one blade to the next – similar to the results presented in Fig. 3 and Fig. 4. However, small manufacturing defects in the blade hardware and blade row interactions within the machine can create substantial variations which must be considered, as identified in rotor wake variability studies by previous authors.\(^{11,16-18}\) As a result, in addition to the influence of upstream stator vane wakes on the tip leakage flow, it is also desirable to investigate blade-to-blade variations of the tip leakage flow.

In this compressor facility, the runout of the rotor blade tips has been evaluated using a dial
indicator with a rolling tip. These measurements, presented in Fig. 9, represent the blade-to-blade variation of static (non-operating) tip clearance height at any fixed location about the circumference of the compressor. The information in Fig. 9 shows that the blade-to-blade tip clearance varies on the order of $3 \times 10^{-3}$ to $5 \times 10^{-3}$ in. (depending on rotor row) around the circumference of the rotor. However, there is also a particularly large discrepancy for Blade 9 of Rotor 3, which shows a significant increase of tip clearance of approximately $1 \times 10^{-3}$ in. compared to its adjacent blades.

Based on the information in Fig. 9, it is expected that a discernable change of tip leakage flow pattern may be identified for Blade 9 of Rotor 3. Indeed, if the RMS results for Rotor 3 are maintained as 30 separate data sets (one for each blade) instead of an “average blade” representing the entire row, Fig. 10 shows that Blade 9 can be easily identified by its leakage flow pattern which differs noticeably from its adjacent blades. In this case, the local tip clearance increase for Blade 9 creates a larger region of high unsteadiness identified by the RMS which is more dispersed throughout the blade passage, compared with the more localized trajectories crossing the blade passage for the other blades. This phenomenon can be qualitatively compared with the change of leakage flow pattern with increased clearance height in Fig. 4.

Aside from the differences for Blade 9 identified in Fig. 10, blade-to-blade variations of the TLF are also visible for other blades which do not necessarily
correlate with the clearance variations in Fig. 9. Figure 11 shows a similar comparison of blade-to-blade leakage flow variability, but for all 30 blades from the Rotor 3 row at the larger tip clearance TC3. In Fig. 11, the 30 blades have been segmented into three 10-blade series.

For this facility, the difference of blade counts between Rotor 1, Rotor 2, and Rotor 3 (36, 33, and 30, respectively) introduces a 3/rev modulation of the flow field for both Rotor 2 and Rotor 3. Fig. 11 highlights the effect of this 3/rev modulation on the tip leakage flow. An assessment of the EA static pressure signals (not shown) highlights troughs of this 3/rev modulation which exist at blade numbers of approximately 3, 13, and 23, and peaks located at blade numbers of 8, 18, and 28. The data in Fig. 11 identify these same trends as the 3/rev troughs at Blades 3, 13, and 23 show lower leakage flow intensities than the peaks at Blades 8, 18, and 28.

However, the results in Fig. 11 tell an additional story as the 3/rev modulation is superimposed with the 1/rev tip clearance variations identified for Rotor 3 in Fig. 9. A specific comparison of the TLF pattern for Blades 3, 13, and 23 (the troughs of the 3/rev modulation) shows highest intensity in the leakage flow for Blade 23 and lowest intensity (with more dispersed unsteadiness) for Blade 3. Referring to Fig. 9, this observation aligns with the discussion for Fig. 8 that larger clearance heights lead to decreased unsteadiness intensity in the leakage flow region as it is more dispersed through the blade passage and located at an increased distance from the sensor.

Based on the data shown in Fig. 10 and Fig. 11, it is determined that the effect of small blade-to-blade tip clearance variations, such as the one identified for Blade 9 from Rotor 3, are most noticeable in the leakage flow unsteadiness if the tip clearance height is small (i.e., TC1 instead of TC3). This observation could be due to the fact that a $1\times10^{-3}$ in. change represents a larger fraction of the overall clearance height for TC1 (nominally 0.030 in.) than for TC3 (nominally 0.080 in.), but the closer proximity of the sensors to the blade tips for TC1 may also make the sensors more sensitive to these small variations.

Similar to the stator wake interaction highlighted in Fig. 8, the blade-to-blade variations identified in Fig. 10 and Fig. 11 are also important for quantifying changes of the tip leakage flow angle. In this case, the leakage flow angle, $\xi$, is defined with reference to an axial reference datum, as shown in Fig. 12. In Fig. 12, two leakage flow trajectories are shown for Rotor 1 and Rotor 3 at each loading condition, NL and HL. The minimum and maximum traces define the envelope of trajectory angles representing the blade-to-blade variations. This angular envelope for Rotor 1 is 1.8 degrees for both NL and HL. The blade-to-blade variation of trajectory angle shows no discernible trend with the tip clearance height (e.g., Fig. 9), and no rotor-rotor interactions are expected for the Rotor 1 results.

However, several differences exist for the Rotor 3 results in Fig. 12 compared to Rotor 1. In particular, the envelope of blade-to-blade trajectory angle variation for Rotor 3 is 1.5 degrees for NL, but increases to 2.2 degrees for HL. Also, the axial location of leakage flow inception and the leakage flow trajectory angle are more similar between the NL and HL positions for Rotor 3 than for Rotor 1. In this case, although a weak trend of trajectory angle with blade-to-blade tip clearance variation exists, there is no discernible trend associated with the other engine-order frequencies introduced previously. Although not shown here, the Rotor 2 envelopes are nearly identical to the Rotor 3 trends (1.5 and 2.2 degrees for NL and HL respectively).

**Quantifying Tip Leakage Flow Trajectory**

Using the leakage flow trajectory identification techniques presented in this paper, the angle of the leakage flow trajectory can be determined. The
results from the present study are shown for each of the three rotors at the four loading conditions in Fig. 13. Several of the NL and HL data points for TC1 in Fig. 13 also include range bars which identify the variations of tip leakage trajectory angle associated with the modulation of the tip leakage flow due to the upstream vane wakes (as calculated using information similar to Fig. 7). The data in Fig. 13 are presented as a function of measured tip clearances instead of the “nominal” design values. These operating clearances account for thermal growth differences related to changes in loading and ambient temperature discussed by Berdanier and Key.

The results in Fig. 13 highlight several important trends. In particular, the leakage flow trajectory angles appear to change differently for Rotor 1 than for Rotor 2 and Rotor 3, both of which show similar trends at all four loading conditions. For all rotors, however, there is a consistent trend of decreasing trajectory angle with increasing tip clearance for LL, NL, and PE, but an increasing trend for the HL condition. At the HL condition, the relative difference of stall margin between the three clearance configurations has a more profound effect on the flow.

The results for Rotor 1 in Fig. 13 show that the leakage trajectory angles vary almost linearly with increasing rotor tip clearance for the range of clearances investigated in the present study. In contrast to these Rotor 1 observations, the data for Rotor 2 and Rotor 3 show insignificant changes of leakage flow trajectory angle for a tip clearance change from TC1 to TC2. This noted difference for Rotor 2 and Rotor 3 is crucial to this study because it shows a trend which would be otherwise overlooked if only two tip clearances were studied (e.g., TC1 and TC3, as was the case for the data presented by Yoon et al.\textsuperscript{3}). This difference of leakage flow trajectory angles between Rotor 1 and the other two rotor rows agrees with previous observations in this facility suggesting a difference of Rotor 1 performance with increased tip clearance compared to the downstream rotors.

Results similar to the measurements presented in Fig. 13 have also been collected by Yoon et al.\textsuperscript{3} over Rotor 3 in a four-stage low-speed research compressor with repeating stages. A comparison of the Rotor 3 results from the present study with the measurements collected by Yoon et al. is shown in Fig. 14.

Previous authors have introduced the clearance derivative as a method for evaluating the change of a particular parameter with changes in tip clearance. In this case, a clearance derivative for leakage flow trajectory angle is defined with respect to the clearance-to-pitch ratio:

\[ \delta_\xi = \frac{\Delta \xi}{\Delta (\tau/s)} \]  

In Fig. 13 and Fig. 14, the clearance-to-pitch ratio was selected instead of the clearance-to-chord ratio used by Yoon et al.\textsuperscript{3} (or clearance-to-span, which is typically used as a metric for efficiency.
changes) to present trajectory angle changes. The clearance-to-pitch ratio was chosen to avoid discrepancies of aspect ratio between the two facilities compared in Fig. 14. A comparison between the four-stage low-speed compressor used by Yoon et al. with the machine used for the present study shows a difference of aspect ratio from approximately 1.2 to approximately 0.7, respectively.

The relationship between pitch and blade loading, as well as the contribution of pitch as a representative length scale in the measurement plane over the rotor, combined to guide the selection of clearance-to-pitch as the representative metric for comparison. However, the relative dissimilarity of blade pitch for Yoon et al. and the present study creates an opportunity to evaluate the dependence of observed trends to a particular compressor design.

A comparison of the results from the present study in Fig. 14 with the data presented by Yoon et al. shows good agreement using the clearance-to-pitch ratio on the abscissa. Especially at the LL and HL conditions, the clearance derivatives are nearly identical when the widest tip clearance range is considered for the present study (TC1 to TC3). However, the absence of a third tip clearance configuration for Yoon et al. makes it difficult to conclude whether the same non-linear trend is expected for any multistage compressor. For reference, if the Rotor 2 results from the present are selected for comparison with Yoon et al. instead of Rotor 3, the results are also similar based on the nearly identical trends between Rotor 2 and Rotor 3 in Fig. 13.

As a rule-of-thumb, these data suggest that the leakage flow trajectory angle will decrease approximately one degree for every one percent increase of clearance-to-pitch when rotor incidence angle is large and negative. Similarly, approximately one degree of trajectory angle increase is expected for every one percent of clearance-to-pitch when rotor incidence is large and positive. Although these data show trends which may apply for large clearance changes, it is very important to observe that a critical clearance may exist, leading to significant differences of the clearance derivative for leakage flow angle assessed here. For the present study, the critical clearance represents the TC2 clearance height. By this proposed effect, a change from a “small” to a “moderate” clearance height (i.e., TC1 to TC2) may be met by little or no change of leakage flow angle, whereas a change from a “moderate” to “large” clearance height (i.e., TC2 to TC3) may incur significant changes.

Further comparison of the present study results in Fig. 14 with the data from Yoon et al. also shows that the peak efficiency point from the comparison study has a positive clearance derivative which was not observed in the present study. Based on the trends suggested here, it is assumed that the rotor may operate at a higher loading at the peak efficiency point for the machine investigated by Yoon et al. than the compressor in the present study.

The range bars in Fig. 14 show that the leakage flow modulation associated with the upstream vane wake has a profound influence on the leakage flow. In fact, the change of leakage flow angle due to this stator-rotor interaction is greater than the change due to a doubling of the tip clearance height (TC1 to TC2). Additional research is required to determine whether the observed leakage flow angle ranges associated with the wake interactions shown for TC1 in Fig. 13 are similarly present for other clearance configurations and loading conditions. At this point, however, there is sufficient information to determine that the stator-rotor interaction plays a significant role in the development of the tip leakage flow, and this relationship must be appropriately considered when comparing experimental data with computational results.

**Summary and Conclusions**

This study has characterized the tip leakage flow trajectory for three rotor tip clearance heights at four loading conditions on the 100% corrected speedline. Changes of the leakage flow trajectory angle with
increased tip clearance are dependent on the loading condition and the proximity to the stall point. Although previous authors have identified an interaction between the rotor tip leakage flow and the adjacent stator vanes, a specific effort was performed here to separate the effect from the upstream stator vane wake from a potential effect associated with the downstream stator. The results from this study showed that the downstream stator vane has a negligible effect on the trajectory of the tip leakage flow. This quantitative evaluation of the significance of this stator-rotor interaction showed the modulation of the leakage flow angle due to the upstream stator vane wake can be more significant than the trajectory change due to a doubling (1.5% increase) of rotor tip clearance height, especially if tip clearance heights are small.

In addition to the influence of the upstream stator vanes on the rotor tip leakage flow, blade-to-blade variations of the leakage flow have also been investigated with a consideration of rotor-rotor interactions. Multistage effects in the compressor utilized for this study create a modulation of the tip leakage flow as a result of a difference of rotor blade counts for adjacent stages. However, separate blade-to-blade variations of the tip leakage flow were also identified as a result of blade-to-blade clearance changes associated with manufacturing tolerances. Although these manufacturing variations exist to some extent in all machines, their effect is often overlooked in experimental methods and computational modeling in favor of a representative average.

The present study emphasizes the importance of using at least three tip clearance configurations to achieve meaningful conclusions from tip clearance studies. Specifically, Rotor 2 and Rotor 3 revealed the presence of a critical clearance, for which smaller clearances showed little or no change of trajectory angle, but larger clearances show noticeable changes. This observation would be overlooked if only two tip clearance heights were investigated. However, a comparison of the overall change from small to large clearance with the results from Yoon et al.\,\(^3\) shows good agreement. Specifically, the trends of trajectory angle collapse when the clearances are normalized by rotor pitch. This comparison suggests a rule-of-thumb clearance derivative for leakage flow angle of approximately negative one for low loading conditions and positive one for high loading conditions (as a function of clearance-to-pitch). Although this rule-of-thumb applies on the average for large tip clearance changes, it comes with the understanding that critical tip clearance heights may exist at which these trends change significantly.

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