

**Draft-GT2017-65099****AN EXPERIMENTAL STUDY OF STALL SUPPRESSION AND ASSOCIATED CHANGES TO THE FLOW STRUCTURES IN THE TIP REGION OF AN AXIAL LOW SPEED FAN ROTOR BY AXIAL CASING GROOVES****Huang Chen**Johns Hopkins University  
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Baltimore, MD, USA**ABSTRACT**

The effects of axial casing grooves on the performance and flow structures in the tip region of an axial low speed fan rotor have been studied experimentally in the JHU refractive index-matched liquid facility. The four-per-passage semicircular grooves are skewed by 45° in the positive circumferential direction, and have a diameter of 65% of the rotor blade axial chord-length. A third of the groove overlaps with the blade front, and the rest extends upstream. These grooves have a dramatic effect on the machine performance, reducing the stall flow rate by 40% compared to the same machine with a smooth endwall. However, they reduce the pressure rise at high flow rates. The flow characterization consists of qualitative visualizations of vortical structures using cavitation, as well as stereo-PIV measurements in several meridional and  $(z,\theta)$  planes covering the tip region and interior of the casing grooves. The experiments are performed at a flow rate corresponding to pre-stall conditions for the untreated groove. They show that the flow into the downstream side of the grooves and the outflow from their upstream sides vary periodically. The inflow peaks when the downstream end is aligned with the pressure side (PS) of the blade, and decreases, but does not vanish, when this end is located near the suction side (SS). These periodic variations have three primary effects: First, substantial fractions of the leakage flow and the tip leakage vortex (TLV) are entrained periodically into the groove. Consequently, in contrast to the untreated flow, The TLV remnants remain confined to the vicinity of the entrance to the groove, and the TLV strength diminishes starting from the mid-chord. Second, the grooves

prevent the formation of large scale backflow vortices (BFVs), which are associated with the TLV, propagate from one blade passage to the next, and play a key role in the onset of rotating stall in the untreated fan. Third, the flow exiting from the grooves causes periodic variations of about 10° in the relative flow angle around the blade leading edge, presumably affecting the blade loading.

**INTRODUCTION**

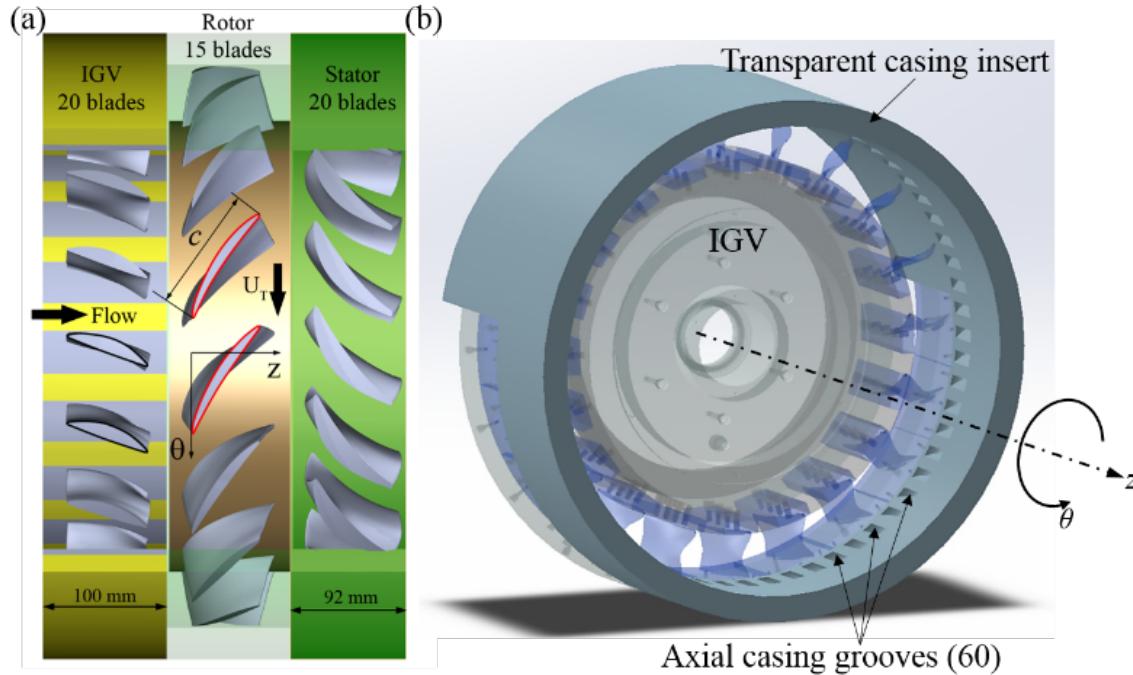
Casing treatments have been used successfully to delay the onset of stall in axial turbomachines with limited penalty in efficiency [1,2]. Among the different approaches, axial casing grooves (ACGs) have been found to be the most effective [3]. Considerable effort has already been invested to understand the flow mechanism involved and to optimize the geometry of ACGs. For example, Takata and Tsukuda [4] show experimentally that the stall is delayed only when the ACG is skewed in the direction of the rotor rotation, and that the grooves generate periodic high-speed jets in front of the blade, which increase the incidence angle at the entrance to the passage. Smith and Cumpsty [5] observe that ACG cause a considerable increase in blade loading near the tip, presumably by reorienting the flow. SPIV measurements in a transonic compressor rotor by Brandstetter et al. [6] show that the ACGs generate periodic jets and suppress the tip region blockage. Experimental and numerical studies by Müller et al. [7], using ACGs that are very similar to those used in the present paper, show a 23% increase in stall margin, and associated reduction

in tip region blockage. Crook et al. [8] show numerically that the improvement to the stall margin by ACG involves suppression of the TLV-induced blockage by sucking the low energy fluid and reinjecting it at high speed in front of the passage. Recent studies mostly involve RANS simulations. For example, Beheshti et al. [9] show that the ACGs are more effective in rotors with large tip gap. Wilke et al. [10] associate the improvement with suppression of the tip leakage flow and reduction in TLV strength. Recommendations for groove geometries based on steady-state single passage simulations are provided by Djeghri et al. [11]. They show that the skew angle and axial position of the groove have more impact on the stall margin improvement than the slot depth and shape.

Among the many reasons for the difficulties in understanding why ACGs delay stall, two are particularly evident. First, the processes involved in the onset of stall are not fully understood [12], and are subject for on-going research, e.g. on “modal-type” [13–15] and “spike-type” stall [15–20]. Houghton and Day [21] argue that groove-based casing treatments are most effective in delaying the intermittent spike-type stall. Second, due to challenges in accessing the tip region of turbomachines, there is very limited experimental data on the flow there, with or without grooves. The refractive index-matched facility at JHU addresses this limited access by matching the refractive index of the acrylic blades and casing with that of the working fluid, an aqueous solution of sodium iodide. Taking advantage of the resulting unobstructed optical access, previous studies have already examined tip clearance

flows [22–26], and recent work [27–29] has focused on the machine described in the present paper, without casing grooves. In particular, Chen et al. [29] show that the onset of stall involves formation and propagation of a large scale backflow vortex (BFV). They form radially inward from the TLV, at the interface between the swirling and jetting tip leakage flow and the main passage flow. Once formed, these structures propagate to the next passage either across the tip gap or upstream of the blade leading edge.

To study the flow structure generated by ACGs, we have installed an acrylic ring containing axial casing grooves with geometry similar to that studied by Müller et al. [7]. Performance tests, qualitative flow visualization by cavitation and quantitative SPIV measurements in multiple planes elucidate the flow in the tip region and within the casing grooves. As discussed below, performance tests show that the grooves cause substantial improvements in performance and delay the onset of stall. These improvements appear to be related to three effects: First, substantial fractions of the leakage flow and the TLV are entrained periodically into the groove, greatly reducing the TLV strength, and confining the TLV remnants to the downstream/entrance corner of the groove. Second, the grooves prevent the formation of backflow vortices that play a dominant role in the onset of stall in the untreated machine. Third, as the flow exiting from the grooves propagates back to the blade, it causes periodic variations of about  $10^\circ$  in the relative flow angle around the blade leading edge, presumably affecting the blade loading periodically.



**Figure 1:** (a) Configuration of the one and a half stages axial low speed fan and (b) a perspective view of the IGV with axial casing grooves.

## EXPERIMENTAL SETUP

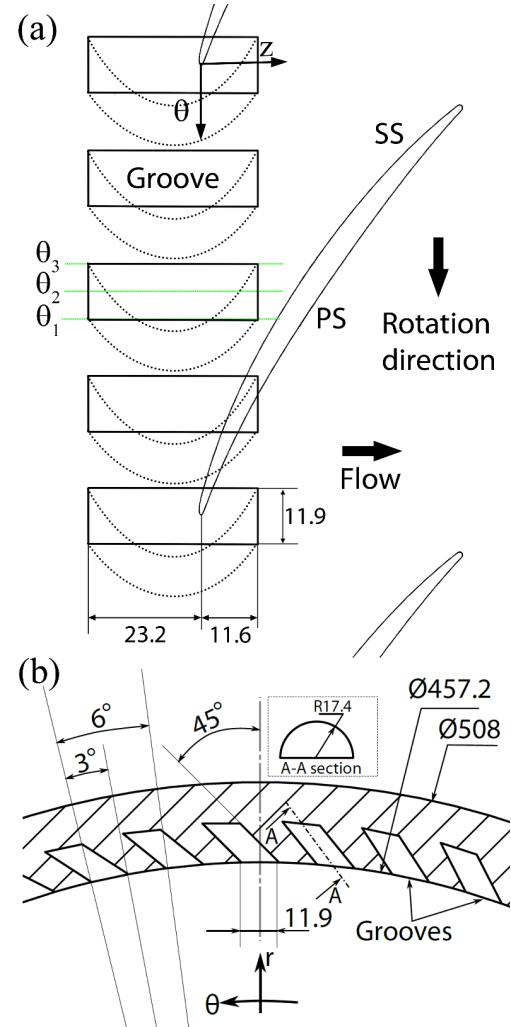
As shown in Fig. 1(a), the one-and-a-half stages low speed axial fan consists of a 20-blade inlet guide vane (IGV), a 15-blade rotor and a 20 blades stator. The blades are based on the low-speed axial compressor (LSAC) facility at NASA Glenn, but their aspect ratio is reduced to accommodate operation in a liquid facility [24,30]. Relevant geometrical parameters of this machine, which has the same rotor as that used in [29] and the wide tip clearance in [28] are listed in Table 1. The tip clearance, as measured directly from the PIV images, is slightly smaller than that reported in these papers, most likely due to slight differences in rotor eccentricity. The casing and the rotor blades are made of acrylic, whose refractive index is the same as that of the fluid, a 62%-63% by weight aqueous solution of sodium iodide. The kinematic viscosity of this fluid is  $1.1 \times 10^{-6} \text{ m}^2 \text{s}^{-1}$  [31], and its specific gravity is 1.8. As illustrated by the perspective view in Fig. 1(b), the casing grooves are machined in an acrylic ring that surrounds the three blade rows. The groove dimensions, location and orientation are shown in Fig. 2, and/or Table 1. The semi-circular grooves have a diameter of  $0.65c_A$ , out of which, 33% overlaps with the rotor, and the rest, extends upstream of the rotor. The evenly spaced, four per passage grooves are skewed by  $45^\circ$  in the positive circumferential direction, i.e. the same as the rotor rotation (Fig. 2b).

**Table 1** Stage relevant geometrical parameters

Casing diameter ( $D$ ) (mm)	457.2
Hub radius ( $r_{\text{hub}}$ ) (mm)	182.9
Rotor passage height ( $L$ ) (mm)	45.7
Rotor diameter ( $D_R$ ) (mm)	453.6
Rotor blade chord ( $c$ ) (mm)	102.6
Rotor blade span ( $H$ ) (mm)	43.9
Rotor blade stagger angle ( $\gamma$ ) (deg)	58.6
Rotor blade axial chord ( $c_A$ ) (mm)	53.5
Measured tip clearance ( $h$ ) (mm)	1.8 (0.0175 $c$ or 0.041 $H$ )
Axial casing groove diameter (mm)	34.8
Groove skew angle (deg)	45
Total number of grooves	60
Shaft speed ( $\Omega$ ) (rad $\text{s}^{-1}$ ) {RPM}	50.27 {480}
Rotor blade tip speed ( $U_T$ ) (m $\text{s}^{-1}$ )	11.47
Reynolds number ( $U_T c / \nu$ )	$1.07 \times 10^6$

Detailed descriptions of the refractive index matched test loop can be found in [22–24]. Briefly, the fan speed is fixed at 480 RPM and the resulting Reynolds number based on the rotor chord length and tip speed is  $1.07 \times 10^6$ . The volumetric flow rate is measured by integrating the velocity profile obtained by translating a Pitot tube radially far downstream of the test section. The mean pressure rise across the machine is measured by a differential pressure transducer connected to pressure ports

located upstream of the IGV and downstream of the stator. The performance is determined while operating the machine under the same conditions for several minutes, and results are based on averaging 200,000 points. The term performance refers to the pressure rise across the machines, and that the present study does not include measurements of efficiency. Cavitation resulting from running the facility at low mean pressure is used for visualizing vortical structures. The SPIV measurements are performed while keeping the facility at a much higher mean pressure to suppress all forms of cavitation.

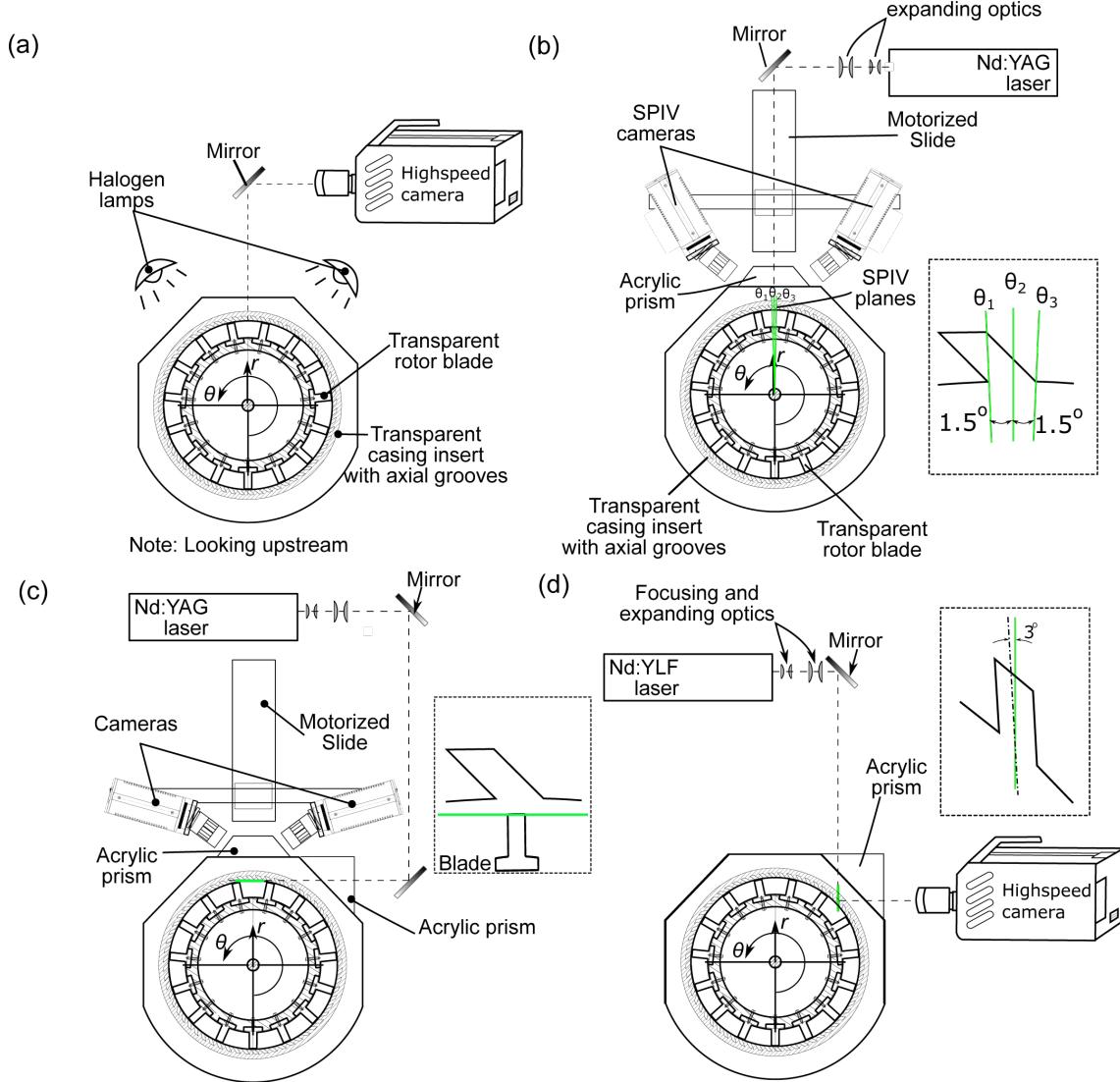


**Figure 2:** The ACG configuration: (a) radial view, and (b) streamwise view (looking upstream). All dimensions are in mm.

Optical setups used in this study are shown in Fig. 3. During cavitation visualizations, the tip region is illuminated by two continuous halogen lamps shown in Fig. 3a. Images have been recorded by a PCO. dimax high-speed CMOS camera at 5760 frames/s, corresponding to 48 images per blade passage,

at a resolution  $960 \times 792$  pixels. The field of view is  $143.6 \times 118.5 \text{ mm}^2$ . Index matching allows acquisition of PIV based data in any desired plane. The setup for SPIV measurements in meridional planes is illustrated in Fig. 3b. The flow field is seeded with  $13\mu\text{m}$ , silver coated hollow spherical glass spheres that have a specific gravity of 1.6, and illuminated by dual head, 200mJ/pulse Nd:YAG laser expanded to a laser sheet thinner than 1mm. The images are recorded by a pair of Imperx B6640 CCD cameras that have  $6600 \times 4400$  pixel arrays. The total field of view is  $44.9 \times 41.1 \text{ mm}^2$ , and the delay between exposures is  $20\mu\text{s}$ . Imaging conditions are satisfied by inclining the lenses by the Scheimpflug angles. Observations are performed through acrylic prisms whose surfaces are perpendicular the lens axes. Data has been recorded in three equally spaced circumferential angles separated by  $\Delta\theta = 1.5^\circ$ ,

which are labeled as  $\theta_1$ ,  $\theta_2$  and  $\theta_3$ . Angles  $\theta_1$  and  $\theta_3$  are located close to the inside edges of a casing groove without cutting through them. Angle  $\theta_2$  cuts the entrance to the groove. As shown in Fig. 3c, SPIV measurements have also been performed at a  $(z, \theta)$  plane, where the radially outer edge of the laser sheet coincides with the blade tip ( $r^*=0.96$ ). In this case, the images are recorded by a pair of  $2048 \times 2048$  pixels, PCO. 2000 cameras, also using exposure intervals of  $20\mu\text{s}$ , and the field of view is  $40.2 \times 52.2 \text{ mm}^2$ . To complete the picture, especially unsteady flow features, time-resolved 2D PIV measurements at  $2.4 \text{ kHz}$  have been performed in a plane that is nearly parallel to the center plane of the grooves (Fig. 3d). In this case, the image pairs have been recorded by the PCO. dimax camera at a resolution of  $864 \times 892$  pixels, using frame straddling to achieve an in-pair delay of  $40\mu\text{s}$ .



**Figure 3:** Experimental setups for (a) cavitation flow visualization, (b) SPIV in three meridional planes, (c) SPIV in a  $(\theta, z)$  plane intersecting the blade tip at  $r^*=0.96$ , and (d) 2D time-resolved PIV inside a groove.

Following Wienke [32], two steps are used for calibrating the SPIV images. Coarse calibration is performed by raising the entire camera and illumination system vertically as a unit and translating a dotted target immersed in a box filled with the same fluid. Subsequently, the system is lowered, and the so-called fine-resolution self-calibration is performed in the actual sample area. Pre-processing of the images consists of background subtraction [33], followed by contrast enhancement using a modified histogram equalization algorithm described in [34]. The FFT-based commercial PIV package, LaVision DaVis, is used for multi-pass cross-correlation of the image pairs. For the meridional planes, the final interrogation window size is  $32 \times 32$  pixels, and with 50% overlap, achieves a vector spacing of 0.137mm. For the  $(z, \theta)$  plane, the final window size is  $24 \times 24$  pixels, providing a vector spacing of 0.214 mm with 50% overlap. For the 2D data, the final window size is  $32 \times 32$  pixels, and using 75% overlap, the vector spacing is 0.345mm. The data are post-processed by a universal outlier detection algorithm [35] that rejects spurious vectors. Based on our previous studies [23], the uncertainty in instantaneous velocity is about 0.1 pixel, corresponding to 0.4~0.8% of the tip speed provided there are more than 5 particles in each interrogation window. The uncertainty in ensemble-averaged velocity is at least an order of magnitude smaller.

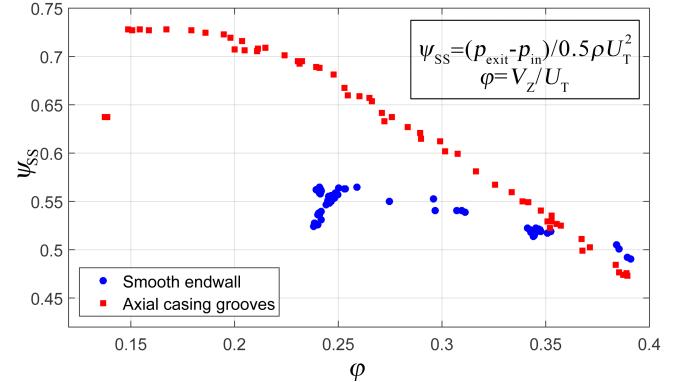
Data are presented in a cylindrical coordinate system  $(r, \theta, z)$  whose origin is located at the center of the fan and coincides with the blade tip leading edge. The corresponding instantaneous velocity components are  $u_r$ ,  $u_\theta$ , and  $u_z$ , and the rotor rotates in the positive circumferential direction. The normalized radial coordinate, defined as  $r^* = (r - r_{\text{hub}})/L$ , is used throughout this paper. At least 2500 instantaneous realizations are obtained for every setup and rotor blade orientation (termed phase). Ensemble-averaged velocity components, denoted as  $U_i$ , are calculated by averaging all the data obtained for each phase. Velocity fluctuations are defined by  $u_i' = u_i - U_i$ , i.e. by deviation of the instantaneous data from the ensemble-averaged result. The orientation/phase of the blade relative to the groove is represented by the blade chord fraction ( $s/c$ ) that the  $\theta_2$  plane (Figs. 2 and 3) intersecting the blade chord. For clarity,  $s/c=0$  represents the phase where  $\theta_2$  is aligned with the blade LE. The SPIV data have been recorded at 14 different phases, covering an entire blade passage, but the planes are not distributed evenly, with finer separation of  $\Delta s/c=0.055$  around the leading edge, and coarser spacing,  $\Delta s/c=0.11$  for high chord fractions.

## RESULTS AND DISCUSSIONS

### Effects of ACGs on the Performance

The performance curves with and without casing grooves are compared in Fig. 4. The flow coefficient is defined by  $\varphi = V_z/U_T$ , where  $V_z$  is the volumetric flow rate divided by the annular area of the rotor passage. The pressure coefficient is  $\psi_{\text{SS}} = (P_{\text{exit}} - P_{\text{in}})/0.5\rho U_T^2$ , where  $P_{\text{exit}} - P_{\text{in}}$  is the pressure difference

across the entire machine. At high flow rates, the grooves cause a slight decrease in pressure rise. However, for most of the measured range, they improve the performance substantially, by as much as 17% at  $\varphi \sim 0.25$ . Furthermore, while the onset of stall in the untreated machine occurs at  $\varphi \sim 0.25$  [29], stall does not occur in the grooved case until  $\varphi < 0.15$ , where the performance drops rapidly. For both cases, the rapid decline in performance is accompanied by an audible increase in vibration-induced noise. Hence, the axial grooves reduce the flow rate at the onset of stall by 40%.



**Figure 4:** Performance curves for  $h/c=1.8\%$ , with and without the axial casing grooves.

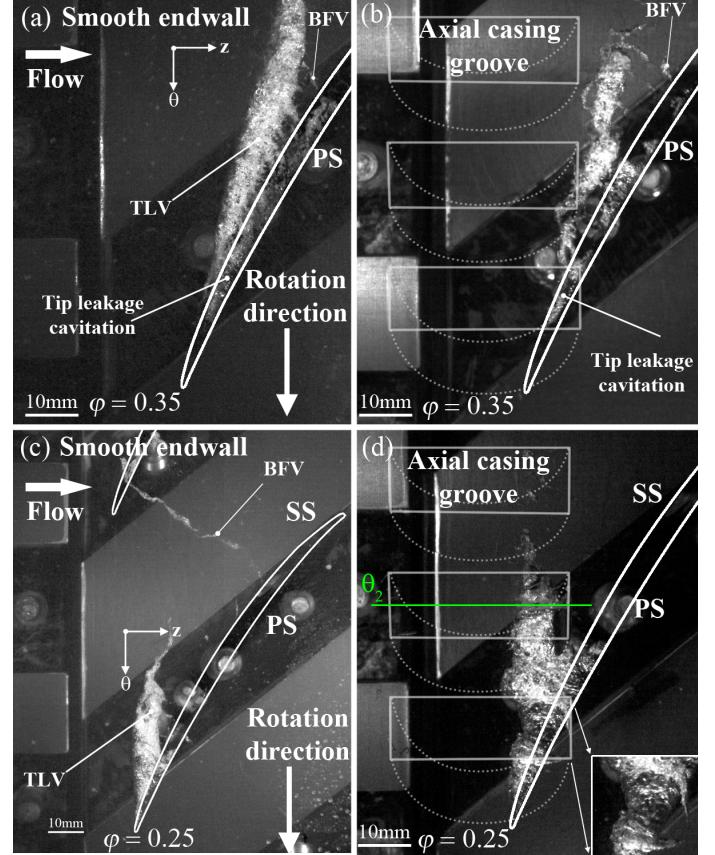
### Visualization of flow structures by cavitation

Figure 5 shows sample images of cavitation at two different flow rates comparing the TLV structure with a smooth casing to that developing when the ACGs are installed. The blade tip profiles and locations of the grooves are highlighted for clarity. At  $\varphi=0.35$ , when the performance of the two machines is not substantially different, Figs. 5a and 5b represent a phase for which one of the grooves is located above the initial rollup point of the TLV in the untreated case. The presence of tip leakage cavitation (see annotations) indicates that there is a leakage flow in both cases. However, the point where a distinct TLV first appears on the SS is located further along the blade in the grooved endwall.

Further downstream, the TLV in the untreated rotor gradually detaches from the blade, and small scale backflow vortices (BFVs) aligned perpendicularly to the blade begin to form between the SS corner and the TLV (marked as BFV). The mechanisms generating them and their influence on the onset of stall are the focuses of our previous publication [29]. For the treated endwall (Fig. 5b), the TLV also begins to migrate away from the blade between the first and the middle grooves (bottom and middle grooves in Fig. 5b). Yet, near the middle groove, the TLV becomes confined to the space between this groove and the blade, resulting in kinks in the TLV trajectory on both sides of this groove. Further downstream, where the aft parts of the blade that are not located under the grooves, the TLV detaches from the blade again, and several BFVs start to appear as well.

At  $\varphi=0.25$ , the pre-stall condition of the untreated endwall, the TLV rolls up near the blade leading edge, and migrates away from the SS at a shallow angle with the circumferential direction. The cavitation disappears at mid passage, as the vortex breaks up under the influence of the BFVs that appear intermittently in the middle of the rotor passage (e.g. Fig. 5c). As discussed in Chen et al. [29], the formation and circumferential propagation of the BFVs diagonally upstream, and their penetration across the tip gap or around the leading edge of the next blade triggers generation of similar structures in the next passage, and so on. The BFV vorticity is originated from the leakage flow, but it rolls up radially inward from the TLV. As the BFV crosses the tip gap, the TLV rollup is disrupted, and in some cases pushed further along the blade briefly, suggesting a temporary reduction in the local blade loading. This process is intermittent under pre-stall conditions, but is a major contributor to the substantial increase in turbulence level over a broad area. At the onset of stall (e.g. at  $\varphi=0.24$ ), the BFVs roll up earlier, and appear more frequently, sometimes more than one per passage. They also grow more rapidly and to a larger size. Once formed, their propagation from one passage to the next persists for a while. Furthermore, they appear to rotate with the blade, while migrating slowly downstream, indicating that the flow the entire tip region is nearly blocked. Hence, generation and circumferential propagation of the BFVs is a primary phenomenon associated with the onset of stall.

For the grooved endwall and  $\varphi=0.25$ , Fig. 5d shows that the TLV boundary becomes wavy, with what appears to be a narrower waist under the first groove combined with decreased tip leakage cavitation in the same area. The high-speed movies show that the waist is formed as part of the TLV cavitation is entrained into the groove. Further away from the blade, the TLV tail seems to diminish under the middle groove, leaving very little cavitation beyond it. As demonstrated quantitatively later, this phenomenon is caused by entrainment of substantial parts of the TLV into this groove, leaving limited amount of circulation in the main channel. Most importantly, however, is the disappearance of the BFVs, both large and small ones. Hence, the grooves eliminate or at least greatly suppress the primary flow phenomenon propagating circumferentially from one blade passage to the next under pre-stall and stall conditions in the untreated passage. Both effects, namely the periodic suction of TLV and elimination of the BFVs persists at flow rates much lower than  $\varphi=0.25$  (not shown). Once stall occurs at  $\varphi<0.15$ , the TLV disappears completely periodically, indicating a broad loss of blade loading over more than one blade passage at a time. We have not focused on this domain fearing for the ability of the blades to withstand the unsteady loading for extended periods.



**Figure 5:** Sample cavitation images showing vortical structures in the rotor passage without (left column) and with ACG (right column). Top row (a and b):  $\varphi = 0.35$ , and bottom row (c and d):  $\varphi = 0.25$ . Entrance to the grooves are indicated by solid gray lines, and their outlines are marked by dashed lines. Insert in (d) is a magnified view of the groove corner.

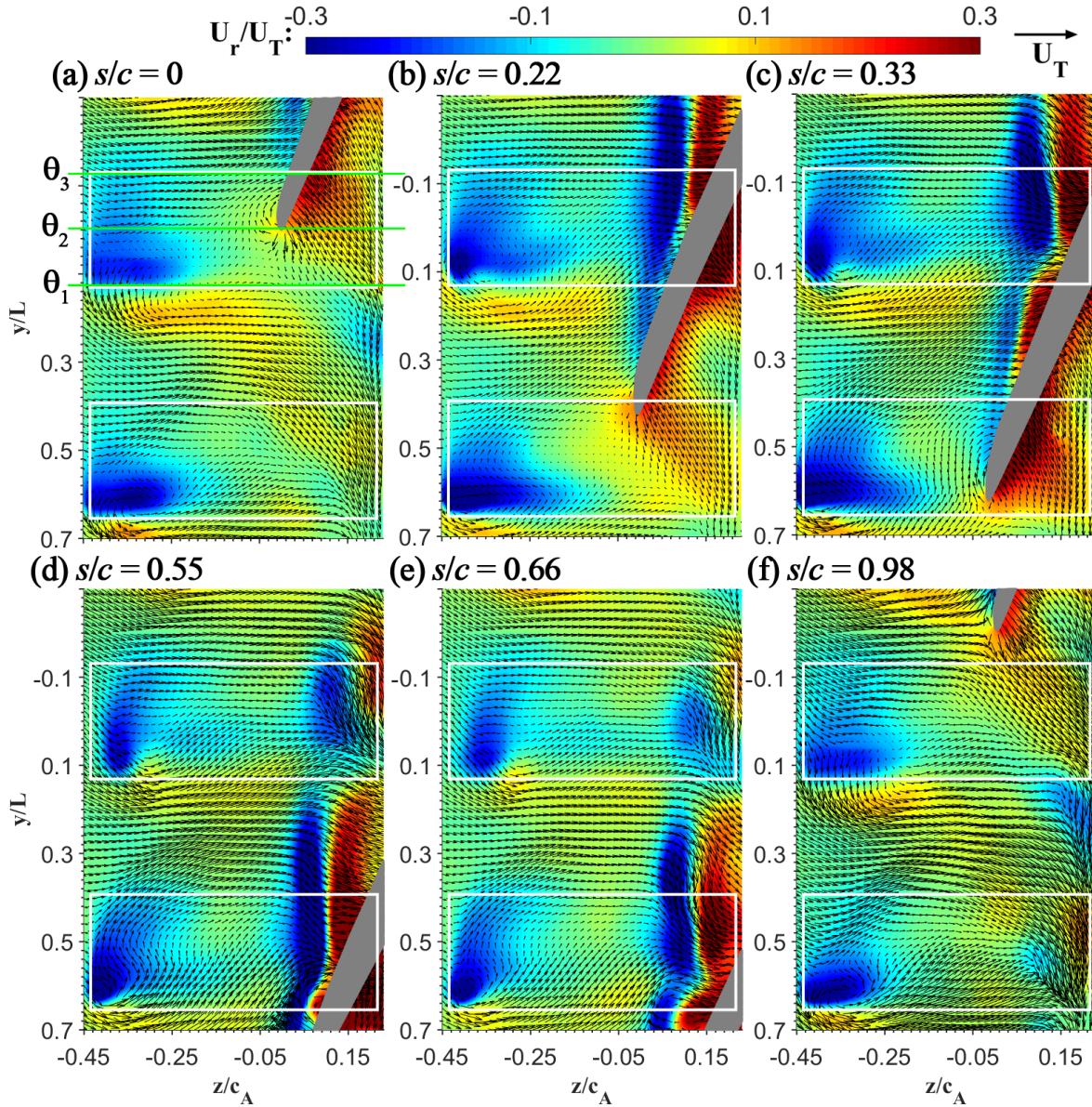
#### Flow characteristics in the $(z, \theta)$ plane

All the PIV data presented in this paper has been acquired at  $\varphi=0.25$  in order to focus on mechanism preventing the onset of stall. Figure 6 shows six super-positions of ensemble-averaged in-plane velocity vector fields and color contours of  $U_r$  in a  $(z, \theta)$  plane intersecting the blade tip. The field of view contains two grooves, and their locations are highlighted, along with the location of the three meridional planes ( $\theta_1$ ,  $\theta_2$ , and  $\theta_3$ ) discussed subsequently. The selected phases elucidate the impact of the grooves on the flow around the blade tip, including the vicinity of the TLV. Corresponding cavitation images for two phases,  $s/c=0.22$  and  $0.33$  based on the location of  $\theta_2$  are provided in Fig. 7 for comparison. Several trends are evident:

(i) Consistent with [29], there is generally an inflow into the groove on the downstream side, which overlaps with the blade, and outflow from the groove on the upstream side. The magnitudes and distributions of inflow and outflow vary spatially and temporally, depending on the orientation of the blade relative to the grooves. Starting with the blade effect,

$U_r > 0$  along the PS of the blade, as expected. Furthermore,  $U_r$  transitions from positive to negative values along the TLV center on the SS, making this transition a convenient qualitative means of identifying the trajectory of the TLV center. A comparison to Fig. 7 shows that the upstream edge of the narrow negative  $U_r$  area nearly coincides with the upstream edge of the TLV cavitation, confirming that the positive and negative  $U_r$  layers correspond to the TLV-induced velocity. In addition, the measurements in meridional planes discussed later confirm that the transition in  $U_r$  sign corresponds to the TLV center.

(ii) Except for periods when the downstream end of the groove is facing the SS of the blade (e.g. Fig. 6c) and strongly



**Figure 6:** Ensemble-averaged in-plane velocity vectors (nearly  $U_z$ ,  $U_\theta$ ) superimposed on color contours of the radial velocity component ( $r^*=0.96$ ). Vectors are diluted by 4:1 in both directions for clarity. The values of  $s/c$  indicate the intersection of the  $\theta_2$  plane with the blade chord.

affected by the TLV,  $U_r$  is largely positive at the downstream end (the inflow/entrance end) of both grooves. This radial inflow, combined with  $U_\theta > 0$  consistent with the groove angle, increases when the entrance is aligned with the PS of the blade. Such an increase occurs e.g. when the LE is aligned with  $\theta_2$  ( $s/c=0$ ) in Fig. 6a, or when the LE is aligned with the lower groove at  $s/c=0.33$  (Fig. 6c). The magnitude of  $U_r$  peaks as the PS approaches the downstream edge of the upper groove in Fig. 7b and the lower groove in Fig. 6d.

(iii)  $U_r < 0$  over the entire upstream/exit end of the grooves, with the exit flow being persistently higher at the high  $\theta$  corner of the grooves, e.g. the corner corresponding to  $\theta_1$  in the upper one.

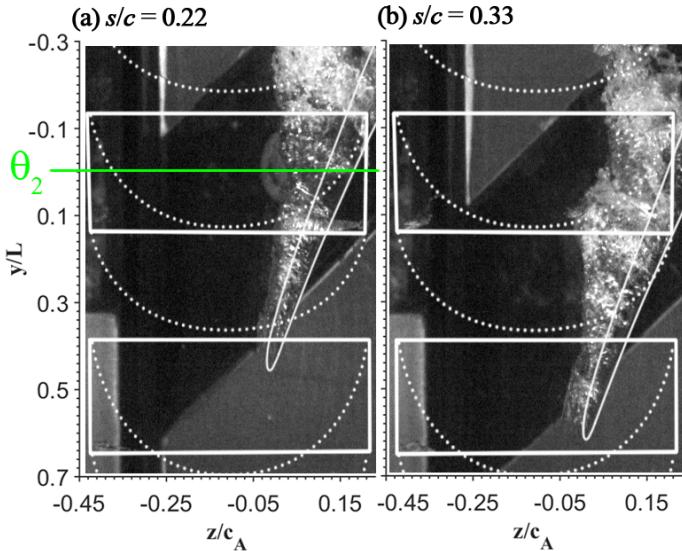


Figure 7: Cavitation images at (a)  $s/c = 0.22$  and (b)  $s/c = 0.33$ .

(iv) Discontinuities in the TLV trajectory. When the blade SS is facing the entrance to the groove, Fig. 6c shows that the TLV signature is divided into two parts, consistent with the corresponding cavitation images (Fig. 7). The first part extends from the LE to vicinity of  $\theta_1$ . It is characterized by a TLV confined to the vicinity of the blade SS with a weak vortex-induced velocity. The second part originates from the region facing the groove entrance, close to the SS surface, where the in-plane flow appears to converge towards this origin. The (positive and negative) TLV-induced velocity is initially quite high, reaching 30% of the tip speed, and the vortex center remains aligned in the circumferential direction, not far from the downstream edge of the groove. As the SS moves away from this groove, e.g. in Figs. 6d and e, the remnants of the TLV-induced radial velocity decrease in magnitude, but remain “latched” to the downstream corner of the groove. As demonstrated below, the decreasing TLV signature is associated with partial entrainment of the TLV into the groove. These decaying TLV remnants finally disappear in Fig. 6f, only as the entrance to the groove begins to be influenced by the PS of the next blade and associated increase in radial inflow. Essentially the same phenomenon occurs around the downstream end of the lower groove, resulting in two TLV discontinuities in Fig. 6d. These grooves are located at different circumferential angles relative to the location of the IGV blades, suggesting that the IGV or its wake are not playing primary roles in the observed phenomena.

(v) The distribution of in plane velocity components ( $U_z, U_\theta$ ) vary substantially with blade orientation relative to the groove but persistently show the stream originating from the exit and propagating diagonally towards the blade with  $U_\theta < 0$  and  $U_z > 0$ . As shown later, this stream causes fluctuations in the relative flow angle around the blade LE. The direction of  $U_\theta$  is negative in most of the area overlapping with the groove, as expected, but transitions rapidly to positive values near the  $\theta_1$

corner, followed by a sharp turn towards the axial direction. The latter phenomenon is likely caused by the main passage flow circumventing the jet exiting from the groove.

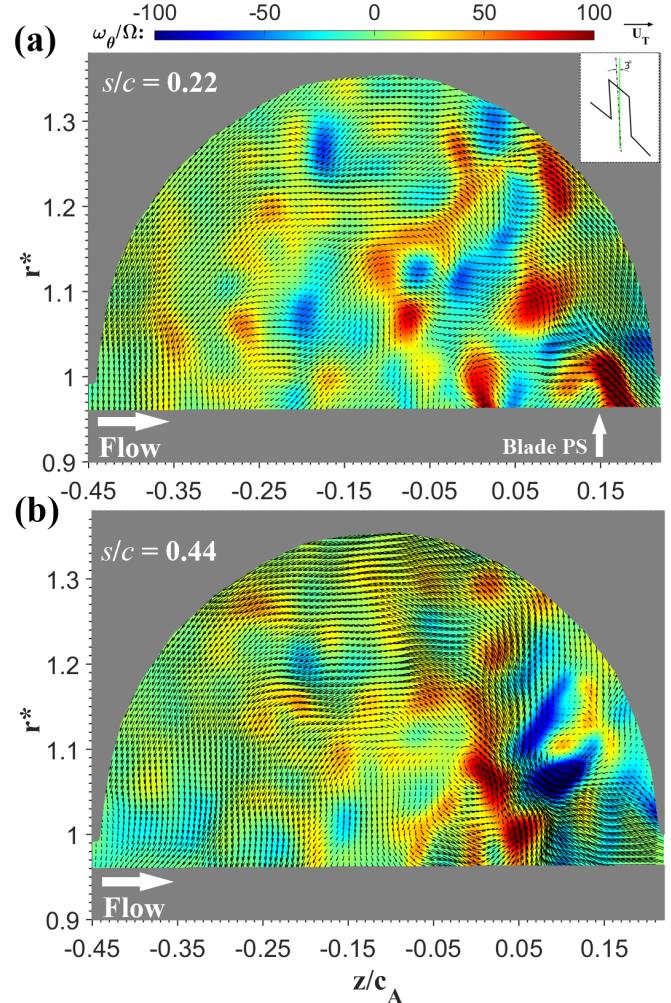
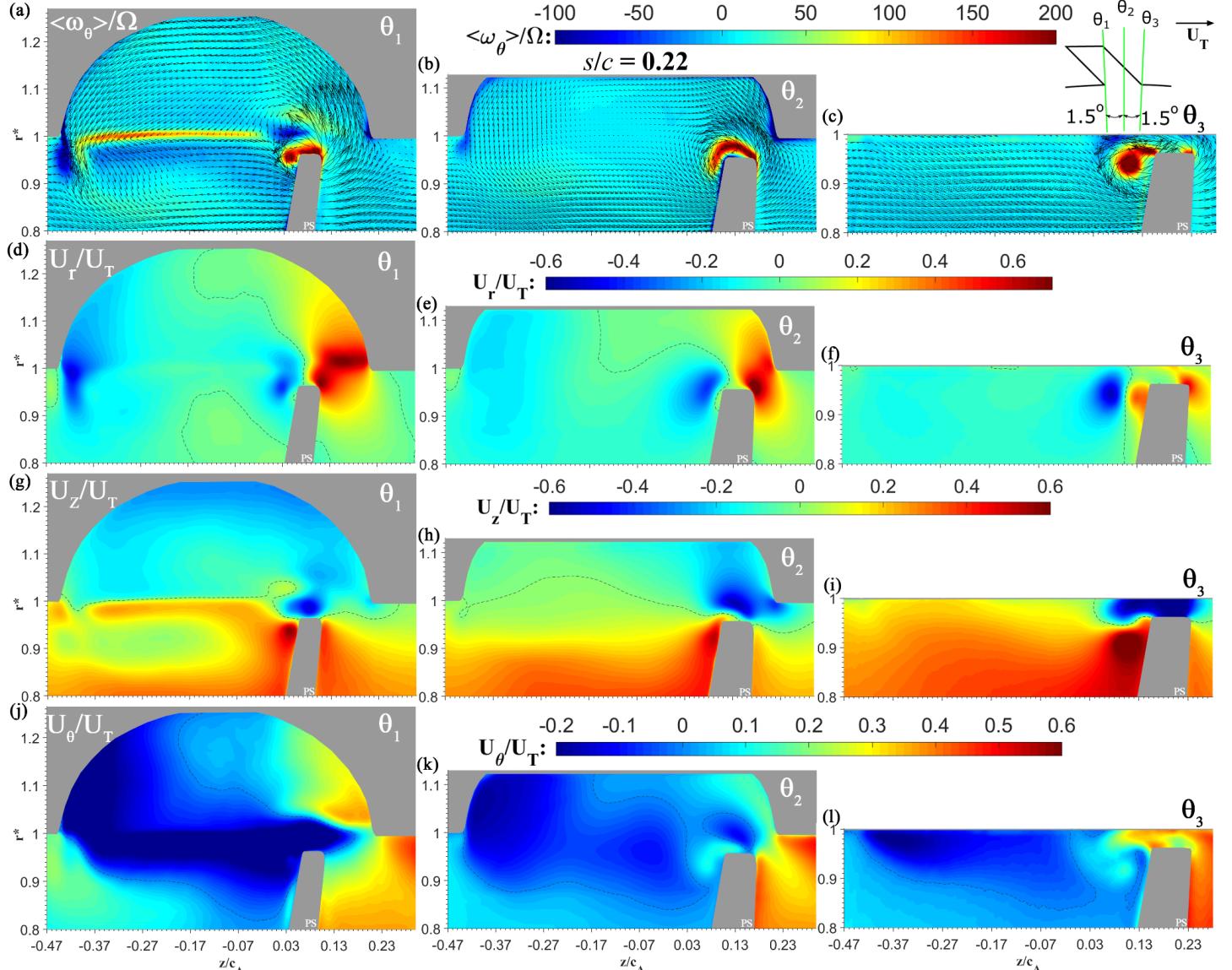


Figure 8: Samples of instantaneous vectors of in-plane velocity components and color contours of plane-normal vorticity in the center plane of the groove. (a)  $s/c = 0.22$ , when the blade PS is exposed to the groove, and (b)  $s/c = 0.44$ , when the SS is located downstream of the groove.

#### Flow characteristics in the meridional planes

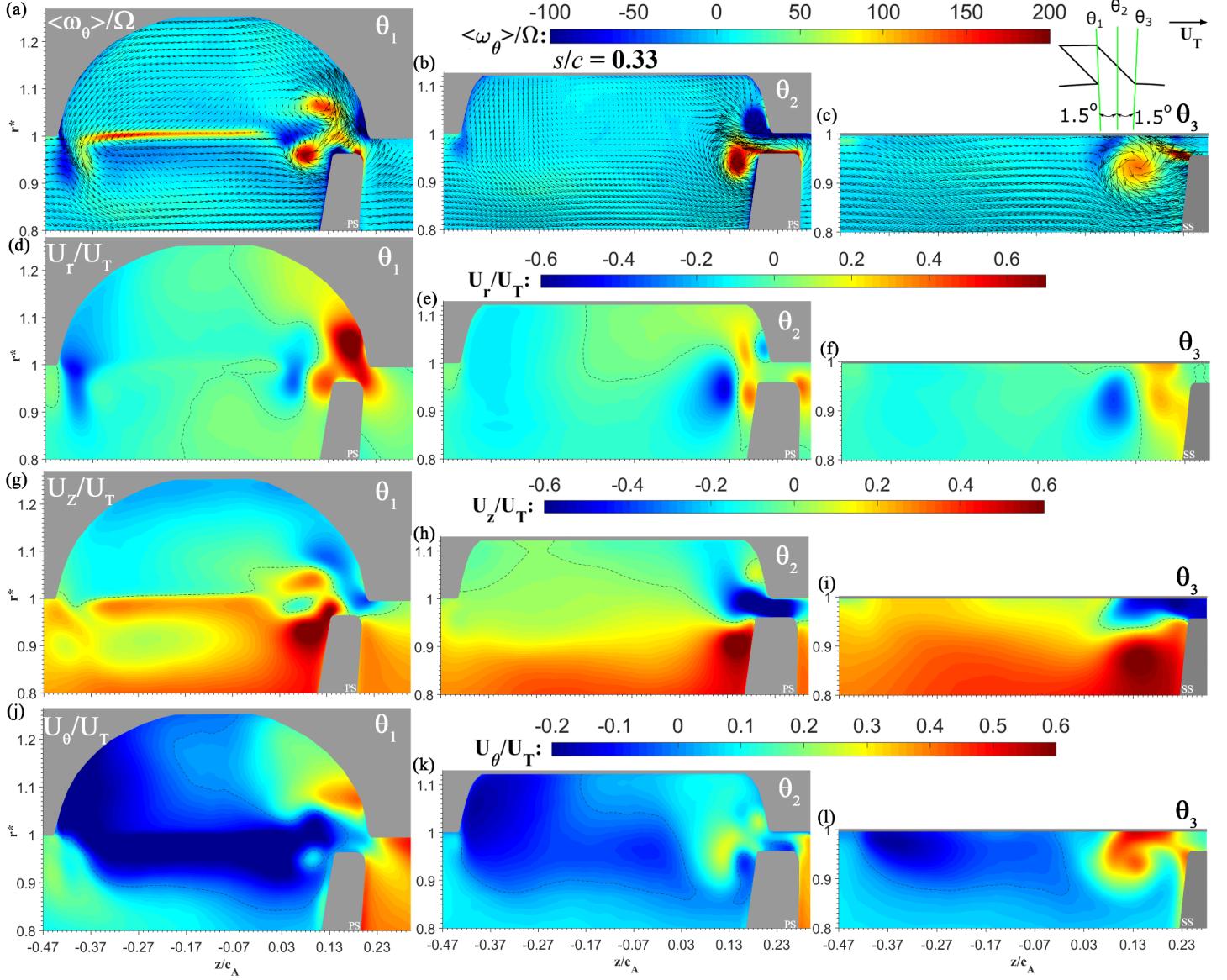
Figure 8 provides two samples of the instantaneous vectors of the in-plane velocity components and color contours of the plane-normal vorticity in the center plane of the groove for different  $s/c$ . In both cases, the flow circulates in the groove, entering from the downstream side and exiting from the upstream side, as already observed in prior numerical studies [10]. The time-resolved data recorded in this plane clearly shows that the flow always circulated in the groove in the same direction, but the velocity magnitude and flow direction around the downstream/entrance end vary substantially with blade location relative to the groove. At  $s/c = 0.22$  (Fig. 8a), when the blade PS is exposed to the downstream end of the groove (see

Fig. 6b), the inflow is dominated by a strong narrow jet originating from the PS. Conversely, at  $s/c=0.44$  (Fig. 8b), when the blade SS is already located downstream of the groove, the tip leakage flow seen on the bottom right corner of the plot is mostly axial. This axial flow persists for about 20% of the groove's length, and then turns into the groove as a diagonal jet while causing flow separation at the downstream corner. As discussed below, the positive vorticity at the upstream side of the jet is originated from the TLV, and the negative vorticity on the downstream boundary is originated from the enwall boundary layer. It is also evident that the peak vorticity in entrained vortical structures persistently decreases as the flow circulates in the slot. Furthermore, the organized vortical layers at the entrance are replaced with what appears to be "scrambled" turbulence at the exit.



**Figure 9:** Ensemble-averaged vorticity and velocity distributions in meridional planes  $\theta_1$  (left column),  $\theta_2$ , (middle column), and  $\theta_3$  (right column) when  $\theta_2$  intersects the blade at  $s/c=0.22$ . Top row:  $\langle \omega_\theta \rangle / \Omega$ ; second row:  $U_r / U_T$ ; third row:  $U_z / U_T$  and bottom row:  $U_\theta / U_T$ . Vectors are diluted by 5:1 in both directions for clarity in the top row. Dash lines indicate the zero values

Figures 9-11 display the distributions of ensemble-averaged circumferential vorticity and three velocity components for three different  $s/c$  selected to represent different phases of blade-groove interactions. Each figure displays data for three meridional planes intersecting the same groove at different location, as illustrated in the top right hand corner. Inherent to variations in their location relative to the groove, the flow structures displayed in them differ significantly. The following discussion highlights and analyzes some of the main features, but not all of them. Yet, we opt to present the entire sets for the selected  $s/c$  (data is available for other regions as well) to give the reader a complete picture on the blade-groove interactions. The datasets show that:

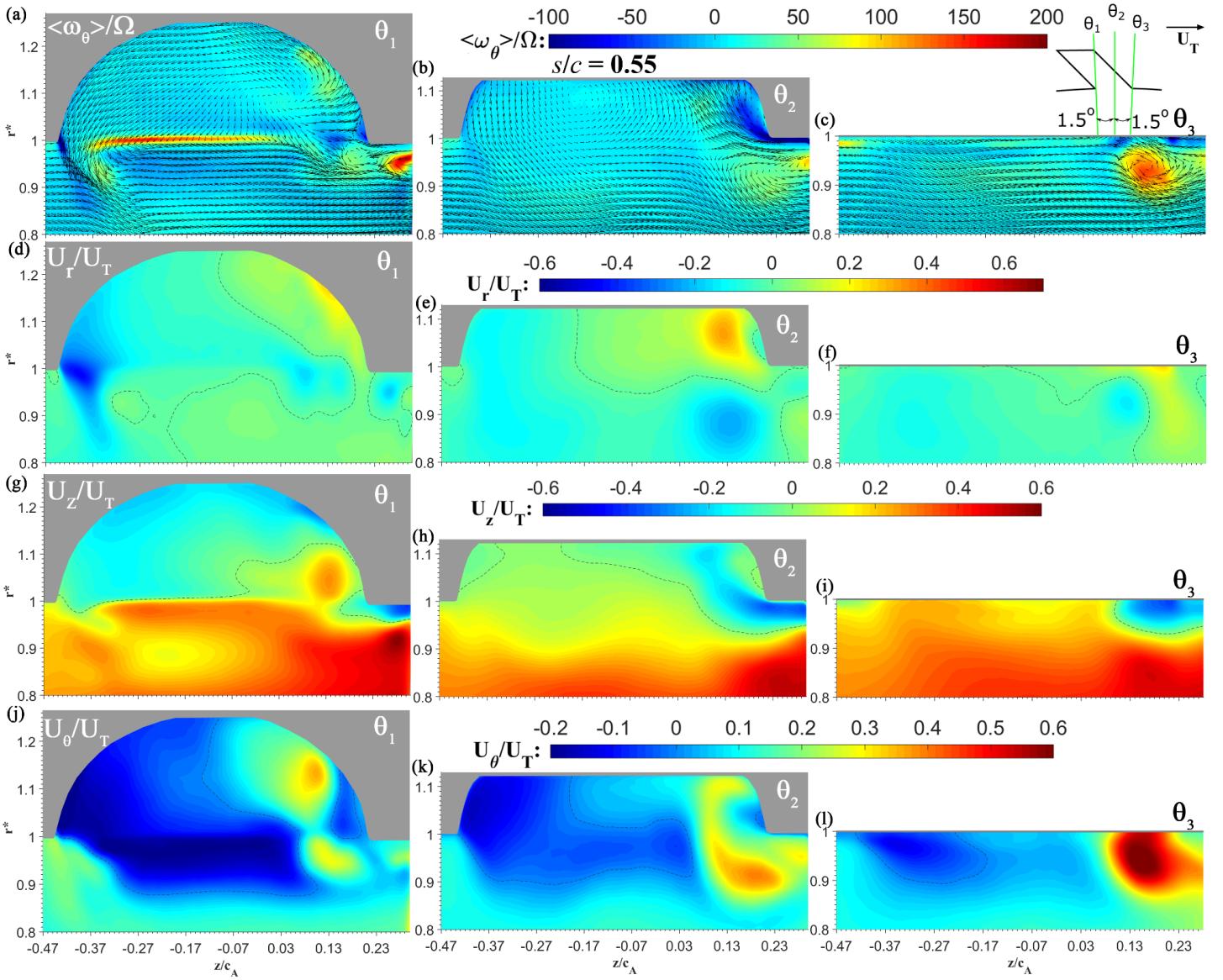


**Figure 10:** Ensemble-averaged vorticity and velocity distributions in meridional planes  $\theta_1$  (left column),  $\theta_2$ , (middle column), and  $\theta_3$  (right column) when  $\theta_2$  intersects the blade at  $s/c=0.33$ . Top row:  $\langle \omega_\theta \rangle / \Omega$ ; second row:  $U_r / U_T$ ; third row:  $U_z / U_T$  and bottom row:  $U_\theta / U_T$ .

**General Persistent Features:** The discussion starts with features that persist at all  $s/c$ . First, consistent with Fig. 6, the outflow from the upstream end of the groove peaks close to the  $\theta_1$  plane irrespective of blade orientation, with  $U_r$  reaching 40% of the tip speed. The corresponding peak negative values of  $U_\theta$  are systematically lower, i.e. the returning flow is not skewed at the same angle as the groove even before entering the passage. A much weaker outflow is also evident in all the  $\theta_2$  (middle) planes. As the radially inward jet penetrates the passage, it creates a region with  $U_z$  deficit bounded by a negative  $\langle \omega_\theta \rangle$  layer upstream. Second, at  $\theta_1$ , a narrow positive vorticity layer aligned along  $r^* \sim 1$  extends along the boundary of the groove, except for the upstream and downstream corners. Being located near the edge of the groove, this layer is originated from the

nearby endwall boundary layer, as the corresponding sharp transition in  $U_z$  confirms. Third, consistent with the skew angle, the inflow into the groove (downstream side) largely involves regions with positive radial and circumferential velocity components. However, the flow phenomena there are complex and phase-dependent due to the blade and TLV interactions with the groove discussed below.

**Phase Dependent Flow Features:** Figure 9 corresponds to  $s/c=0.22$ , the same orientation as Figs. 6b and 7a, providing multiple viewing angles on the same phase. Here, part of the blade PS, which includes the  $\theta_1$  and  $\theta_2$  planes, is exposed to the downstream corner of the groove, resulting in a high speed jetting inflow, with radial velocity well above 50% of the tip speed. The circumferential velocity is much smaller, indicating,



**Figure 11:** Ensemble-averaged vorticity and velocity distributions in meridional planes  $\theta_1$  (left column),  $\theta_2$ , (middle column), and  $\theta_3$  (right column) when  $\theta_2$  intersects the blade at  $s/c=0.55$ . Top row:  $\langle \omega_\theta \rangle / \Omega$ ; second row:  $U_r / U_T$ ; third row:  $U_z / U_T$  and bottom row:  $U_\theta / U_T$ .

like in the exit, that the inflow is not aligned with the groove skewness. In the  $\theta_1$  plane, a small TLV is already rolled up near the SS tip corner, but substantial fraction of the vorticity is concentrated above the blade tip. In the  $\theta_2$  plane, which corresponds to the origin of the second part of the TLV and where the vortex is “discontinuous”, the vorticity is mostly confined to the blade tip. At the end of the groove, in the  $\theta_3$  plane, the distinct vortex reappears. These observations suggest that the TLV discontinuity occurs as inflow separates in the PS corner, and jets straight into the groove, with substantial fraction of the vorticity originating from the PS boundary layer remaining above the blade tip (a comparison to the untreated

case follows). Also notable for this phase is the wide layer with negative  $U_\theta$  in the  $\theta_1$  plane, just below  $r^* \sim 1$ . As Fig. 6b demonstrates, this phenomenon is originated from the outflow from the next (lower) groove, and as discussed later, causes periodic variations in the relative flow angle around the blade LE.

At  $s/c=0.33$  (Fig. 10), which corresponds to Figs. 6c and 7b, the TLV is split in the  $\theta_1$  plane, as part of it is entrained into the groove by the narrow but fast radial jet ( $U_r / U_T > 0.6$ ) still originating from the PS. The abovementioned wide layer with  $U_\theta < 0$  at  $r^* \leq 1$  originating from the neighboring groove is evident in this plane as well. In the  $\theta_2$  plane, the PS is no longer

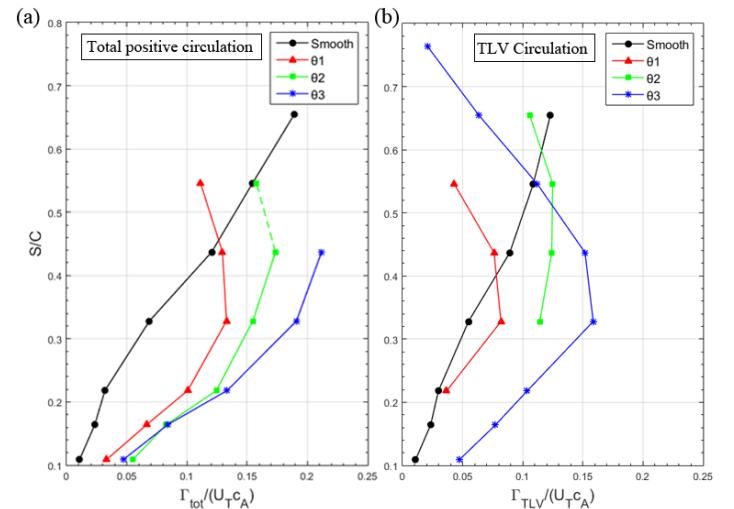
exposed to the groove, the high-speed tip leakage flow becomes predominantly axial with  $U_z$  exceeding 60% of the tip speed, and the radial flow into the groove diminishes. Flow separation occurs at the corner of the groove, near the SS tip, creating a pair of counter-rotating vortices, one associated/ coinciding with the TLV, and the other occupying the corner with the vorticity originating from the endwall boundary layer. In the  $\theta_3$  plane, a distinct TLV is already detached from the blade, and a shear layer with the same vorticity sign, seen also in all the untreated cases [28], connects it to the SS tip corner. The circumferential velocity around the vortex center is positive and higher than the surrounding area, i.e. the TLV behaves like a swirling jet. A negative vorticity layer extends from the endwall upstream of the TLV. A similar layer forms in untreated tip flows [28], where the vorticity is originating from the endwall boundary layer, above the blade tip, and separates from the wall at the point where the main passage flow meets the backward tip leakage flow where.

At  $s/c=0.55$  (Fig. 11), which corresponds to Fig. 6d, the blade SS is located just to the right of the present field of view. There are two vortices near the downstream/entrance corner of the groove in the  $\theta_1$  plane. The first is “latched” to the corner and the second is a small, concentrated, and recently rolled-up vortex located near the blade SS. In the  $\theta_2$  plane these two structures are already merged into a larger single vortex with low peak vorticity centered slightly upstream of the groove corner. The location of this vortex remnants just upstream of the corner is consistent with the cavitation images and the  $(z, \theta)$  plane data. The leakage flow with negative  $U_z$  along the endwall and the upper part of the vortex turns upward and becomes a diagonal jet at the entrance to the groove. It carries with it positive vorticity originating from the blade (or TLV) and negative vorticity originating from the endwall boundary layer. Yet, the negative axial velocity (Fig. 11h) and elevated circumferential velocity (Fig. 11k) in this area, the latter implying a reduced velocity relative to the blade, indicate that the TLV is still powerful enough to block part of the passage flow. As discussed before, in the untreated endwall, the radial gradients of circumferential velocity at the bottom the elevated  $U_\theta$  region are involved in the formation and propagation of BFVs at the onset of stall [29]. As proposed below, one of the effects of the axial casing grooves is to confine the area with elevated  $U_\theta$  to the immediate vicinity of the downstream corner of the grooves, preventing the formation of large BFVs that extend across the passage.

Finally, in the  $\theta_3$  plane the TLV becomes distinct again and its peak vorticity is higher than that measured in the middle of the groove (Fig. 11c). The circumferential velocity in this area is also very high (Fig. 11i), implying that the TLV behaves like a swirling jet. At higher  $s/c$  (not shown), as the flow begins to be influenced by the PS of the next blade, the peak vorticity and circumferential velocity in this plane also diminish. It appears that the last remnants of the vortex latched to the corner are entrained into the groove.

To show the impact of vorticity entrainment on the TLV strength, Fig. 12a show the evolution of the total positive

circulation on the SS of the blade, which is calculated by integrating the circumferential vorticity over areas where it is positive. The dashed lines correspond to the region where the blade is outside of the field of view, hence part of the total circulation is not accounted for in the integration. Figure 12b shows the TLV circulation obtained in a similar integration, but only over areas restricted to the TLV. The vortex boundary is defined as the points where the vorticity drops to 10% of its peak value. Trends are compared to those of the untreated endwall. As is evident and expected, in the untreated case, the TLV and total circulation increase monotonically with  $s/c$ . The treated endwall has a higher circulation than the untreated casing at low  $s/c$ , in accordance with the higher pressure rise (Fig. 4) at the same flow rate. However, in the treated endwall the circulation peaks at mid chord and then decreases to low values in the aft part. This trend confirms quantitatively that interaction with the groove reduces the TLV strength, presumably by entrainment of part of the TLV circulation as the SPIV data demonstrates. To explain the observed trends, one should note that the magnitude of circulation is affected by two main effects, namely shedding from the blade, which increases the circulation, and entrainment by the groove, which reduces it. At low  $s/c$ , newly shed circulation appears to compensate for the entrained fraction. However, at mid chord, the generation of new circulation weakens, as the minimal tip leakage cavitation confirms, and entrainment into the groove persists. Hence, the TLV circulation diminishes



**Figure 12:** Positive circulation (a) over the entire blade SS, and (b) in the TLV.

## DISCUSSION

The experimental data summarized in the previous sections provides intricate details on the effects of axial casing grooves on the flow structure in the tip region at  $\varphi=0.25$ , which corresponds to pre-stall conditions in the untreated case. This section focuses on certain features believed to play an important role in the suppression of stall and the associated substantial increase in pressure coefficient. The discussion is

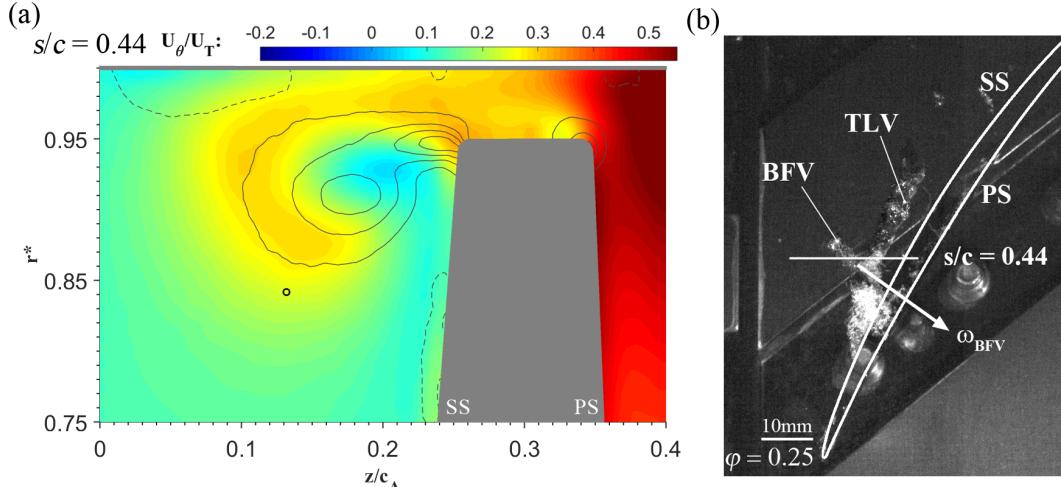
based on comparisons to the flow structure near the untreated endwall. The grooves periodically entrain parts of the TLV circulation, limiting the vortex size and strength, and confining its remnants to downstream corner of the groove. By the time the leading edge of next blade arrives, the signature of the vortex latched to the corner essentially vanishes.

It should be noted that the present measurements enable us to determine that vorticity shed from the blade up to  $s/c \sim 0.66$  is entrained into the groove. However, as the axial distance between SS of the blade and the groove increases, a TLV that is not entrained might rollup between the SS and the groove, outside of the present field of view. Such rollup occurs e.g. at high flow rates, as Fig. 5b shows. However, Fig. 5d demonstrates that at  $\varphi = 0.25$  there is essentially no cavitation in the tip gap at  $s/c > 0.5$ , suggesting that the leakage flow in the aft part of the passage is weak, consistent with the rotor operating at a low flow rate. Furthermore, the movies do not show a cavitating vortex downstream of the groove, suggesting that if shed vorticity does roll up at  $s/c > 0.66$ , the resulting vortex is weak. Hence, it is reasonable to assume that most of the shed circulation remains within the field of view, and sucked into the grooves before the next blade arrives. In contrast, the strength of the ensemble-averaged TLV in the untreated machine increases monotonically with  $s/c$  ([28] and Fig. 12), eventually exceeding that of the treated machine in spite of the lower pressure coefficient. This vortex is initially aligned at a shallow angle to the circumferential direction (Fig. 5c and [29]), but the concentrated multiple vortex filaments comprising its center “break up” in the middle of the passage and scatter over a broad area [24,29]. Once, the TLV becomes fragmented, the area occupied by the scattered filaments grows rapidly, eventually reaching the PS of the next blade, but only at mid chord. Hence, the onset of stall in the untreated rotor also does not involve direct interaction of the TLV with the vicinity of the leading edge of next blade.

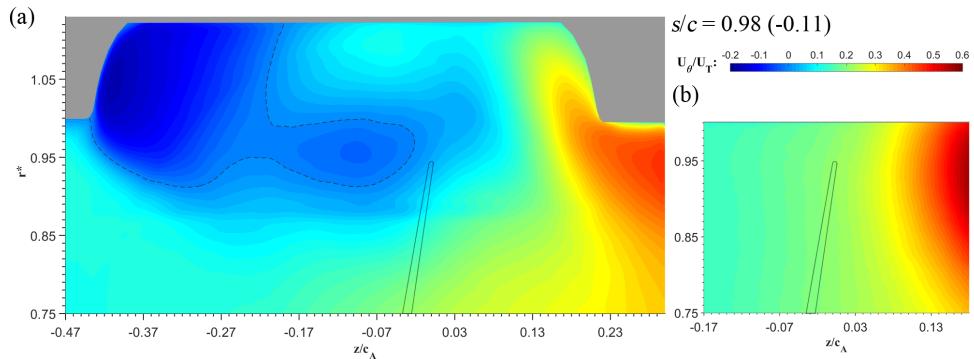
Instead, in Chen et al. [29], it is shown that the onset of stall in the untreated rotor involves formation of BFVs that extend diagonally upstream, from the SS mid-chord of one blade to the PS near the leading edge of the neighboring blade. They propagate to the next passage across the tip gap or around the blade LE. A sample image of a pre-stall BFV is shown in Fig. 5c. Much larger structures that appear when the rotor is stalled are displayed in [29]. The BFVs essentially disappear in the treated rotor, as the cavitation images and absence of their signature in the vorticity distributions under the TLV clearly indicate. To explain the reason for their disappearance, one must describe their origin briefly, summarizing the findings in [29]. As a visual aid, Fig. 13a shows the distribution of  $U_\theta$  in the untreated passage at  $s/c = 0.44$  and  $\varphi = 0.25$ . The layer of elevated  $U_\theta$  is originated from the tip gap (and the PS) and surrounds the TLV center. The 3D vorticity distribution, determined by recoding SPIV data in a series of closely spaced planes, indicates that the radial gradients in  $U_\theta$  under the TLV, e.g. at the point marked in Fig. 13a, are primary contributors to formation of a layer with elevated positive axial vorticity at this location. Similarly, a layer with negative radial vorticity is

associated with the axial gradients in  $U_\theta$  upstream of the TLV. Combined, they create a vortical “ring” originated from the tip gap that surrounds the TLV center. A similar ring has been observed around the TLV of an axial waterjet pump, so this phenomenon is not unique to the present machine [22]. Figure 13b is a sample image of a BFV located radially inward from the TLV during early phases of development. It also shows the *measured* direction of vorticity at the point marked in Fig. 13b. As is evident, the BFV is parallel to the measured vorticity, confirming its origin, and both are nearly perpendicular to the TLV and the SS surface. The BFV rolls up into a distinct vortex when the vorticity layer is perturbed. As the TLV migrates away from the blade with increasing  $s/c$ , especially after it breaks up, the region of elevated  $U_\theta$  expands to a substantial fraction of passage, eventually reaching the PS of the next blade. In fact, the high  $U_\theta$  on the PS of the endwall in Fig. 13a is associated with the TLV of the previous blade. As a result, the BFVs also expand intermittently to the PS of the next blade (Fig. 5c), and then propagate across the tip gap or around the LE to the SS of this blade. Upon penetration, the BFVs delays the rollup of a TLV, increases the flow angle around the LE, and trigger generation of similar vortices in the next passage. Under stall conditions, the frequency and size of BFVs increases, and they regularly expand upstream of the leading edge of the next blade.

Admittedly, as Figs. 10k and 11k clearly show, regions with radial gradients in  $U_\theta$  (and presumably axial vorticity) develop also under the TLV of the treated machine. Hence, BFVs could still form there. However, the grooves persistently limit the axial extent of this region, preventing the region with  $\partial U_\theta / \partial r > 0$  from reaching the leading edge of the next blade. As a demonstration, Fig. 14 compares the distributions of  $U_\theta$  at  $s/c = 0.11$ , a short distance ahead of the LE of the blade. Without the grooves (Fig. 14b), the elevated  $U_\theta$  region extends well upstream of the blade LE ( $z/c_A = 0$ ). Conversely, with the grooves,  $U_\theta$  is near zero around the LE (Fig. 14a), and is negative in other phases (Figs. 9k, 10k, and 11k). There are two possible reasons for this effect: first, the high positive  $U_\theta$  fluid is sucked periodically into the groove (Figs. 14a, and 11k). Second, the flow direction around the blade LE is influenced by the outflow from the grooves, where  $U_\theta < 0$ . In fact, all the  $(z, \theta)$  vector maps in Fig. 6 show the diagonal “jet” exiting from the upstream end of the groove, and reaching the plane of the blade leading edge. Hence, while BFVs could roll up periodically in the  $\partial U_\theta / \partial r > 0$  region, they could not propagate to the next passage. Interestingly, evidence of BFVs beginning to form can be seen in the cavitation images of the treated passage at  $\varphi = 0.35$  (Fig. 5b) under the TLV segment located downstream of the grooves. Finally, Wilke & Kau [10] argue based on RANS simulations that the delay of stall in a compressor with axial grooves is associated with weakening of the TLV. In a way, the present results agree with this observation since the formation and growth of the BFVs appear to be direct outcomes of the tip leakage flow, as well as rollup, migration, and breakup of the TLV.



**Figure 13:** (a) Color contours of the distribution of  $U_\theta$  at  $s/c=0.44$  and  $\varphi = 0.25$  in the untreated rotor. The black lines are contours of circumferential vorticity. (b) A sample image shown an early phase of BFV formation at  $s/c=0.44$ . The arrow shows the measured direction of ensemble averaged vorticity at the point indicated in (a).



**Figure 14:** Color contours comparing the distribution of  $U_\theta$  at  $\varphi = 0.25$  and  $s/c = -0.11$ , i.e. upstream of the blade LE, in the: (a) rotor with axial casing grooves, and (b) untreated rotor. Dashed lines indicate the location of the blade leading edge at  $s/c=0.0$ .

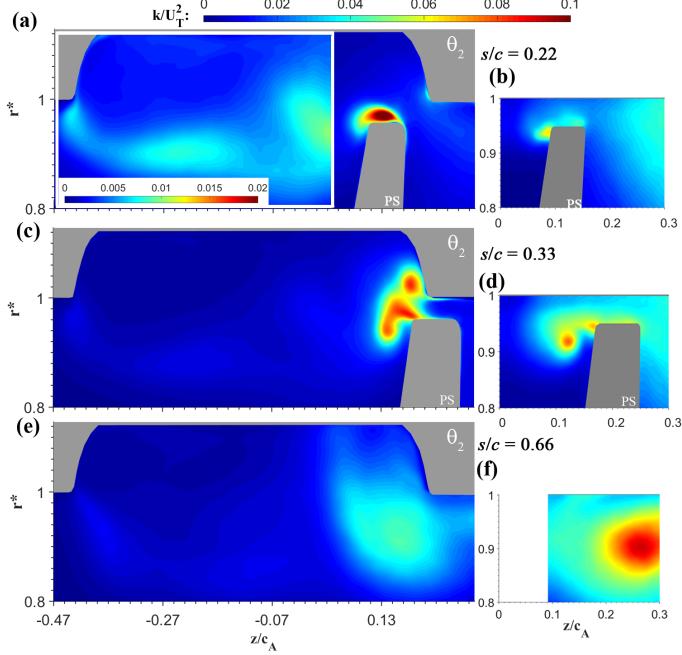
Comparisons between distributions of turbulent kinetic energy (TKE) could be used as statistical means of highlighting the effect on the flow stability of the TLV suction into the grooves in the treated rotor, and penetration of BFVs across the tip gap in the untreated machine. Figure 15 compares sample

distributions of TKE defined as  $k = 0.5(\overline{u_r u_r} + \overline{u_\theta u_\theta} + \overline{u_z u_z})$ , where  $u_i'$  is the velocity fluctuation. In the tip region of the grooved endwall, the high turbulence is confined to the vicinity of the TLV (Figs. 15a, c and e), and to the region of flow separation around the downstream corner of the groove (Fig. 15c). Once most of the TLV is sucked into the groove, at  $s/c=0.66$ , the turbulence around the TLV remnants (Fig. 15e) is much lower than that near the center of the fragmented TLV after it breaks up in the untreated machine (Fig. 15f). Deeper within the groove, the TKE is substantially lower. Elevated turbulence levels, which are significantly lower than those at the entrance to the groove, develop also when the outflow from the groove interacts with the passage flow, as shown using different contour scales in the insert in Fig. 15a. In other parts of the tip region, especially near the PS and entrance to the tip

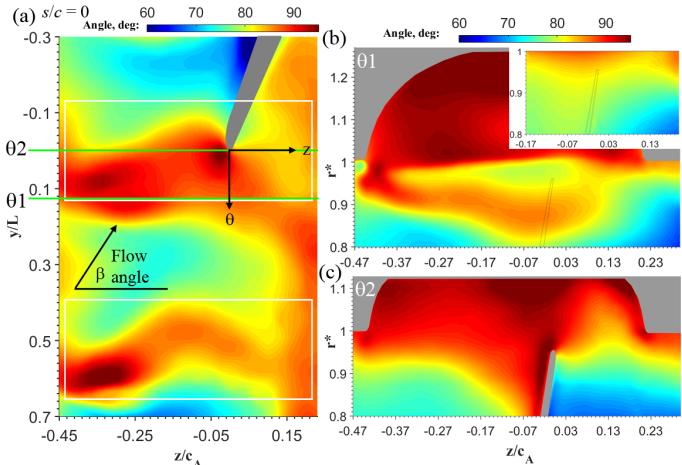
region, the TKE in the grooved machine is much lower than that in the smooth endwall. As demonstrated in [29], the high TKE near the PS and at the entrance to the untreated tip clearance (Figs. 15b and d) is dominated by the formation and penetration of the large BFVs across the tip gap. At higher flow rates, when the BFVs and fragmented TLV in the untreated machine do not make it to the PS of the neighboring blade, the turbulence at the entrance to tip gap is much lower.

The abovementioned periodic exposure to the outflow from the groove affects the flow direction near the leading edge of the blade, and presumably the blade loading. As illustrated in Fig. 16a, in the present discussion, the relative flow angle is defined as the angle between the  $z$  axis (positive axial direction) and the flow direction in the blade reference frame. Perpendicular views of the distribution of this angle are presented in Fig. 16. The insert in Fig. 16b also provides a reference distribution of this angle for a smooth endwall at the same blade orientation, with the dotted lines indicating the location of the LE. Figure 16a clearly shows that the outflow from the groove generates a layer with high flow angle, which propagates downstream and periodically increases the flow

angle around the blade leading edge. The corresponding  $\theta_1$  and  $\theta_2$  planes (Figs. 16b and c) show the significant radial extent of this phenomenon. Evidently, the outflow from the grooves periodically increases the relative flow angle upstream of the LE by more than  $5^\circ$ . This increase does not appear to cause leading edge separation, at least for the present flow conditions.



**Figure 15:** Comparison of TKE distributions with casing grooves (left column) and smooth casing (right column). (a and b)  $s/c=0.22$ , (c and d)  $s/c=0.33$ , and (e and f)  $s/c=0.66$ . Note the scale for the insert in (a) has a significant smaller range.



**Figure 16:** Distribution of relative flow angles in the rotor reference frame. (a)  $(z, \theta)$  plane at  $r^*=0.96$ , (b and c)  $\theta_1, \theta_2$  planes when the  $\theta_2$  plane is at  $s/c=0$ . Insert in (b) shows the relative flow angle in  $\theta_1$  for the untreated case.

## CONCLUSIONS

Performance tests, flow visualization and PIV measurements confirm that axial casing grooves suppress the onset of stall in the tip region of an axial fan rotor and elucidate the flow mechanisms involved. Following Müller et al. [7], the semi-circular grooves are skewed by  $45^\circ$  in the positive circumferential direction. There are four evenly spaced grooves in each rotor passage, with 33% of the groove overlapping with the rotor blade, and the rest extending upstream. Most of the present velocity measurements and flow visualizations are performed at a flow rate corresponding to pre-stall condition for the smooth endwall. The grooves reduce the stall flow rate by 40%, and cause substantial increase in pressure rise at low flow rates. Substantial improvements in stall margin are also reported in Müller et al. [7] for a transonic compressor. However, the grooves reduce the pressure rise at high flow rates. The reason for this performance degradation remains an open question that will be investigated in future experiments.

The velocity measurements show that a circulating flow enters the grooves at the downstream end and exits at the upstream edge, confirming prior results for axial casing grooves reported e.g. by Smith & Cumpsty [5], Wilke & Kau [10] and Müller et al. [7]. The present measurements also show that the flow structure over the entire tip region and within the grooves, including the primary inflow location, varies periodically with blade orientation. The periodic outflow agrees with measurements performed by Takata & Tsukuda [4]. The inflow peaks when the downstream end of the groove is aligned with the pressure side (PS) of the blade, and decreases, but does not vanish, when this end is located near the suction side (SS). Flow separation occurs at the groove corners as the flow enters and leaves the groove, creating a secondary flow there, and suggesting that the groove geometry could be optimized further. The radially and circumferentially negative outflow is concentrated at the upstream corner of the groove. When this flow propagates to the blade leading edge, it causes a periodic increase in flow angle by as much as  $5^\circ$ . Yet, this periodic increase does not cause leading edge separation. Although not measured, the increase in relative flow angle is expected to increase the blade loading. Hence, it might play a role in the 17.5% increase in pressure rise under the present flow conditions. Increase in relative flow angle caused by the outflow from the grooves is consistent with trends reported by Takata and Tsukuda [4], and Müller et al. [7]. However, the improved performance could also be associated with reduced TLV-induced blockage and/or secondary flows reported in Smith and Cumpsty [5], Crook et al., [8], and Müller et al. [7].

As the blade rotates past the grooves, the circulating flows entrain parts of the TLV, reducing its size and strength, as well as confining its remnants to the downstream corner of the groove. By the time the next blade arrives, the vortex signature essentially vanishes. Furthermore, the grooves prevent the formation of large-scale backflow vortices, which play a key role in the onset of rotating stall in the untreated rotor [29]. These vortices propagate diagonally upstream from the SS of one blade to the PS of the next blade and then to the next

passage. Comparisons between the untreated and treated flows reveal likely causes for BFV disappearance in the latter. The BFVs roll up in a region of high radial gradients in circumferential velocity under the TLV, and migrate to the next passage as the TLV grows and migrates away from the blade SS to the PS of the neighboring blade. By sucking the flow with high positive circumferential velocity and reinjecting it back in the negative direction, the grooves constrain the region where the BFVs can form, and prevent them from reaching the leading edge of the next blade. The distributions of turbulent kinetic energy provide statistical evidence that in contrast to the untreated casing, very little turbulence originating from a previous TLV, including the BFVs, propagates from the PS to the SS of the blade across the tip gap. Hence, the high TLV-related turbulence remains confined to the entrance to groove. Elevated, but lower turbulence is also generated as the outflow from the groove is injected into the passage.

## NOMENCLATURE

$c$	= rotor blade tip chord
$c_A$	= rotor blade tip axial chord
$h$	= width of the rotor blade tip gap
$H$	= rotor blade span
$k$	= turbulent kinetic energy
$L$	= nominal distance from the hub to the inner casing endwall
$p_{exit}$	= static pressure at stator outlet
$p_{in}$	= static pressure at IGV inlet
$r, z, \theta$	= radial, axial and circumferential coordinates
$r^*$	= normalized radial coordinate
$s$	= rotor blade chordwise coordinate
$u_r, u_z, u_\theta$	= radial, axial and circumferential velocity
$U_T$	= rotor blade tip speed
$u'$	= velocity fluctuation
$\rho$	= NaI solution density
$\varphi$	= flow coefficient
$\psi_{ss}$	= static-to-static pressure rise coefficient
$\omega_\theta$	= circumferential vorticity
$\Omega$	= rotor angular velocity
$\langle \rangle$	= ensemble averaged quantity

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