FLOW CHARACTERISTICS OF A HIGHLY ROTATING TURBINE CAVITY SYSTEM
WITH DISCHARGE HOLE

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ABSTRACT
An experimental investigation is performed to analyze the flow characteristics of a turbine cavity system containing discharge holes installed in a rotating disk. The turbine cavity system is composed of a rotating disk and two stationary disks on both sides of the rotating disk. The air flow is induced into the upstream cavity, and then discharged into the downstream cavity through 8 discharge holes in the rotating disk. The flow field in each cavity at high-speed rotation of the rotor was measured by a three-dimensional LDV system. The measured flow field is analyzed to understand the flow structures, and further provide information for studying the heat transfer behaviors of the turbine disk system. The overall flow field in the upstream cavity shows a negligible axial velocity with a relatively small rotational velocity, less than 10% of the rotor speed. The downstream cavity flow has a high rotational velocity close to the rotational speed of the discharged jets, due to the direct circumferential momentum transfer from the discharged jets. The interaction between the discharged jet and the downstream stator disk induces an asymmetric development of the spreading wall jet, which results in a relative circumferential motion to the revolving discharged jet. The whole flow field in the downstream cavity is divided into several flow regions according to their features.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>Re_0</td>
<td>Rotational Reynolds number ( = ( \frac{\omega R_h^2}{v} ))</td>
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<tr>
<td>r</td>
<td>radial coordinate</td>
</tr>
<tr>
<td>r_c</td>
<td>inlet corner radius of discharge hole</td>
</tr>
<tr>
<td>S</td>
<td>standard deviation for the measured velocity</td>
</tr>
<tr>
<td>s</td>
<td>gap distance between rotor and stator disks</td>
</tr>
<tr>
<td>( \bar{v} )</td>
<td>phase averaged velocity</td>
</tr>
<tr>
<td>( \ddot{v} )</td>
<td>phase averaged value of root-mean-squared fluctuation velocity</td>
</tr>
<tr>
<td>( \bar{\bar{v}} )</td>
<td>root-mean-squared fluctuation velocity ( = ( \sqrt{\ddot{v}} ))</td>
</tr>
<tr>
<td>x</td>
<td>axial location from the exit of discharge hole in the discharged flow direction</td>
</tr>
<tr>
<td>xx</td>
<td>axial location from the inlet of discharge hole in the direction opposite to the discharged flow discharged flow direction</td>
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<tr>
<td>z</td>
<td>confidence coefficient</td>
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Greek Symbols

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<tr>
<td>( \Delta )</td>
<td>error or change in quantity that follows</td>
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<tr>
<td>( \phi )</td>
<td>circumferential coordinate</td>
</tr>
<tr>
<td>( \theta )</td>
<td>LDV probe angle</td>
</tr>
<tr>
<td>( \nu )</td>
<td>kinematic viscosity</td>
</tr>
<tr>
<td>( \omega )</td>
<td>rotational speed of rotor</td>
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Subscript

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<tr>
<td>i</td>
<td>data number index</td>
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<tr>
<td>p,1</td>
<td>LDV probe 1</td>
</tr>
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<td>p,2</td>
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<td>s</td>
<td>axial</td>
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<tr>
<td>\phi</td>
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INTRODUCTION

Recent advances in the design technology of gas turbines has led to a complicated internal structure due to the demand for a more compact and light system. The internal air system of a gas turbine is thus receiving more attention.

Internal air is defined as the flow which does not contribute directly to the turbine power but performs various functions for the safe and more efficient operation of the gas turbine. Among these functions, several important ones are as follows: Cooling of hot components, prevention of hot gas ingestion into the turbine disk cavities, disk cooling, control of turbine blade tip and seal clearances, and control of bearing axial loads.

Cooling air flow for the hot components such as combustors, turbine blades and guide vanes is supplied by the internal air flow. The total amount of internal air reaches up to more than 20% of the main compressor flow. Thus, the proper estimation of the supply characteristics in each component is essential to a balanced supply of air flow while minimizing the overall amount of air flow. Furthermore, flow analysis of the turbine cavity system is required to prevent the ingestion of hot gas into the cavity, controlling the air flow inside the cavity in proper temperature and pressure level. The internal air flow in the turbine disk cavity performs another important role for the removal of the thermal load on turbine disks. The tip clearance between blade and shroud as well as the other seal clearances are under control by prevention of the excessive thermal expansion from the design condition.

Many studies have been performed for the balanced supply of the internal air flow and the reduction in the thermal load on the turbine disk system. The turbine cavity system comes in many different shapes and configurations, with quite complex geometry in some cases. Numerous investigations have been, thus, performed for various different models. Dijkstra and van Heijst (1983) measured the flow field of a rotating disk system enclosed by a cylindrical shroud. Owen et al. (1985) studied the rotor-stator system with a radial mass flux. They analyzed the overall flow structure, by estimating the source and the sink flow intensities deduced from the theoretical pumping flow rate of a rotating free disk. Farthing et al. (1992) studied the flow structure of a rotating cavity system with an axial throughflow. They found a powerful toroidal vortex inside cavities with large gap ratios and weak counter rotating toroidal vortices for cavities with small gap ratios. Bhavnani et al. (1992) presented an experimental result of fluid flow in an unshrouded plane rotor-stator disk system, and reported the minimum coolant flow rates for the prevention of ingress, determined for the case of a simple axial rim seal, as a function of the rotational Reynolds number. Owen and Bilimoria (1977) observed the flow structure in a rotating cylindrical cavity with an axial influx and a radial outflux. According to the rotational speed, they divided the flow structure change into three stages; a core-dominated regime, a developing Ekman-layer regime and a fully-developed Ekman-layer regime. These changes are considered to severely influence the heat transfer behavior on the disk wall. Kim and Metzger (1994) experimentally investigated the heat transfer behavior of a downstream disk influenced by 19 concentric cooling jets located on the upstream rotor. They provided a comparison of local heat transfer distribution obtained with a single center-supply of disk coolant. Wittig et al. (1994) presented the experimental and numerical flow field around the orifices in a rotating disk. In their experiment, the flow in a rotating cavity, which is located at the upstream of the orifices, is discharged into the free space.

A rotating turbine cavity system containing flow passages on a rotor disk, is frequently adopted in the internal air system of a gas turbine. In this study, the flow characteristics of a turbine cavity system which consists of the enclosed upstream and downstream cavities with the rotating discharge holes will be investigated through flow measurement. The dimensions, the geometry and the operating conditions are similar to those of a real turbine. The flow field inside the cavity is measured by Laser Doppler Velocimetry. The measured flow field will be analyzed to construct a model of the flow structure. This will provide basic physical insight into the comprehension of the heat transfer phenomenon in a rotating turbine disk system as well as for the supply characteristics of internal air systems.

EXPERIMENTAL APPARATUS AND PROCEDURE

Test Rig

The test rig has been constructed at the Institut für Thermische Strömungs- maschinen in Karlsruhe University, Germany. Figure 1 depicts the test rig schematically. A rotor disk where eight discharge holes are installed and two stator disks on each side of the rotor compose a turbine disk system. A turbulence mesh is placed to remove any large scale motions from the settling chamber. Air flow comes into the upstream cavity through the annular slot between the stator disk and the shaft, and then air flow is discharged into the downstream cavity through the rotating discharge holes. The exit to the ambient air is positioned in the downstream stator disk. The geometric parameters used in the experiment are listed with their dimensions in Table I. The gap distances of both upstream and downstream cavities are kept identical in this study.

A 85.3 kW compressor supplies air flow at a maximum mass flow rate of 500 g/s. The compressed air passes through an oil filter, a cooler, and the orifice system. The air then passes through a heater into a 1.5 m settling chamber before being supplied to the test rig.

The rotor disk is driven by a 23 kW DC-motor, which is connected to the drive plug of the shaft by a flexible clutch. Due to the rigorous requirements for high-speed rotations, high precision bearings are used to achieve a maximum allowable rotation speed of 10,000 rpm. To ensure robust sealing at high speed, a labyrinth seal is adopted to seal the gap between the rotor and the housing. The gap between the sealing fins and the housing for this experiment is about 0.2 mm. Thermocouples and pressure taps are installed inside of the upstream and downstream cavities to identify inlet and exit conditions of the discharge hole. Thermocouples of NiCr-Ni type are used, whose junctions are welded to a sensing diameter of 0.5 mm.

LDV System

The test rig operates at high rotational speed and high pressure level, and includes relatively small discharge holes compared to
the disk space. These features render the downstream flow field extremely complex and require a high spatial resolution of the measurement. In addition, the focus of the incident beams from the probes is located inside of the rotating system. Consequently, a stationary measuring volume is required to cover all 360° of the circumferential positions that simultaneously contain strong jets and recirculation zones. The measurement system is thus extremely sensitive to the adjustment of each component of the experimental set-up. The experimental set-up for the LDV measurement is schematically depicted in Fig. 2. Differential doppler technique for heterodyning the scattered light is used.

Measurements are performed with a three component LDV system. A 10 W Argon-ion laser (Coherent INNOVA 90) is used in multi-mode operation as a light source. The multi-wavelength beam passes the transmitter box (Dantec), in which it is first split into two beams, one of which passes a bragg cell to be shifted with a frequency of 40 MHz for the detection of the flow direction. Both of these beams are subsequently separated into different colors for the three components (476.5 nm, 488 nm and 514.5 nm) and linked to two fiber probes (Dantec, 60x65; 60x67). The focal lengths of the two probes are both 400 mm. Backward scattering mode is adopted, so the scattered light is collected by identical probes and then transmitted into the PM-tube (Dantec, 9057x0081). Three channels of the Burst Spectrum Analyzer (Dantec, 57N20, 57N35) are then used to process the multiplied signals simultaneously.

**Flow seeding**

A commercial aerosol generator (Pallas AGF5.0) is used to seed the flow with atomized particles of DEHS solution. Physical properties of DEHS (Sebacic acid di (2-ethylhexyl)-ester) are shown in Table 2. The aerosol generator with DEHS can generate up to $5.1 \times 10^{14}$ particles/s, which is equivalent to a maximum concentration rate of $8.7 \times 10^{10}$ cm$^{-3}$. The mean diameter of the generated particles ranges from 0.21 to 0.27 μm.

Feller and Meyers (1975) proposed a model of the particle's response to a step change in velocity field. According to their model, the characteristic frequency of the particle response in the present study is high enough to follow the dynamic characteristics of the flow motion by virtue of small size and high viscosity of the particle.

Generated particles are divided into 4 streams to be injected into the settling chamber to seed the flow uniformly.

**Access of the measuring volume**

In this study, a two-dimensional velocity field was measured for the upstream cavity while a three-dimensional velocity field was measured for the downstream cavity. To obtain the optical paths, two windows are installed. One is located at the side of the housing while the other is located at the front area of the downstream stator as shown in Fig. 2. Due to the limited size of the window and the obstacles such as the drive shaft, the belt, the motor and the supporter of the test rig, the probe location is severely restricted. The measurable area is therefore limited and the two probes for three-dimensional measurements cannot be positioned at a right angle.

The focal points of the two probes should be overlapped on a point, but precise alignment is required due to the extremely small size of each measuring volume, the diameters of which are 0.189, 0.116, 0.116 mm, respectively. Thus, a probe mount is specially devised with 6 degrees of freedom in its movement. For each measurement, the overlapping of the measuring volumes is checked.
Table 2 Physical properties of the flow seeding material (DEHS)

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
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</thead>
<tbody>
<tr>
<td>Name</td>
<td>Sebacic acid di-(2-ethylhexyl)-ester</td>
</tr>
<tr>
<td>Density (kg/m$^3$)</td>
<td>912</td>
</tr>
<tr>
<td>Dynamic viscosity (Pa·s)</td>
<td>$23 \times 10^{-3}$</td>
</tr>
<tr>
<td>Melting point (k)</td>
<td>225</td>
</tr>
<tr>
<td>Boiling point (k)</td>
<td>529</td>
</tr>
<tr>
<td>Vapor pressure at 293k (Pa)</td>
<td>$1.9 \times 10^4$</td>
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</table>

by optimizing the coincidence level of the output signals from the three components.

A mirror is installed to enlarge the accessible area and to increase the probe angle. A small probe angle allows stronger light scattering from the rotor disk surface, which may cause a severe noise signal when the light is incident on the receiving optics. The angles $\theta_{\mu,1}$ and $\theta_{\mu,2}$ are set to 19.9$^\circ$ and -16.7$^\circ$, respectively. The probe configuration is schematically shown in Fig. 3.

In order to maintain the relative positions between the two probes and the mirror, these are mounted on one arm of a three-dimensional traverse unit with a step distance of 25 $\mu$m.

Data processing

Three burst spectrum analyzers (BSA) are applied to process the LDV output signal. Only the dominant signals are accepted through a validation and filtering process. Due to the non-orthogonal configuration of the two probes, the radial and the axial components are calculated from the two measured velocity vectors in the probe plane. The transformation relation from the measured components to the orthogonal components is expressed as

$$
V_{r} = \frac{1}{\sin(\theta_{\mu,1} - \theta_{\mu,2})} \left[ \cos \theta_{\mu,1} - \cos \theta_{\mu,2} \right] V_{r,1} + \left[ \sin \theta_{\mu,1} - \sin \theta_{\mu,2} \right] V_{r,2}
$$

The circumferential velocity component is directly determined from probe 1.

In a rotating system, the determination of the relative angular position is required for a discrete burst signal of the velocity. A rotary encoder, which generates 4096 pulses and 1 reset signal per revolution, is installed to obtain the information on angular position. These signals are connected to the signal processors. The encoder pulse signal is used as the input clock reference. All discrete velocity burst data from the continuous doppler signal are handled with this clock reference which is later converted into the relative angular positions with the reset signal indicating one revolution. The statistical properties of the flow such as mean velocity and turbulent velocity are ensemble averaged according to the angular position. Width of the angular segment which is represented by a discrete angular value is specified considering the spatial resolution and convergency.

For the rotational position, $\phi$, of the specific radial and axial measurement position, the phase averaged mean velocity component is defined as

$$
\bar{V}(\phi) = \frac{\sum_{i=1}^{N(\phi)} V(\phi)_i}{N(\phi)}
$$

where $N(\phi)$ denotes the number of burst data within the angular segment of $\phi$. Similarly, the phase averaged fluctuation velocity component is defined as follows.

$$
\tilde{V}(\phi) = \frac{\sum_{i=1}^{N(\phi)} \left( V(\phi)_i - \bar{V}(\phi) \right)}{N(\phi) - 1}
$$

The spatial variation of the velocity field across an angular segment may result in error of the phase averaged fluctuation velocity component. An algorithm which approximates the mean velocity profile within a spatial segment as a linear profile is adopted to reduce errors of this kind. A precise procedure for the correction is described in detail in Jakoby et al. (1994).

Statistical uncertainties in the measured velocity field are estimated using the procedure described by Snyder et al. (1984). The uncertainty level is determined from the following equation:

$$
\Delta V = \frac{z \cdot S_V}{\sqrt{N}}
$$

where $S_V$ is an estimate of the standard deviation for the measured velocity and $N$ is the number of sampled data. The value of $z$ is 1.96 for a 95% confidence level. The uncertainty level of the
circumferential velocity component is directly obtained from Eqn. (4), while those of the axial and the radial components are determined from the standard deviations of $S_x$ and $S_y$, which are calculated by the following equation:

$$
\Sigma_{S_x} = \frac{1}{\sin^2(\theta_x - \theta_{xx})} \left[ \cos^2 \theta_x \sin^2 \theta_{xx} - \cos \theta_x \cos \theta_{xx} \right] \Sigma_{S_x}
$$

The above equation is deduced from Eqn. (1) and the statistical relations described by Benedict (1984).

In this study, a large number of data (53,000) are collected for the whole rotational angle in a given axial and radial location, while the numbers of the sampled data and the standard deviations are changed from point to point. The uncertainty level, thus, varies depending upon the location. The high level of uncertainty are normally found near the axial jet and the stator wall region which have the high characteristic velocity.

The uncertainty values for each component of the axial, radial, and the circumferential velocities are estimated to be less than 2.5 m/s, 1.3 m/s and 1.5 m/s, respectively, over the whole measured area. It is found that the high uncertainty level of the axial velocity occurs at the strong axial jet regions and the high level of the circumferential velocity occurs at the jet interaction regions. Thus, it is estimated that the characteristic scales of the high level uncertainties are large. Generally, the uncertainty level of the small velocity scale region is low.

RESULT AND DISCUSSION

A two-dimensional flow field with axial and circumferential velocity components for the upstream cavity and a three-dimensional flow field including the radial velocity component for the downstream cavity are measured. The pressure ratio between upstream and downstream cavities is 1.05 while the pressure level in the downstream cavity is 2 bar. The measured flow rate through a discharge hole is about 2.8 g/s. Accordingly, the average velocity at the exit of the discharge hole is estimated to be about 97 m/s. The detailed measurement procedure of the flow rate is well described in Maeng et al. (1998). The rotational speed of the rotor disk is fixed at 6,000 rpm which corresponds to a rotational Reynolds number, $Re_\infty$, of 2.0 x $10^6$. This value far exceeds the turbulence transition condition of around 3.0 x $10^5$ for a rotating free disk. Meanwhile, the flow rate coefficient, $C_w$, is 18300.

The coordinate system and disk configuration is depicted in Fig. 4. $x$, $r$, and $\phi$ are axial, radial and circumferential coordinates, respectively, which are fixed at the rotor disk rotating in the clockwise direction viewed from the downstream face. The origin of $x$ and $xx$ coordinates are located on the disk surfaces facing the downstream and upstream cavities, respectively.

**Upstream cavity flow characteristics**

The overall circumferential velocity field of the upstream cavity is shown in Fig. 5. An $xx/s$ value of 0 represents the rotor wall surface while an $xx/s$ value of 1 represents the upstream stator surface in the upstream cavity. The circumferential velocity, $v_c$, is presented in absolute velocity not the relative one to the rotating reference frame. Flow measurement near the rotor surface in the upstream cavity cannot be made because of difficulty in probing. As shown in Fig. 5, the circumferential velocity of the whole upstream cavity is determined by the radial position irrespective of the circumferential position, an important feature in the circumferential velocity of the cavity core region, respectively. The comparison of the result with a typical rotor-stator system, however, reveals that the circumferential velocity component is directly obtained from Eqn. (4), while those of the axial and the radial components are determined from the standard deviations of $S_x$ and $S_y$, which are calculated by the following equation:

$$
\Sigma_{S_x} = \frac{1}{\sin^2(\theta_x - \theta_{xx})} \left[ \cos^2 \theta_x \sin^2 \theta_{xx} - \cos \theta_x \cos \theta_{xx} \right] \Sigma_{S_x}
$$

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**Upstream cavity flow characteristics**

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Fig. 5 Absolute circumferential velocity contour of upstream cavity flow

The overall axial velocity field of the upstream cavity is shown in Fig. 6. The axial velocity, \( \dot{V}_x \), is very small over the whole measurement region. The average axial velocity calculated through the measured flow rate of the annular slot is about 10 m/s, but the inner and outer radii of the annular slot are respectively 0.206 \( r/R_D \) and 0.25 \( r/R_D \), which are out of the measurement range of this study. The behavior of the annular slot jet is thus not shown in Fig. 6. Most of the flow in the cavity does not contain an axial velocity component. A weak leakage flow from the gap between the rotor disk and the housing is detected in the outermost region. A weak reverse flow in the axial direction is also observed near the discharge hole.

**Downstream cavity flow characteristics**

The velocity field in the downstream cavity is measured at five planes parallel to the rotor disk within the accessible radial range of the measuring volume. The axial distance from the hole exit is normalized by the cavity length, \( s \). An \( x/s \) of 0 represents the location of the rotor disk surface in the downstream cavity while a \( x/s \) of 1 represents the downstream stator disk surface. A reference frame rotating at the speed of the rotor disk is adopted for the convenience of the data analysis.

Figures 7 - 11 show three-dimensional velocity fields on each measurement plane. Since the flow field is periodic in the circumferential direction with eight jets discharged into the cavity, a region containing only two jets is presented. In these figures, the circumferential velocity is relative to the rotating reference frame. Figure 7 - 11 (a) illustrate the circumferential velocity, \( \dot{V}_\theta \), and radial velocity, \( \dot{V}_r \), vectors while the axial velocity, \( \dot{V}_z \), distribution is depicted in figure 7 - 11 (b). The relative positions of the holes on the rotor and the rotational direction of the rotor disk are also shown for the plots.

It is shown that the jets maintain high axial velocity up to the region near the stator. The tangential velocity at the center of the discharge hole is 66 m/s when the rotational speed of the rotor disk is 6,000 rpm. Despite the large tangential velocity, the position of the jet at each cross section does not deviate much from its initial discharge position as shown in the figures. The overall cavity flow indeed corotates with the rotor disk at the rotational speed corresponding to the tangential velocity of discharge hole. This behavior is shown regardless of the radial location, except for the small region directly affected by the jet interaction with the stator wall. The difference of the rotor-stator cavity system with discharged jets from the typical rotor-stator system without discharged jets can be stated as follows. For a cavity system without jet discharge, the circumferential flow is induced by the shear transfer from the wall, while for a system with discharged jets the circumferential flow is induced by the circumferential momentum supplied by the discharged jets. The circumferential momentum is transferred to the jet flow inside the long discharge holes in the rotor. Thus, the circumferential velocity in the cavity is determined mainly by the rotational speed of the discharge hole.
Fig. 7 Three-dimensional velocity field at $x/s = 0.24$

(a) circumferential and radial velocity vector
(b) axial velocity contour

Fig. 8 Three-dimensional velocity field at $x/s = 0.40$

(a) circumferential and radial velocity vector
(b) axial velocity contour
Fig. 9 Three-dimensional velocity field at $x/s = 0.56$

Fig. 10 Three-dimensional velocity field at $x/s = 0.76$
The phase averaged root-mean-squared velocity fluctuation of the circumferential component, $\tilde{V}_r$, is shown to present the turbulence fluctuation velocity field in Fig. 12 ~ 13. The axial reverse flow caused by the interaction of wall jets from two consecutive discharge jets can be observed more clearly from the fluctuation velocity field. In Fig. 12, a region of high turbulence can be observed at the boundary of the impinging jet where the interaction with the cavity flow is taking place. In Fig. 13, however, another interaction region of high turbulence can be observed ahead of the impinged jet in the rotational direction of the rotor. The flow is similar in shape to the horseshoe vortex in a crossflow passing through an obstacle on a flat plate, and is developed on both sides of the impinged jet. The interaction flow is advected in the opposite direction of the rotor rotation due to the shear from the stator. Of the two, the flow at the outer location in the radial direction is seen to be stronger, extending to the cavity core region.

To observe the flow behavior near the stator wall, which is out of the measurement range, a flow visualization on the stator wall was performed by the oil film method. The visualization using black spray paint and oil mixture is shown in Fig. 14. The streakline averaged in the circumferential direction is shown with the trajectory of the impinged jets. A typical streakline of an Ekman layer is observed, in which a radial motion induced by the radial pressure gradient is shown along with the rotational motion. The flow in the Ekman type layer near the stator wall has a large velocity gradient with the radial velocity component directed towards the center of the cavity. The radial flow in the Ekman type layer, thus, interacts with the spreading wall jet, generating vortical flow around the boundary of the impinged jet. The vortical flow becomes stronger especially at the outer radial edge of the impinged jet due to the opposite radial flow direction of the spreading wall jet. The vortical flow is finally advected radially towards the center of the cavity by the Ekman layer flow, as can be seen in Fig. 13.

The flow structure of the downstream cavity characterized by the jets discharged through the holes in the rotor is schematically shown in Fig. 15. The flow field can be divided into the following regions based on the characteristics of each region.

1. Jet region
2. Crossflow region
3. Impingement region
4. Stator wall region dominated by the Ekman type layer
5. Spreading jet region dominated by the wall jet
6. Interaction region of the two branches of the spreading wall jets
7. Vortical motion region containing reverse flow

**SUMMARY AND CONCLUSIONS**

Flow characteristics of a rotating turbine cavity system with discharge holes were investigated. Flow fields inside both upstream and downstream cavities of the discharge holes were measured with a three-dimensional LDV system. The rotor disk containing the discharge holes rotates at a high rotation speed of...
Fig. 11 Three-dimensional velocity field at $x/s = 0.92$

Fig. 12 Circumferential fluctuation velocity field at $x/s = 0.40$

Fig. 13 Circumferential fluctuation velocity field at $x/s = 0.92$
6,000 rpm, while the pressure ratio between upstream and downstream cavities is set to 1.05. Some important observations are noted and summarized below.

1. The overall flow field of the upstream cavity has a relatively small rotational velocity component of about 10% of the rotor speed with negligible axial velocity component.

2. Rotational flow in the downstream cavity is dominated by the rotational speed of the discharged jets, and the strong jets reach the downstream stator with little deflection from their original angular locations in the rotating reference frame.

3. The three-dimensional velocity field in the downstream cavity containing strong interactions between the axial jets and the stator can be divided into a jet region, cross flow region, jet-impingement region, stator wall region, spreading jet region, spreading wall jets interaction region, and axial reverse flow region.

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Daily, J. W., Ernst, W. D., and Asbedian, V. V., 1964, "Enclosed Rotating Discs with Superimposed Throughflow," Report No. 64, Dept. of Civil Engineering, Hydrodynamics of Laboratories, Massachusetts Institute of Technology.


