PROGRESS OF FILM COOLING IN INDUSTRIAL GAS TURBINE VANES AND BLADES

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ABSTRACT

Cooling air can be chilled for industrial gas turbines. By taking advantage of this feature, a deep digging fan-shaped film cooling hole equipped the function of a trench has been devised. The characteristics of this film cooling hole were investigated using quantitative measurement methods. It was revealed that this film cooling hole had high film cooling effectiveness at high mass flux ratio $M$ of 1–1.5. It was also revealed that film cooling effectiveness could be greatly improved when the film cooling air was swirled. Cooled turbine blades may be manufactured with the additive manufacturing method in the future, and the relationship between the positions of the film cooling hole and internal cooling structure can be determined. In this case, it becomes possible to design turbine blades utilizing the swirling flow generated by the internal cooling structure to attain high film cooling effectiveness.

KEYWORDS
film cooling, trench, swirling flow, LIF, PIV, PSP

NOMENCLATURE

$C$ concentration
$D$ diffusion factor
$d$ film cooling hole diameter
$Le$ Lewis number
$M$ blowing ratio
$S$ swirl number
$T$ temperature
$u, v, w$ velocity
$x, y, z$ coordinate system
$\alpha$ thermal diffusivity
$\theta$ impingement jet angle
$\eta$ film cooling effectiveness
$\rho$ density
$\omega$ angular velocity

Subscripts
$a$ air
$aw$ adiabatic wall
$c$ coolant
$f$ spatially local film cooling
$w$ wall
$\infty$ mainstream flow

INTRODUCTION

Large liquid natural gas (LNG)-burning gas-steam combined-cycle power plants, with thermal efficiencies more than 35% higher than coal-burning thermal plants, have drawn increased attention with the advent of global warming, and are now in use all over the world. The latest combined-cycle plants use 1600 °C-class gas turbines as a topping cycle system. A higher turbine inlet temperature (TIT) ensures more effective CO$_2$ reduction. Figure 1 shows the evolution of the TIT of industrial gas turbines compared to aircraft engines. It shows the emergence of industrial gas turbine development about 10 to 15 years after aircraft engine development became established in the 1980s. Thus, the technical origins of industrial gas turbines can be traced back to aircraft engines. However, since the 1980s, the progress of industrial gas turbine development has become more independent of aircraft engine development. Industrial gas turbines differ from aircraft engines in the following ways:
Today, industrial gas turbines have established their position as one of the solutions to global warming [1].

The blades and vanes of a high-temperature gas turbine are cooled externally using film cooling. This is achieved by ejecting relatively cooler air from the internal coolant passages to the blade surface to form a protective layer between the blade surface and hot gas path flow. The interaction between the film cooling air and mainstream flow forms a shear layer that leads to mixing and the decline of film cooling performance along the blade surface. It is important to improve film cooling effectiveness by minimizing the film cooling airflow rate.

One of the characteristics of industrial gas turbines is the absence of a total weight limit even if the weights of auxiliary units are increased to improve thermal efficiency. Turbine cooling air can be cooled by a heat exchanger using the water of the bottoming cycle, as shown in Fig. 2. With this method, the total amount of cooling air can be reduced to cool the turbine blades and vanes. However, the cooled air still has sufficient capacity to cool after it has cooled the turbine blade internally. There is still the possibility of ejecting this low-temperature cooling air as film cooling air.

Looking back at the history of film cooling applied to turbine blades and vanes, inclined circular film cooling holes have been used since the beginning. The invention of fan-shaped film cooling holes, wherein the circular film hole exit was converted into a diffuser-shaped hole, was a groundbreaking development that improved film cooling efficiency [2]. However, even with the use of a diffuser-shaped film cooling hole, film cooling effectiveness did not match that of blowing air from a two-dimensional slot. Therefore, the height of film cooling effectiveness remains that of...
ejecting air through a two-dimensional slot. In order to achieve this, optimized fan-shaped film cooling holes, crater-shaped film cooling holes, etc. were developed [3] [4]. R. Bunker devised a trench film cooling hole based on a crater-shaped film cooling hole. This trench structure can be manufactured by a method which does not apply a thermal barrier coating on the belt zone of the film cooling holes. The cooling air ejected from the inclined circular film cooling holes mixes with a part of the main hot gas flow in the grooves of the trench, and the mixed refrigerant blows out two-dimensionally from the corner of the trench and covers the blade surface. Hence, it is believed that high film cooling effectiveness can be achieved by the trench film cooling hole.

Cooling air with sufficient capacity to cool internally yet mixes well with part of the mainstream flow was described above for trench film cooling holes. It is now considered whether this is possible with fan-shaped film cooling hole configurations. A deep digging fan-shaped film cooling hole was designed based on a fan-shaped film cooling hole, which can be manufactured by electric discharge sparking machining in the past. In this paper, the characteristics of the deep digging fan-shaped film cooling hole were investigated using quantitative measurement methods—particle image velocimetry (PIV), laser induced fluorescence (LIF), and pressure sensitive paint (PSP)—and a low-speed wind tunnel.

In the future, air-cooled turbine blades with a complex cooling structure can be manufactured using the additive manufacturing (AM) method. With the AM method, the location of air intake openings for film cooling holes can be accurately determined in relation to the internal cooling structures. This makes it possible to use the swirling flow generated by internal cooling of the turbine blades and vanes for improving film cooling effectiveness. Today, however, film cooling holes are drilled into turbine blades and vanes surfaces independently from the internal cooling structures. The latter part of this paper describes the link between the swirling flow generated by internal cooling of the turbine vanes and blades, and the improvement of film cooling effectiveness. It also discusses the results of experiments using a the low-speed wind tunnel to determine the effects of swirling film cooling air on the circular and fan-shaped film cooling holes, using the force generated by the swirling flow as a parameter.
EXPERIMENTAL METHODS

The experiments were conducted using a scaled-up model of film cooling holes installed at the bottom surface of a low-speed wind tunnel in order to allow detailed examination of flow features. The wind tunnel has an inlet nozzle with a 9:1 contraction ratio. The test section is 300 mm wide, 300 mm high, and 1950 mm long. The air speed inside the test section ranged from 0 to 40 m/s. For a free-stream velocity of 20 m/s, flow at the inlet test section showed excellent spatial uniformity, with free stream turbulence level less than 0.36%. The Reynolds number used in this experiment was one tenth of the number in actual industrial gas turbine conditions. The film cooling hole was located 625 mm downstream from the end of the contraction section.

Figure 3 shows the geometries of the test models for the film cooling holes [6]. The models are made of a low thermal conductivity material, Bakelite. The guide channels to the exits of the film cooling holes were inclined at 30° in the main flow direction. The shaped holes were composed of a round tube section with an exit, and a fan-shaped diffuser with 15° divergence angles on both lateral sides. Shaped hole (a) has the same geometry as in Takeishi [7]. The geometries of shaped holes (b) and (c) were numerically optimized by RANS using FLUENT 6.3, beginning with shaped film cooling hole (a). Shaped hole (b) has no diffused angle in the flow direction, while shaped hole (c) has a flow channel expanded from a deeper point than shaped hole (a). All the guide channels have diameters of 5 mm. The coordinate axes and the origins are also shown in Fig. 3.

![Figure 3: Film Cooling Hole Geometries](image)

Film cooling effectiveness was defined using mainstream temperature $T_\infty$, coolant temperature $T_c$, and local air temperature $T_f$ as follows:

$$\eta = \frac{T_\infty - T_f}{T_\infty - T_c} \tag{1}$$

The blowing ratio, which is defined as Eq. (2), is controlled by changing the film cooling flow rate.

$$M = \frac{\rho_c u_c}{\rho_\infty u_\infty} \tag{2}$$

The experimental conditions are listed in Table 1.

<table>
<thead>
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<th>Table 1: Experimental Conditions</th>
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<tr>
<td>Mainstream velocity $u_\infty$ [m/s]</td>
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<td>Blowing ratio $\frac{\rho_c u_c}{\rho_\infty u_\infty}$</td>
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<td>Turbulence intensity $Tu$ [%]</td>
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<td>Boundary layer thickness $\delta$ [mm]</td>
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Quantitative measurement methods are needed not only to understand the film cooling phenomena, but also to verify the accuracy of the numerical analysis. Accordingly, film cooling was measured using LIF, PIV, and PSP in the experiment.

LIF is a method for measuring mass concentration in gaseous flows. The LIF is non-intrusive, instantaneous, and has high fluorescence intensity. In addition, LIF can realize high spatial resolution for the tracer concentration field. The flow was illuminated by a laser sheet at a wavelength tuned to excite the specific absorption transition of a molecular tracer, which was added for this purpose. A fraction of the molecules in the lower energy level absorbed the incident light and was excited to a higher energy state. When the excited state returned to the lower energy states, fluorescence light was emitted at a wavelength different from that of the incident light. Thus, the fluorescence light from the tracer was extracted easily from the scattered light using an interference filter.

If the values of the thermal diffusivity of air and mass diffusivity of the tracer are close, then thermal diffusion can be replaced by mass transfer based on heat/mass transfer analogy. Hence, local film cooling effectiveness $\eta$ can be replaced by Eq. (3).

$$\eta = \frac{C_{\infty} - C_f}{C_{\infty} - C_v}$$

Lozano et al. [8] showed that acetone was one of the tracers for concentration measurements in gaseous flows by LIF; and that the ratio between the acetone vapor mass diffusion coefficient through air $D$, and the thermal diffusivity of standard air $\alpha$, was Lewis number $Le$, and was approximately 2 ($D = 11.2 \text{ mm}^2/\text{s}$, $\alpha = 22.3 \text{ mm}^2/\text{s}$). This was close enough to apply the heat/mass transfer analogy. In addition, acetone LIF shows good linearity with respect to the partial pressure of acetone in atmospheric pressure gas.

Fig. 4: Schematic Layout of Acetone LIF Method
An Nd:YAG laser operating at 266 nm was used to produce excitation laser light. The beam was expanded to a sheet 25 mm wide and 1 mm thick with four cylindrical lenses in order to pass through the test section, as shown in Fig. 4. The saturated acetone vapor in air was produced by bubbling the air in bottles containing acetone liquid. In order to obtain optimal acetone vapor concentration, dry air was mixed with air saturated with acetone vapor at a specific ratio.

PIV was used to capture instantaneous velocity fields. A schematic view of the experimental apparatus of PIV is shown in Fig. 5. The secondary flow blowing through the film cooling holes contained fine particles of olive oil as a tracer, which were 2–3 $\mu$m in diameter. The olive oil particles were generated by bubbling compressed air into a pool of olive oil through a Raskin nozzle in a pressure vessel. In the vessel, air bubbles burst at the olive-oil/air interface, and 2–3 $\mu$m olive oil particles were released into the air. A dual-pulsed Nd:YAG laser of wavelength 532 nm was employed to illuminate the tracers. Three cylindrical lenses were used to form a laser sheet. Images of particle patterns formed by the light reflected on the particle surfaces were taken with a charge-coupled device (CCD) camera located at the side of the test section, and the images were digitally stored in hard disks using the acquisition software HIPIC 8.0. To reduce the effect of reflection from the bottom surface, a background image for each pulse was subtracted from each frame to eliminate the effect of laser light reflection. The pairs of captured images were processed by a recursive local correlation method to obtain the velocity vector fields. The PIV measurements were conducted on a plane of symmetry, which was normal to the bottom and parallel to the sidewalls. Images of tracer particles illuminated by the laser were captured by the CCD camera (1024 × 1024 pixels) above the film cooling holes. The measurement field of the camera was 40 mm × 40 mm. The time interval between each double-pulsed laser shot was 10 $\mu$s. The corresponding displacement of a seeding particle between the two frames was approximately 5–6 pixels with free stream velocity 20 m/s. The average flow field was obtained by averaging and processing 500 frame pairs per measurement.

![Fig. 5: Experimental Apparatus and PIV Measuring System](image-url)
PSP is an optical pressure sensor that uses a special pigment that changes its luminescence intensity by reacting with oxygen molecules. The change in luminescence intensity is caused by optical quenching of the pigment by oxygen molecules. Therefore, a change in oxygen concentration can be measured by the luminescence intensity of the pigment. By using air (oxygen concentration 21%) as the mainstream flow and nitrogen (oxygen concentration 0%) as the film cooling jet, the oxygen concentration distribution, which is determined by the mixing of the mainstream (air) and coolant (nitrogen), can be measured on the wall downstream of the film cooling hole. The film cooling effectiveness was defined by eq. (3). A schematic view of the experimental apparatus of PSP is shown in Fig. 6. The wall of the wind tunnel was painted with PSP (ISSI PtTfPP FIB-UF405). Nitrogen gas for film cooling was supplied from a nitrogen gas tank. LED light sources were used as an excitation light source. The images were taken with a CCD camera (Hamamatsu photonics C9440-05C 1344 × 1024 pixels, 12bit) located at the opposite side of PSP painted wall. The light, except for phosphorescence (wavelength 650 nm), was cut by a bandpass filter. The images were digitally stored in hard disks using the acquisition software HIPIC 8.30. Three images—a background image, an image when the wind tunnel was paused, and an image when the wind tunnel was running—were captured. Twenty images were captured for each case and these were averaged. The PSP’s luminescence intensity was affected by temperature, and the temperature of the wall during the experiment was monitored by a K-type thermocouple. In the plenum, there were screen and glass beads, and the airflow had a uniform velocity profile before entering the film cooling hole.

RESULTS AND DISCUSSION
Fan-Shaped Film Cooling Geometries
Figure 7 shows the time-averaged velocity fields obtained by PIV near the exits of the circular hole and shaped hole (a) at blowing ratio $M = 1.0$. In the figures, the colors represent the velocity magnitude, which is given by $\sqrt{\nu^2 + \nu^2}$. In the case of the circular hole, the injected air had a large velocity magnitude over the entire hole exit. However, in the case of shaped hole (a), the injected air was concentrated at the upstream side of the hole, as shown by the large velocity magnitude.
\(-1 < x/d < 0\), while having a small velocity magnitude at the downstream side of the hole.

Figure 8 shows the time-averaged film cooling effectiveness distribution obtained by acetone LIF near the exit of the circular hole and shaped hole (a), at blowing ratio \(M = 1.0\). In the case of the circular hole shown in Fig. 8, the cooling air penetrated the main flow and peeled off from the bottom surface. The counter rotating vortex (CRV) was clearly captured by the LIF method. CRV entrained the hot gas between the film cooling air and surface. Because of these factors, film cooling efficiency in the vicinity of the wall surface decreased. However, in the case of shaped hole (a), cooling air penetration was suppressed; the film cooling air adhered to the wall and the fan-shaped film cooling hole attained higher film cooling effectiveness compared to the circular hole.

![Fig. 7: Flow Field Measurement by PIV at \(M=1.0\)](image)

![Fig. 8: Film cooling effectiveness measured by acetone LIF](image)

Figure 9 shows the instantaneous and simultaneous distributions of velocity vectors and film cooling effectiveness values near the exits of the circular hole and shaped hole (a), measured by acetone LIF and PIV at \(M = 1.0\). The arrows represent the velocity vectors in the \(z\) equal zero plane, while the color contours show the film cooling effectiveness. In the case of the circular hole, positive vertical components of velocity were observed at the cooling hole exit at \(-2 < x/d < -1\). This indicates that the film cooling jet penetrated the mainstream flow. Further downstream, but still over the circular hole exit, the velocity vectors showed stronger penetration relative to shaped film cooling hole (a). Downstream of the cooling hole (between \(2 < x/d < 4\)), the measured vertical velocities were positive, indicating that some of the mainstream fluid was ingested under the film cooling air, which elevated the film cooling air. The film cooling air through the circular film cooling hole was lifted from the surface.

In the case of shaped film cooling hole (a), less penetration was observed, as shown by the slightly positive vertical velocity at the exit of hole (a). Upstream of the hole centerline between \(-2\)
$< x/d < 2$, the measured vertical velocities were quite small, indicating that a coolant film was being produced well over shaped film cooling hole (a), but an instantaneous upward penetration was detected in the turbulent shear layer at $M = 1.0$. A detailed and enlarged view of the yellow square region is shown in Fig. 10. A shear layer with large velocity gradients can be seen downstream. This shear layer creates a complex boundary between the film core and mainstream flow. The shape and velocity gradients of this shear layer were influenced by the blowing ratio. A Kelvin–Helmholtz instability was generated and developed along the turbulent shear layer. The details of the mixing phenomena generated by the Kelvin–Helmholtz instability in the turbulent shear layer were successfully captured by the simultaneous velocity and concentration fields using PIV and acetone LIF.

Fig. 9: Velocity vectors and film cooling effectiveness by LIF and PIV at $M = 1.0$

Fig. 10: Detail distribution of yellow square zone
Figure 11 shows the distribution of the film cooling effectiveness in the instantaneous field. In film cooling holes (b) and (c), the interface between the film cooling jet and mainstream flow showed a complex shape in the instantaneous field. The vortex induced by the shear layer formed the complex interface. It can be seen that the vortices combined with each other and grew gradually toward the downstream direction. In addition, the instantaneous flow field results captured the moment when some parts of the coolant air were isolated from the film cooling jet. The diffusion of the coolant jet was conducted by large-scale turbulent eddies in the shear layer. As shown in the figure, the film cooling jet finally dissipated and disappeared at downstream. From the experiments, it became clear that shaped film cooling hole (c) had higher film cooling effectiveness than shaped film cooling hole (b) when the blowing ratio $M$ was 1.0–1.5.

Figure 12 shows the time-mean film cooling effectiveness of shaped holes (b) and (c), which were analyzed by detached eddy simulation (DES) at $M = 0.5$. As the figure indicates, the DES predictions show a longer coolant coverage than LIF. That is, the coolant flows in DES were less diffused than in the experiments. This is mainly because the present DES did not have a function to create fluctuating velocities at the inlet boundary, which play a key role in initiating flow instability in the shear layer between the coolant jet and mainstream flow. As mentioned earlier, instantaneous fields by DES in Fig. 12 provide insight into why shaped hole (c) had relatively lower film effectiveness at the holes exits when $M$ increased, as shown in Fig. 11. From Fig. 12, it is found that pre-mixing of the coolant and mainstream flow occurred inside shaped hole (c) from the influx of the mainstream flow due to the longer and deeper shape of hole (c) compared to shaped hole (b). The frequent occurrence of this phenomenon promoted the mixing of film cooling air with the mainstream flow in a statistical sense. As a result, film cooling effectiveness for shaped hole (c) was estimated to be higher than that for shaped hole (b) at the far downstream region from the film cooling hole exit.

![Fig. 11: Film Cooling Effectiveness in Instantaneous Fields for Shaped Hole (b) and (c)](image)

![Fig. 12: Instantaneous Results by DES at $M = 0.5$](image)
Effect of Swirling Flow on Film Cooling Effectiveness

To investigate the effect of swirling flow on film cooling effectiveness, swirling coolant flow was generated artificially in a cavity located in a plenum before entering the film cooling hole. A schematic view of the cooling structure is shown in Fig. 13. The cross-section shape of the cavity is hexagonal. It has two impingement jet holes inclined at $\theta$ degrees toward the vertical direction. The two impingement jets generated swirling flows inside the cavity, and this swirling flow entered the film cooling hole while maintaining angular momentum. The Swirl number $S$ at the film cooling hole exit was measured by PIV method. Two impingement jet angles were changed to $\theta = 0^\circ, 10^\circ, 20^\circ, 30^\circ$, and the corresponding Swirl numbers were $S = 0, 0.0289, 0.116, \text{ and } 0.168$, respectively. The experimental conditions were same as those listed in Table 1.

Figure 14 shows the film cooling effectiveness contours of the circular and shaped holes at $M = 1.0$, measured by PSP method with $\theta = 0^\circ$–$30^\circ$. For non-swirling flow with a circular hole shown in Fig. 14, the value of film cooling effectiveness just downstream of the film cooling hole was very low. The reason for this low value was the penetration of the film cooling jet into the mainstream flow and its separation from the wall. At the downstream, the film turned to the wall directed by the mainstream flow and reattached to the surface at $x/d = 3$, and the film cooling effectiveness recovered to approximately 0.2. At the Swirl number of $\theta = 10^\circ$, the area covered by film cooling air decreased. However, increasing Swirl number suppressed the penetration of the film cooling jet and a high film cooling effectiveness area appeared just downstream of the film cooling hole exit. At $\theta = 30^\circ$, the highest film cooling effectiveness of approximately 0.6 was obtained just downstream the film cooling hole exit. The result indicates that the penetration of the film cooling jet into the mainstream flow was suppressed by the swirling motion of the film cooling air. It appeared that the center line of the cooling jet flow was deflected to the negative $z$ direction at the film cooling hole exit. Considering the relation between the swirling direction and deflected direction of the film jet, it was found that the cooling jet moved in the same direction as the mainstream flow. This bending force is believed to be the Magnus effect.

Figure 15 shows the spanwise averaged film cooling effectiveness for the circular hole at $M = 1.0$ with $\theta = 0^\circ$–$30^\circ$. This figure shows that swirling coolant at $\theta = 30^\circ$ improved the averaged film cooling effectiveness by approximately 50% at $x/d = 10$ and 100% or more at far downstream of $x/d = 20$. This figure suggests that swirling coolant flow was effective in improving the film cooling effectiveness of circular holes by suppressing both the penetration of the cooling jet into the mainstream flow and the generation of an anti-kidney vortex structure.

![Figure 13: Structure of film cooling with swirling flow](image)
Figures 16 and 17 show the time-averaged film cooling effectiveness distribution at $z/d = 0$, and the cross sections at selected locations obtained by acetone LIF near the circular hole exit at $M = 1.0$. For the circular hole, the penetration of the film cooling jet into mainstream flow was suppressed by the swirling coolant flow and attained higher film cooling effectiveness compared to the non-swirling coolant flow case.

**Fig. 14: Film Cooling Effectiveness Measured by PSP (Circular Hole, $M = 1.0$)**

**Fig. 15: Spanwise Averaged Film Cooling Effectiveness (Circular Hole at $M=1.0$)**
Figure 17 shows the spatial distributions of time-mean film cooling effectiveness at $x/d = 0, 2, \text{ and } 4$ obtained by LES and acetone LIF measurement with $\theta = 30^\circ$. Comparison of Figs. 17 and 8 clearly shows that the penetration of the coolant jet at $M = 1.0$ was suppressed by the swirling motion of the film cooling jet. It seems that one of the two jet core regions, which were formed by a secondary swirling flow induced by the primary swirling motion of the coolant jet (or by the axial velocity of strongly ejected jet at $M = 1.0$), was pressed against the bottom wall by the primary swirl of the coolant jet itself. This mechanism appears to prevent the detachment of the film coolant from the wall and maintained relatively higher film effectiveness downstream.

Figures 18 shows the time-averaged velocity vectors and vorticity distributions measured for the circular film cooling hole at $M = 1.0$ by PIV method. The generation of the anti-kidney vortex structure was affected by the swirling motion of the film cooling flow at $\theta = 30^\circ$, and film cooling air adhered to the wall.

Fig. 17: Film Effectiveness at $x/d = 0, 2, \text{ and } 4$ in Swirling Film Coolant Cases
Fig. 18: Time-Mean Vorticity Distribution of Circular Hole ($M = 1.0$)

Figures 19 shows the film cooling effectiveness contours of shaped hole (a) at $M = 1.0$ measured by PSP method with $\theta = 0^\circ$–$30^\circ$. Comparison of Figs. 14 and 19 shows that the film cooling jet blowing through shaped cooling hole (a) spreads wider in both the lateral and longitudinal directions. Shaped film cooling attained higher film cooling effectiveness compared to the circular hole by the incorporation of film hole exit shaping, which resulted in lower momentum coolant injection and greater surface coverage. The film cooling effectiveness reached a maximum at $\theta = 10^\circ$ and decreased with a smaller or larger $\theta$. Figure 20 shows that the swirling coolant also improved the averaged film cooling effectiveness of shaped film cooling hole (a) by approximately 100% at far downstream of $x/d = 10$. Figure 21 shows the measured cross section film cooling effectiveness at $\theta = 0^\circ$ and $10^\circ$. The film cooling effectiveness reached a maximum value at $\theta = 10^\circ$ with the shaped geometry shown in Fig. 3. It is clear from Fig. 21 that the swirling film cooling flow suppressed the penetration or diffusion of the film cooling air in the $y$-direction, and spread the film cooling air in the spanwise direction. As a result, it covered a wide area on the wall and attained the highest film cooling effectiveness.

Fig. 19: Film Cooling Effectiveness Measured by PSP (Shaped Hole (a), $M = 1.0$)
Fig. 20: Spanwise Averaged Film Cooling Effectiveness (Shaped Hole (a), $M = 1.0$)

Fig. 21: Cross Section Film Cooling Effectiveness (Shaped Hole (a), $M = 1.0$)
CONCLUSIONS
In industrial gas turbines, it is possible to use cooling air that was externally cooled using the water of the bottoming cycle for turbine cooling. A deep digging fan-shaped film cooling hole with the property of a trench was developed, and the characteristics of the film cooling hole were clarified by quantitative measurement methods. Film cooling with swirling flow was also clarified, and it was shown that film cooling effectiveness greatly improved for both circular and shaped holes. The following conclusions are drawn from the experiment.

1) The deep digging-shaped hole functioned like a trench, a phenomenon wherein part of the mainstream flow was mixed with the film air in a space at the film hole exit, and high film cooling effectiveness downstream of the hole was obtained at mass flux ratio $M = 1–1.5$.

2) Quantitative measuring methods, such as LIF, PIV, and PSP, were applied to clarify the film cooling phenomena. These methods were very useful for understanding the mixing phenomenon of complicated film cooling and verifying the numerical analysis of film cooling.

3) Swirling flow generated in film cooling improved film cooling effectiveness for both circular and shaped holes. In the future, if it is possible to accurately determine the positional relationship between the position of the film cooling hole and the internal cooling structure, for example AM method, if it becomes possible to utilize the swirling flow generated by internal cooling, film cooling is possible.

REFERENCES