IMPROVEMENT OF TURBINE VANE FILM COOLING PERFORMANCE BY DOUBLE FLOW CONTROL DEVICES

Hirokazu KAWABATA, Ken-ichi FUNAZAKI, Yuya SUZUKI
Department of Mechanical Engineering
Iwate University, Iwate, Japan

Hisato TAGAWA, Yasuhiro HORIUCHI
Mitsubishi Hitachi Power Systems, Ltd. Research & Development Center
Ibaraki, Japan

ABSTRACT
This study deals with the studies of the effect of flow control double device (DFCD) on a turbine vane film cooling. Aiming for improving film effectiveness, two semi-elliptical DFCDs per a pitch were attached obliquely upstream of the cooling hole. Since the DFCDs were applied to flat plate film cooling in the previous study, the applicability to the turbine vane was investigated in this study. In order to observe a flow field in detail, RANS CFD was conducted first. The DFCDs were installed upstream of each cooling hole of the pressure and suction sides of the vane to investigate the effect of the device position. In this paper, the effects of blowing ratio and cooling hole pitch were also investigated. The results obtained by CFD showed that the vortex generated from DFCD suppressed lift off of the secondary air. As a result, the film effectiveness became significantly higher than that without DFCD condition at high blowing ratio. Moreover, the improvement in the film effectiveness by DFCD was observed by both the pressure and suction sides of the turbine vane. Based on the findings through RANS simulation, adiabatic effectiveness and total pressure loss coefficient measurement were performed in a linear cascade test facility. The experiment confirmed that the film effectiveness improved when DFCDs existed.

1. INTRODUCTION
In order to raise thermal efficiency of gas turbine, higher turbine inlet temperature (TIT) is needed. Since higher TIT increases thermal load to its hot-section components, reducing their life span, very complicated cooling technology such as film cooling and internal cooling is required especially for HP turbine vanes and blades. Among several cooling methods, film cooling is a very effective one because the cooling air injected onto the blade surface form a protective layer between the surface and hot mainstream gas. However, because of limited amount of cooling air permitted in a gas turbine especially in an aeroengine, the development of new technologies for turbine cooling needs to be explored in order to minimize the cooling air consumption.

One of the research trends in turbine cooling technology is providing flow-control structure around cooling holes. Barigozzi et al. [1][2] observed that film effectiveness was improved by use of a ramp combined with various cooling hole shape. Lu et al. [3] changed several kinds of shape of trench applied to cooling holes with conventional round hole exit shape, and carried out cooling performance comparisons between fan-shaped hole and the cooling hole with trench. Rallabandi et al. [4] installed step upstream of the cooling hole, and observed film cooling effectiveness using PSP. Sakai et al. [5] clarified flow structure and temperature field when putting bump on the downstream of a cooling hole experimentally and numerically. Kawabata et al. [6] proposed a protrusion-type flow-control device (FCD) installed onto the upstream surface of the cooling holes to increase the film effectiveness. They examined the aero-thermal effects of the device height as well as off-set distance between the hole centreline and the device. It was confirmed that the tall device provided higher film effectiveness due to a strong vortex structure generated by the device. Funazaki and Kawabata [7][8] proposed double flow control device (DFCD) which installed two FCDs per one cooling hole. Figure 1 shows a flow model around DFCD and cooling hole. A counter rotating vortex pair (CRVP) is generated by the interaction of the mainstream and the secondary air injected from the cooling hole. This CRVP promotes lift-off of the secondary air from the surface. On the other hand, DFCDs generate streamwise vortices rotating in opposite directions each other nearby the injected air. The secondary air tends to attach to the surface due to the downwash induced by the device-based streamwise vorticies (DBV).
However, most studies that flow control structure was attached to the cooling holes were performed by using flat plate film cooling hole.

This research aims at observing the effect of flow–control structure using a turbine vane airfoil model. Sundaram et al. [9] installed trench and bump to the cooling hole of vane endwall. They showed that comparatively short bump did not produce a major effect on effectiveness, although a tall bump did show an enhancement of effectiveness. Comparatively detailed research is carried out about turbine vane airfoil with trench [10][11][12]. In this paper, the effect of DFCD on turbine vane airfoil film cooling was investigated.

![Flow model of DFCD](image)

### NOMENCLATURE

- **BR**: Blowing ratio ($\eta = \frac{p_2U_2/p_1U_1}{\eta}$)
- **C**: Actual chord length
- **Cax**: Axial chord length
- **Cp**: Static pressure coefficient
- **DR**: Density ratio
- **d**: Diameter of film cooling hole, mm
- **h**: Height of DFCD
- **L**: Distance of cooling hole and DFCD
- **P**: Pressure, Pa
- **p**: Cooling hole pitch
- **Re**: Reynolds number based on actual chord length and exit velocity
- **s**: Coordinates along pressure side surface
- **t**: Pitch of the cascade
- **T**: Temperature, K
- **V**: Outlet velocity, m/s
- **x, y, z**: Cartesian coordinates, mm
- **η**: Film cooling effectiveness
- **ρ**: Density, kg/m$^3$

### Abbreviation

- **CRVP**: Counter-Rotating Vortex Pair
- **DBV**: Device-Based Vortex
- **DFCD**: Double Flow-Control Device
- **FCD**: Flow-Control Device

### Subscript

- **w**: Adiabatic wall
- **s**: Static quantity
- **t**: Total quantity
- **∞**: (Relative to) mainstream
- **in**: Inlet ($x=1.25C_{ax}$)
- **out**: Exit ($x=0.25C_{ax}$)
- **2**: (Relative to) secondary air

## 2. EXPERIMENTAL SETUP

### 2.1 Experimental Apparatus

Figure 2 shows the experimental facility consisting of six-vane cascade at Iwate University. The test section duct was built from acrylics plates, with cross section is $580\text{mm}(width) \times 117\text{mm}(height)$. The mainstream flow was supplied by a centrifugal blower. Secondary air was supplied from the compressor and branched to two channel. This is because the secondary air was supplied for every plenum of test vane. The heater and the Azbil corporation CMS series mass flowmeter were installed in each channel. The test vane with cooling holes is shown by the red in this figure. In this research, since the wall temperature of the test vane was measured by IR camera(NEC Avio), Asahi Kasei Engineering Corporation’s polyolefin window (thickness of 0.5mm) was installed in the tip of test model. Three IR camera were installed diagonally on the upper side of the measurement windows. Figure 3 shows the location of IR camera. Since suction side of vane has strong curvature, two IR cameras were installed in the test section. The Pitot tube was installed in the $2.0C_{ax}$ upstream and $0.25C_{ax}$ downstream from the trailing edge from the test vane.

### 2.2 Test vane and DFCD configuration

Figure 4(a) and Table 1 show test vane configuration and geometric parameters. The vane has four rows of film cooling holes. Two rows of cooling holes are at suction and pressure side surface. The diameter of the cooling hole($d$) is 1.1mm. As for the pitch($p$) of the cooling holes, 3.0d and 4.5d were investigated. In this research, the coordinate system along the test model surface($s_p$ and $s_s$) was defined. The origin position is leading edge and trailing edge is $s_p = s_s =1.0$. Figure 4(b) shows the observation area of each of IR cameras. The spanwise observation area is the part for the 25% span height of midspan. The position and measurement area of IR camera correspond to Figure 3. The test vane was produced by Stratasys Objet 30 Pro. High temperature material RGD525 was used for 3D printing. The thermal conductivity of the test model is 0.22W/(mK).

Figure 5 and Table 2 shows the DFCD configuration and installation position. The shape of DFCD was determined as reference in research of Funazaki et al.[7]. Since the height of DFCD($h$) has strong influence against DBV, it was changed to
0.5d and 1.0d. The inclination angle cooling hole differ for every row. Therefore, the sizes of L vary in each position.

2.3 Experimental method

2.3.1 Static pressure coefficient measurement

The following equation defines the static pressure coefficient in this study,

\[ C_p = \frac{P_{1,in} - P_s}{0.5 \rho V_s^2}, \]

where \( V_s \) is the averaged exit velocity measured by a Pitot tube on the traverse line located at outlet position.

2.3.2 Film effectiveness measurement

The performance of a film cooling technique was evaluated by means of adiabatic film cooling effectiveness as been given by equation (2). \( T_s, T_w \) and \( T_2 \) are the corrected wall temperature, mainstream temperature and secondary air temperature respectively. \( T_s \) and \( T_2 \) were measured by thermocouples at mainstream inlet region and the midspan position of each plenum. The measurement lasted for 20 seconds to enable steady state analyses of film cooling effectiveness.

\[ \eta = \frac{T_s - T_w}{T_w - T_2}, \]

Blowing ratio \( (BR = \rho \frac{U_s}{U_e} \) was defined by the average blowing ratio of each cooling hole row in this measurement. Furthermore, \( BR \) has been set to be the same in all the cooling hole rows. There, the \( U_e \) was calculated from static pressure coefficient measurement result. The \( U_e \) was calculated from the mass flow rate of the secondary air and holes area. When film effectiveness was measured, suction and pressure side were measured respectively. Note, the suction side was measured with the IR camera of view1 and view2 at the same time.

In order to measure the surface temperature, the IR camera had to be installed at an angle in this study. Therefore, image obtained from IR camera had curvature. IR image had to be transformed into rectangular coordinates in order to evaluate the temperature data. Colban [13] carried the transformation to rectangular coordinates by obtaining a grid image on a test vane surface. In this study, the model surface was marked with grid points every 2% in the direction along the surface and 5% in the vane span direction respectively. Temperature image and grid point were collated, and the image was converted to rectangular coordinates by the in-house program.

2.3.3 Aerodynamic loss measurement

In this research, aerodynamic loss was measured by each vane. In this measurement, each cooling hole was closed by the tape and \( BR=0.0 \). Primary loss coefficient was defined by equation (3).

\[ \xi_{pr} = \frac{P_{1,in} - P_{2,out}}{0.5 \rho V_{out}^2}, \]

\( P_{2,out} \) is total pressure measured by a Pitot tube on the traverse line located at outlet position.

2.4 Test Condition

The Reynolds number in this study, based on the actual chord length and averaged exit velocity, 497,000. The flow velocity and temperature in the duct entrance were about 17 m/s and 300 K, respectively. Blowing ratios examined were 0.5 and 1.0. The density ratio, \( DR \), were 0.9 and 1.0 for the thermal and aerodynamics measurements, respectively. Table 3 shows test conditions in this research.
Table 2 Geometric specification for film cooling holes

<table>
<thead>
<tr>
<th>Row name</th>
<th>Position ($x/C_{ax}$)</th>
<th>$L$</th>
</tr>
</thead>
<tbody>
<tr>
<td>PS1</td>
<td>0.34</td>
<td>2.0$d$</td>
</tr>
<tr>
<td>PS2</td>
<td>0.80</td>
<td>3.3$d$</td>
</tr>
<tr>
<td>SS1</td>
<td>0.15</td>
<td>1.3$d$</td>
</tr>
<tr>
<td>SS2</td>
<td>0.62</td>
<td>1.5$d$</td>
</tr>
</tbody>
</table>

2.5 Uncertainty analysis

The accuracy of the measurement was determined by performing uncertainty analysis using the methodology of Kline and McClintock [14]. The accuracy of the pressure transducer was ±0.25%. The uncertainty of inlet velocity was about ±2.2%. The uncertainty of a blowing ratio is 2.8% when there is least blowing ratio condition. In aerodynamic investigation, the uncertainty of the static pressure coefficient and total pressure loss coefficient was about ±1.6% and ±2.1% respectively around the peak region of the coefficient.

Normally, it is preferred that surface temperature is to be measured vertically in the IR camera measurement. However, because IR camera is installed diagonally, it is thought that the measured temperature is lower than an original temperature in some areas. Measurement angular dependency was investigated using iso-temperature flat plate. When a measurement angle was 60 degrees, measurement temperature dropped by about 1K from original temperature.

The uncertainty of the film cooling effectiveness was evaluated in consideration of mainstream temperature, secondary air temperature, wall temperature. The reference wall temperature assumed in this analysis corresponded to the case of $\eta$=0.5. The uncertainty of film cooling effectiveness became ±3.9% as a result.

3. NUMERICAL SIMULATION

In this study, it should be checked whether DBV occurs on the test vane. Then, CFD approach was applied. A commercial software, ANSYS CFX 13 was used in this CFD. Reynolds-Averaged Navier-Stokes (RANS) approach using Shear-Stress Transport (SST) two-equation model was employed. Figure 6 shows the computational domain simulating the 1 pitch cooling hole and the mesh. The span height of the domain (=cooling hole pitch) was 6$d$. This cooling hole pitch was larger than experimental condition. This is because mesh creation is difficult, when DFCD and a boundary position are near. At this CFD, although interference of DBV of the spanwise direction cannot fully be observed, the main tendencies of DBV can be investigated.

Although tetra meshes were mainly used for the computational grid, prism meshes were also used in order to resolve boundary layer at near wall region. From the mesh dependency test it was found that about 9 million cells in this domain were adequate, where 8 million cells were used for the test duct region and 1 million cells were used for the plenum and film holes region. The value of $y+$ for the computational point of
the first cell above the wall was less than unity so that the wall function approach was not applied on the wall in RANS. The mainstream flow velocity and temperature measured in thermal experiment were specified at the mainstream inlet boundary condition. The mass flow rate and temperature were imposed at the secondary air entrance. The boundary condition of periodic was used for vane and cooling holes pitch direction.

4. RESULTS AND DISCUSSION

4.1 Static pressure coefficient

Figure 7 shows the static pressure coefficient distribution obtained by experiment and CFD. In the figure, the line of red shows the installation position of each cooling hole. The local velocity for computing $BR$ is based on this data. The CFD result and the experimental result were in agreement mainly. Because mesh quality near the trailing edge was low, gap with an experimental value was locally large.

4.2 Flow fields around DFCD (CFD results)

Figure 8 (a) shows the film cooling effectiveness distribution and iso-surfaces of Q criterion in suction side at $BR=0.5$. Upper 2 surface are without and with DFCD ($h=0.5d$) condition respectively. With DFCD condition indicated the cooling pattern more uniform than without DFCD condition. The iso-surfaces of Q criterion shows clearly the vortex core of DBV diagonally generated from DFCD. Figure 8 (b) shows the pressure side result. The effect of DFCD was also confirmed in pressure side. However, there is non-cooled area due to a concave curvature on which the secondary air is easier to lift off.

4.3 Film cooling effectiveness (Experimental result)

Figure 9 shows the film cooling effectiveness distribution at $p=3.0d$, $BR=0.5$. In $s_z=0.24$, the contour is discontinuous slightly because there is uncertainty of two IR cameras. In Area1, the difference was observed by film cooling effectiveness distribution by the existence of DFCDs. Film cooling effectiveness distribution expanded in the spanwise direction as $h$ became large. However, damping of film cooling effectiveness in the $s_z$ direction also became large. Kawabata [6] and Nakata [15] indicated that the RMS velocity fluctuation increased, when DFCD existed at flat plate film cooling. This is because mixing of the mainstream and secondary air is promoted by DBV. By with DFCD condition, although film cooling effectiveness distribution expanded in the spanwise direction, damping of the $s_z$ direction was not observed Area2. Since pressure side has concave curvature, the secondary air from PS1 separates from the model surface at without DFCD condition. When DFCD was installed, secondary attach to the model pressure side, but damping of the $s_p$ direction was similarly observed with suction side. In the trailing edge area, the performance is falling in with DFCD condition by the effect of mixing.

Figure 6 Computational domain and mesh

Figure 7 Static pressure coefficient distribution

Figure 8 Film cooling effectiveness distribution and vortex core region at $BR=0.5$

Figure 9 Film cooling effectiveness distribution at $p=3.0d$, $BR=0.5$. In $s_z=0.24$, the contour is discontinuous slightly because there is uncertainty of two IR cameras. In Area1, the difference was observed by film cooling effectiveness distribution by the existence of DFCDs. Film cooling effectiveness distribution expanded in the spanwise direction as $h$ became large. However, damping of film cooling effectiveness in the $s_z$ direction also became large. Kawabata [6] and Nakata [15] indicated that the RMS velocity fluctuation increased, when DFCD existed at flat plate film cooling. This is because mixing of the mainstream and secondary air is promoted by DBV. By with DFCD condition, although film cooling effectiveness distribution expanded in the spanwise direction, damping of the $s_z$ direction was not observed Area2. Since pressure side has concave curvature, the secondary air from PS1 separates from the model surface at without DFCD condition. When DFCD was installed, secondary attach to the model pressure side, but damping of the $s_p$ direction was similarly observed with suction side. In the trailing edge area, the performance is falling in with DFCD condition by the effect of mixing.
Figure 9 Film cooling effectiveness distribution 
($p=3.0d, BR=0.5$)

Figure 10 shows the film cooling effectiveness distribution at $p=3.0d, BR=1.0$. In Area1, cooling area became small for lift-off of secondary air in without DFCD condition compared with $BR=0.5$ case. This is because the momentum ratio of the secondary air and the mainstream was increased. About the with DFCD condition, the uniform cooling pattern was formed in the $s_t$ and spanwise direction. The $h$ becomes higher, lift-off was more suppressed. Furthermore, unlike $BR=0.5$, damping of the film cooling effectiveness of the $s_t$ direction was not observed. Since there was much flow rate of secondary air, all the Area2 were cooled in all the conditions. DBV made the local film cooling effectiveness near cooling holes increase. In pressure side, the secondary air lifts off compared with the case of $BR=0.5$. By the effect of DFCD, lift-off of secondary air was also suppressed. However, although local film cooling effectiveness increased with $h$ at suction side, change of the film cooling effectiveness by $h$ is not observed by pressure side. It is thought that the lift-off control effect by DFCD was saturated under the influence of the concave curvature of pressure side.

Figure 11 shows the film cooling effectiveness distribution at $p=4.5d, BR=0.5$. In this condition, cooling holes pitch widened, so non-uniform cooling pattern was formed. In Area1, expansion of the film cooling effectiveness distribution by DBV was limited near the cooling holes. Although uniform cooling pattern in the spanwise direction was observed in $h=1.0d$, damping of the $s_t$ direction was large. In the case of $h=0.5d$, DFCD was the most effective at Area2 in terms of cooling pattern. As for the pressure side, uniform cooling pattern was hard to be formed compared with suction side.
Figure 11 Film cooling effectiveness distribution

Figure 12 Film cooling effectiveness distribution

Figure 13(a) shows spanwise averaged film cooling effectiveness at $p=3.0d$ condition. At $BR=0.5$, the effect of DFCD in the Area1 and Area2 was different. Although film cooling effectiveness distribution expanded in the spanwise direction by DFCD, the average value of without DFCD condition was the highest at Area1. In Area2, the averaged film cooling effectiveness near the cooling hole increased by DFCD. However, the effect of DFCD was very restrictive and there were few advantages by DFCD installation. Also about pressure side, the effect of DFCD was observed only some areas. For high $BR$, unlike $BR=0.5$, the amount of increase in film cooling effectiveness by DFCD was increased. Result of two conditions attached with DCFD were almost identical in Area1. This is because the effect of DBV becomes restrictive, because strong concave curvature promoted separation of secondary air in Area1. On the other hand, in Area2 with the comparatively flat surface, film cooling effectiveness increased as $h$ became large. Unlike
Figure 13 Spanwise averaged film cooling effectiveness

(a) $p=3.0d$ condition

(b) $p=4.5d$ condition

**BR=0.5**, on this condition, the effect of DFCD been observed to the downstream region. As for the pressure side, the film cooling effectiveness of DFCD condition was high in all the areas. A significant difference of the film efficiency by $h$ was not observed compared with the suction side result.

Figure 13(b) shows spanwise averaged film cooling effectiveness at $p=3.0d$ condition. By pitch to expand, film cooling effectiveness decreased overall. The interference of DBV in the spanwise direction was changed compared to the $p=3.0d$ at $p=4.5d$, but there was no change in the basic trend of the averaged film cooling effectiveness. Funazaki [7] proved that the condition with DFCD at $p=4.5d$ exceed the condition with DFCD at $p=3.0d$ in terms of the averaged film cooling effectiveness. As a result, the improvement of film cooling effectiveness and reduction of the number of cooling holes was achieved at flat plate film cooling. However, in this research, the above-mentioned tendency was not observed on $p=4.5d$ conditions. Therefore, the effect of DFCD fell rather than flat plate model film cooling.

### 4.4 Aerodynamic loss (Experimental result)

Figure 14(a) shows the primary loss coefficient distributions of without and with DFCD vanes. The wake width on the suction side of with DFCD vane was larger than that of without DFCD vane, while the wake on the pressure side had comparatively small change. This is guessed to originate in mixing being promoted by DBV. On the other hand, in the case of the $4.5d$ pitch (Figure 14 (b)), the variation of wake width was reduced rather than the $p=3.0d$ case. Especially, in the case of $p=4.5d$, $h=0.5d$, wake width of pressure side was slightly smaller than without DFCD condition. Therefore, it is suggested that interference of DBV of spanwise direction has remarkable influence in aerodynamic loss.
3. In Pressure side, the effect of the height of DFCD is not observed rather than suction side. Because secondary air easily separates on concave surface, it is thought that the effect of DBV is saturated.

4. The aerodynamic loss changes depending on the height and installation space of DFCD. In the future, it is necessary to optimize DFCD in terms of aero-thermal performance.

ACKNOWLEDGMENTS
This research and development has received funding for advanced technology development for energy use rationalization from the Electricity and Gas Industry Department of the Agency for Natural Resources and Energy at the Ministry of Economy, Trade and Industry. We would like to express our thanks to the organizations.

REFERENCES

5. CONCLUSIONS
The effects of double flow control device on turbine vane film cooling were experimentally and numerically studied. In this study, aero thermal performance was investigated by changing the height of DFCD, cooling hole pitch and the location, and the following conclusions were acquired.

1. It was shown clearly by CFD that DBV generated form DFCD even if DFCD was installed in Curved surface. This DBV has an effect which extends secondary air in the span direction like research of the past flat plate film cooling with DFCD.

2. Because suction surface has a concave curvature, the secondary air easily attach to the model surface. The effect of DFCD increased with the height of DFCD in this surface. Particularly, in a high BR, film cooling effectiveness was increased effectively due to DBV. However, because DFCD promotes mixing of mainstream and secondary air, the increment of film cooling effectiveness is restrictive at low BR condition.

![Figure 14 Primary loss coefficient distribution](image-url)


