An Experimental Study of Heat Transfer and Film Cooling on Low Aspect Ratio Turbine Nozzles

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ABSTRACT

The effects of the three-dimensional flow field on the heat transfer and the film cooling on the endwall, suction and pressure surface of an airfoil were studied using a low speed, fully annular, low aspect ratio h/c=0.5 scale vane cascade.

The predominant effects that the horseshoe vortex, secondary flow, and nozzle wake increases in the heat transfer and decreases in the film cooling on the suction vane surface and the endwall were clearly demonstrated. In addition, it was demonstrated that secondary flow has little effect on the pressure surface. Pertinent flow visualization of the flow passage was also carried out for better understanding of these complex phenomena. Heat transfer and film cooling on the fully annular vane passage surface is discussed.

NOMENCLATURE

C = vane chord
D = film cooling hole diameter
h = vane height
M = blowing rate, M = \frac{p_2U_2}{\rho_0U_0}
P = film cooling hole pitch
P = circumference pitch of the vane
Re = Reynolds number
r = radius
s = equivalent film cooling slot width = \frac{b^2}{\nu \rho}
s = vane throat width
T = temperature
U = velocity
x = distance downstream of film cooling holes or leading edge
\alpha = heat transfer coefficient
\delta_{g\theta} = boundary layer thickness;
\eta_f = film cooling effectiveness
\rho = specific density
\tau = mainstream turbulence intensity

Subscript
aw = adiabatic wall
m = main stream
ex = exit
in = inlet
L.E. = leading edge
T.E. = trailing edge
w = wall
2 = injected air

INTRODUCTION

Large, LNG burning gas-steam combined cycle power plants with about 10% higher efficiency than the latest turbine thermal plants have been commercially operating in Japan with much success (Sudo et al., 1980). A high temperature heavy duty gas turbine with a 1150°C turbine inlet temperature has been adopted to such combined cycle power plants. To achieve higher efficiency, high temperature heavy-duty gas turbines have been actively developed (Sato et al., 1985, Brandt, 1987) and the turbine inlet gas temperature of the latest one will reach a level of 1300°C.

By increasing the turbine inlet temperature, the turbine vane and blade will be exposed to a high gas stream and subjected more and more to a severe environment (Sato et al., 1986). As there is a temperature distribution into the combustion exhaust gas, the first stationary vanes are put in the most severe thermal condition. Therefore, the first stationary vane for such high temperature gas turbines should adopt a cooling system using an effective convection cooling and film cooling method for not only the vane surface but also the endwall. As the 1st stationary vane needs such an elaborate cooling system, the turbine airfoil design reaches a low aspect ratio and low solidity because of the optimization of performance and manufacturing cost by reducing the number of vanes. In such a low aspect ratio guide vane, 3-D flow field strongly affects the vane and endwall surface heat transfer and film cooling. Therefore, the usual cooling design data based on the 2-D cascade test is insufficient to develop a highly reliable and high performance first vane.

The complicated aerodynamic nature of turbine secondary flows have been studied by a number of investigators. And comprehensive review of such investigations until 1984 is given by Sieverding (1985). However, only minimal investigation of the passage secondary flow effect on the heat transfer and film cooling are available. Blair (1974) investigated the film cooling effectiveness and convective heat transfer coefficient distributions on the endwall of a large-scale turbine passage. Graziani et al. (1980) studied the endwall and blade surface heat transfer in a large scale linear cascade of blades and observed that the inlet boundary layer flows greatly influenced the heat transfer. York et al. (1984) measured the endwall heat transfer coefficient using double layer grid of...
thermocouples by hot cascade. Gaugler and Russell (1984) compared the heat transfer distribution and the visualized secondary flows on an enlarged replica of York's turbine endwall, and directly compared secondary flow with heat transfer distributions. Georgiou et al. (1979) and Dunn et al. (1979) measured the film cooling effectiveness on a turbine endwall and the heat transfer coefficient on a turbine airfoil and endwall by short duration method, respectively. Goldstein et al. (1981)(1987a) investigated the film cooling effectiveness on the turbine blade near the endwall and in literature (1987b) measured the detailed distribution of mass transfer Stanton number on the turbine endwall by using the mass transfer analogy.

Those were the experimental approaches to the investigation of turbine secondary flow effects on heat transfer. Recently, some numerical predictions of three dimensional heat transfer field have been developing (Sharma et al., 1987).

The heat transfer and film cooling influenced by the passage secondary flow is extremely complex, thus further understanding and measurement is necessary. The present experimental work has been done to study the influence of the passage secondary flow on heat transfer and film cooling of the airfoil and the endwall for further understanding and utilization of the cooling design. The tests were carried out using a fully annular, three-dimensional cascade for identifying the radial pressure gradient effects and with low aspect ratio of h/c = 0.5 model vanes.

FLOW FIELD

It is very significant to review briefly the flow field in the turbine passage to understand the effect of the passage secondary flow on the heat transfer and film cooling of the airfoil and the endwall before the discussion of our test results. The fluid mechanism in the passage has been studied by a number of investigators with flow visualization and direct measurements. Here the essential results will be reviewed.

Figure 1 shows the conception of the three-dimensional flow field inside a first stage turbine vane passage (Breugelmans).

A fluid particle in the inlet boundary layer was forced downward by pressure variation at the leading edge-endwall intersection. It then rolls up to generate the so-called horseshoe vortex. The pressure side leg of the horseshoe vortex combines with the low momentum flow near the endwall to form, what is known collectively as the passage vortex. The other leg is convected around the leading edge to the suction surface and remains close to the suction surface till it comes to the separation lines of the endwall boundary layer. At the separation line, suction side leg of the horseshoe vortex lifts off the endwall and grows rapidly downstream along the suction surface adjacent to the passage vortex, the so-called counter vortex. The relative position of the counter vortex and passage vortex depend on the cascade geometry and overall flow conditions. The passages of both legs of the horseshoe vortex and the low momentum flow (crossflow "B") adjacent to the endwall were strongly influenced by the strong pressure gradient across the passage determined by the cascade geometry and the aerodynamic loading.

The separation line, which was formed as the endwall boundary layer approached the turbine vane, and the attachment line, which divides the incoming boundary layer flow entering a vane passage from the flow entering the adjacent passage, are shown in Figure 1. The intersection of these two lines is a so-called saddle point.

The final region to be noticed is the wake region just downstream of the turbine vane trailing edge. The intensity of the nozzle wake of the air cooled turbine vane is stronger compared with that of the non-cooled vane because of the thick trailing edge.

EXPERIMENTAL APPARATUS

The experiment was conducted with a low speed, open circuit fully annular cascade wind tunnel facility of 400mm inner diameter and 550mm outer diameter. Figure 2 shows a schematic diagram of the test apparatus. The inlet velocity was measured by the Pitot tube mounted 35mm in front of the leading edge of the vane. Inlet velocity was 15m/s and exit velocity was 64m/s under typical test conditions. The boundary layer thickness measured at a point 35mm in front of the leading edge of the vane was δ₉₉ = 1.9mm on the inner and outer endwall and the ratio to the vane height was 2.5%. The turbulence intensity of the mainstream was w = 2.7% at the leading edge in the middle channel of the model vane.

The annular cascade was constructed with 13 vanes with an aspect ratio of h/c = 0.5.
The vane chord length and the height is 150.25mm and 75.0 mm respectively. The Reynolds number based on chord length is $6.1 \times 10^5$. This valve is about one tenth of the actual engine condition. The Reynolds and Mach number are low compared with the actual engine condition, but sufficient to understand the three-dimensional flow effect on heat transfer and film cooling. This data is important for verification of the three-dimensional viscous flow code. The vane profile shown in Figure 3 (Nakahara et al., 1981) is the typical 1st stage vane for a heavy duty gas turbine and is the same as the previous two-dimensional film cooling test (Nakahara et al., 1981). The model vane simulates the air cooled turbine vane blowing through the trailing edge, so the trailing edge is thicker than that of the non-cooled vane. The pitch-chord ratio ($p/c$) and the trailing edge radius-slot width ratio ($r/s$) vary 0.644 to 0.885 and 0.0784 to 0.0456 from inner side to outer side respectively.

Six of the thirteen vanes were made of the low thermal conductivity material, Bakelite, to measure the heat transfer and film cooling. The remaining seven vanes were made of metal to measure the aerodynamic conditions. The two flow passages were made of acrylate for visibility into the passage.

The double rows of film cooling holes, whose configuration and dimensions are shown in Figure 3, were positioned at the suction surface and pressure surface on the airfoil, and inclined at a 30° angle to the surface. The configuration and the dimension of the film cooling were also the same as those of the two-dimensional previous cascade test (Nakahara et al., 1981).

On the inner and outer endwall, the film cooling holes were placed at three locations between the leading edge and the nozzle throat as shown in Figure 4 and each location was labeled from I to III. A single row of holes was located near the leading edge and two rows of holes were located at the remaining two positions. The diameter of the injection holes is 1.5mm; they are inclined at a 30° angle to the endwall surface and spaced three diameters apart.

MEASURING METHOD

The local heat transfer coefficient on the airfoil and on the endwall was measured using the airfoil and the endwall without film cooling holes with electrically heated, instrumented, isothermal models. 5μm thick nickel foil, was pressed on a 0.3mm-thick epoxy sheet, and rectangular shaped heaters were made by photoetching the nickel foil. The width of the foil heater used near the leading edge on the airfoil is 2.5mm and that used on the suction and pressure surface is 5.0mm. The narrow gap between each of the heaters was 50μm. To measure the wall temperature, the airfoil test region was instrumented with 134 0.1mm diameter chromel-alumel thermocouples at the location shown in Figure 5.

The embedded surface was polished and the foil heater sheet was adhered with a thin epoxy adhesive. The photograph of the vane heat transfer model is shown in Figure 6. The model vane can be moved radially through slots in the endwalls, so the necessary wall temperature can be measured. In this study the wall temperature of seven sections at a height of 16.7; 33.3; 50.0; 58.3; 66.7; 83.7; 91.7 were measured.

Thin Nickel Foil Heater
The endwall heat transfer foil heater was constructed with 47 5.0mm wide stripes. The 235 0.1mm diameter chromel-alumel thermocouples were instrumented in the inner and outer endwall respectively.

The accuracy of the heat transfer measurement by using such foil heater adhered on the Bakelite was confirmed by the flat plate model under mainstream velocity keeping 15m/s. The results were compared with the well-known flat plate heat transfer equation and they fall within ±10%.

The film cooling test was conducted using a different airfoil and endwall with film cooling holes from the heat transfer test under the conditions in which the mainstream was kept at an ambient temperature and heated air was blown through the film cooling holes.

To get an ideal adiabatic wall, rigid uretan foam was embeded in the Bakelite airfoil and endwall and the adiabatic wall temperature was measured by the embedded 0.1mm diameter chromel-alumel thermocouples in the airfoil and the endwall to derive the following film cooling effectiveness:

\[
\eta_f = \frac{T_{\text{wall}} - T_{\infty}}{T_{\text{wall}} - T_2}
\]

The total number of embedded thermocouples on the airfoil is 192, and 102 on both endwalls.

The infrared pyrometer was also used to measure the wall temperature of the region where the temperature gradient is steep. The surface of the vane and the endwall was painted with thin carbon black paint to obtain a surface with an emissivity approximately equal to one, and the measured temperature by the infrared pyrometer was calibrated by the embedded thermocouples.

**EXPERIMENTAL RESULTS AND DISCUSSION**

**Velocity distribution on the airfoil**

The velocity distribution around the airfoil was calculated by the static pressure distribution measured by using static taps on the airfoil with the inlet velocity being 13.3m/s as shown in Figure 7. It is apparent that the aerodynamic loading is increasing toward the outer side because of the high P/C ratio. The authors checked the velocity distribution at the mean section of the annular cascade with the previous two-dimensional cascade test results (Nakahara et al., 1981) and it was confirmed that the two dimensionality was maintained.

**Flow visualization**

Flow visualization on the airfoil and the inner and outer endwall was made. The surface-streamline flow-visualization technique, which is suitable for use in low-speed wind tunnels, developed by Langston and Boyle, (1982) was applied (Gangler and Russell, 1984). In this test, a matrix of ink dots was not applied, but polyester drafting film was painted uniformly with a felt-tipped pen (Pental OPMNW) containing water-insoluble blue ink and sprayed with wintergreen.

The flow visualization results of the vane and endwall surfaces ink trace by the above mentioned method are shown in Figures 8 to 10. The intent of these results is to point out the predominant aerodynamic characteristics of the flow field on the airfoil and on the endwall which can help to interpret the heat transfer and film cooling. Clearly evident in Figure 8 is that the limiting streamlines converge toward the midspan of the airfoil suction surface. And the separation line immigrates longer on the outer side than the inner side because of high aerodynamic loading on the outer side.
The reason why stagnation of paint occurred at $x/x_{\text{max}} \approx 0.5$ is a laminar bubble seemed to be generated at the de-acceleration region on the suction airfoil surface. On the contrary, airfoil pressure surface limiting streamlines exhibit the same apparent two-dimensional pressure surface flow field.

The horseshoe vortex, endwall crossflow "A", "B" separation line and nozzle wake on the inner and outer endwall are clearly indicated in Figure 9 and Figure 10. When these figures were combined with Figure 8 and the streamline was carefully investigated, we could understand the three-dimensional flow field of the low aspect ratio turbine vane passage. The characteristic separation lines on the endwalls reached the highest velocity point on the suction surface of the vane. The convergent separation line on the suction surface started just from this point and from here, the passage vortex lifted up the endwall and continued downward along the suction surface of the vane. It is clear by comparing Figure 9 and Figure 10 that crossflow "B" is stronger on the outer endwall and the nozzle wake is stronger on the inner endwall, which attributed to the aerodynamic loading and the r/s effect.

Airfoil heat transfer and film cooling

The heat transfer distribution measured on the suction and on the pressure surfaces are shown in Figure 11 and Figure 12 respectively.
The lines in Figure 11 and Figure 12 were plotted using computer graphics according to the measured data, therefore, measured locations did not fit exactly along the lines. By looking at the heat transfer coefficient on the suction surface shown in Figure 11, it can be seen that the heat transfer coefficient of the leading edge and the region between the leading edge till x/xMAX > 0.3 are constant in the height direction. However, after x/xMAX < 0.3, the heat transfer coefficient on the airfoil near the outer endwall increases rapidly compared with that of the mean section. The rapid increase of the heat transfer coefficient in this region is attributed to the passage secondary flow developed along the suction surface near the endwall and this is clear with the flow visualisation test result of the suction surface shown in Figure 8.

The heat transfer coefficient of the suction surface near the inner endwall increased rapidly at x/xMAX < 0.6, but the rate of increase is not so strong as that near the outer endwall. It is considered that near the inner endwall the passage vortex is not so strong as that near the outer endwall, and the intersection point of measuring section with separation line on suction surface of the inner endwall is situated downward to the trailing edge compared with that of the outer endwall.

On the pressure surface, the heat transfer coefficient distribution on the pressure surface is constant in the height direction from the leading edge to the trailing edge except near the leading edge. Separation babble might occur near the leading edge and influence the heat transfer coefficient. From the endwall secondary flow model, it is expected that the three-dimensional flow field has little effect on pressure surface. These heat transfer test results support the evidence.

In this test, the effect of the horseshoe vortex did not appear clearly on the leading edge heat transfer near the endwall.

Next, we will discuss the film cooling test results on the airfoil. The typical film cooling effectiveness distributions associated with coolant injection through the film cooling holes on the suction surface with blowing parameter of M=0.426, typical actual vane condition, of the vane is shown in Figure 13. It is clear from this figure that the film cooling effectiveness decreased rapidly toward the endwall. Especially, the triangular region, where the coolant is swept from the convex surface by the passage vortex system and the film cooling effectiveness decreases to 0, exists near each endwall.

The triangular low film cooling effectiveness zone fits well with the triangular region defined by the separation line and the endwall region in flow visualisation results shown in figure 8.

To evaluate these test results quantitatively, the film cooling effectiveness of a typical suction and pressure surface was plotted against x/MS where S is an equivalent film cooling slot width diffined by \( \frac{D^2}{4P} \) and shown in Figure 14 and Figure 15. The film cooling test was carried out by changing the blowing parameter M in wide range. However, only two test results of M were shown in Figure 14 and Figure 15 because the film cooling effectiveness changed linearly between these two M values. On the suction surface, the film cooling effectiveness decreased more rapidly at 83% height compared with at 23% and mid-height as shown in Figure 14. Another characteristic shown in Figure 14 is that the film cooling effectiveness decreased more rapidly with the higher blowing parameter M near the outer endwall. It was considered that in the triangular region, there is a screwed flow and the film cooling air was swept easily when the coolant penetrated the main flow with the high blowing parameter M.

Fig.14 Variation of Film Cooling Effectiveness on Vane Suction Surface

The same plots were made for the pressure surface and shown in Figure 15. Contrary to the results of the suction surface, the rate of the decrease of the film cooling effectiveness on the pressure surface was not so rapid as that on the suction surface, and that near the endwall was the same as at mid-height. Therefore, the secondary flow effects on the film cooling effectiveness on pressure surface is weak as expected from the flow visualisation.
visualization test results shown in Figure 8. These film cooling effectiveness test values were compared with the results of the previous two-dimensional film cooling study to obtain the film cooling effectiveness value without the influence of the three-dimensional flow field. The authors derived the following experimental equations from the two-dimensional film cooling test (Sato and Takeishi, 1987a,b):

\[ \eta_f = \begin{cases} 
2.8 + 0.027(x/MS) & \text{C} = 1.5/M^{0.5} \quad M < 1 \\
& C = 1.5/M^{0.8} \quad M \geq 1
\end{cases} \]  

The film cooling effectiveness values recorded mid-height on the suction and pressure surface of the full annular cascade fit well with equations (2) and (3).

Endwall heat transfer and film cooling

Typical heat transfer contours of the inner and outer endwall without film cooling holes are shown in Figure 16 and Figure 17. These contour plots show complex and remarkable shapes. On the inner endwall, the heat transfer coefficient increases concentrically around the leading edge. This increase results from the horseshoe vortex formed at the endwall near the vane leading edge. The iso-heat transfer coefficient line of \( a = 130w/m^2K \) travels around the leading edge to the suction surface of the opposite vane. This high heat transfer region was caused by the passage vortex near the suction surface. The heat transfer increases strongly as a result of the suction side corner vortex. Another significant heat transfer increase can be found at the trailing edge region, a result of a strong wake which is clear from Figure 9.

Endwall film cooling effectiveness distributions were measured at the coolant flow rates of \( M = 0.5 \) to 2.5. Typical iso-film cooling effectiveness contour blowing through the film cooling holes I (holes II and III exist without blowing) of the outer endwall is shown in Figure 18. The film cooling effectiveness value is very low near the leading edge of the suction surface. The reason for this is that the horseshoe vortex strongly rolls up the inlet boundary layer flow and the coolant could not remain near the endwall. In the region of crossflow "A", shown in Figure 1, the film cooling effectiveness decreases when the downstream distance from the film cooling hole edge increases. The contour line of \( \eta_f = 0.1 \) near the pressure surface is expected to exist close to the separation line of the passage vortex when Figure 18 was contrasted with the flow visualization test results of Figure 10. The serpentine contour line \( \eta_f = 0.1 \) is also the effect of the passage vortex. The coolant was sent from the pressure surface across the passage to the adjacent suction surface in the crossflow "A" region. The low film cooling effectiveness in the trailing region was caused by the nozzle wake flow.

On the other hand, there is high heat transfer region near the leading edge on the outer endwall caused by the same reason as the inner endwall. However, a high heat transfer zone appears on the suction surface near the leading edge.

This is the result of a highly turbulent flow caused by the reverse pressure gradient turning the upstream boundary layer flow and forcing it to flow toward the suction surface. When Figure 17 was contrasted with Figure 10, we recognized that the crossflow "B" occupies a much larger area and the heat transfer coefficient of the wake region in the trailing edge region is weaker than that on the inner endwall.

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To evaluate the film cooling effectiveness on the endwall qualitatively, film cooling effectiveness blowing through each holes I to III were plotted against the normalized distance from the hole edge along the potential flow lines and shown in Figure 19 to Figure 22. The test was conducted with the wide blowing parameter \( M \), but only typical test results of \( M \) were shown in Figure 19 and Figure 20 to know the effect of location.
Interesting characteristics appear from the film cooling effectiveness when Figure 19 was contrasted with Figure 20. On the outer endwall, the film cooling effectiveness along the potential flow line near the suction surface is 0.035 near the leading edge. It increases to a maximum value of 0.1 when the distance increases because of the supply of coolant by the crossflow "B". The film cooling effectiveness along the potential flow line near the pressure surface decreases rapidly when the distance increases. The reason for this is that the coolant is swept by the crossflow "B" and little coolant remains along the potential flow line. The film cooling effectiveness between these two lines decreases gradually but slightly wavy, when the distance increases.

In comparison with the film cooling on the outer endwall, the same characteristics did not appear in the film cooling effectiveness on the inner endwall as shown in Figure 19. Secondary flow effect on film cooling effectiveness is weaker on the inner endwall than on the outer endwall because of the difference of the aerodynamic loading. The characteristic evidence in Figure 19 is rapid decrease of the film cooling effectiveness downward of the nozzle wake region. However, on the outer endwall, such rapid decrease did not appear in the nozzle wake region. This is caused by the generation of strong nozzle wake as trailing edge radius vs throat width; r's is larger on the inner side. Thick trailing edge must be adopted to maintain cooling flow pass. Cooling designer must take into account such film cooling results.
The measured film cooling effectiveness blowing through the film cooling holes II and III was plotted along the center line of the passage potential flow and shows in Figure 21 and Figure 22 respectively. The coolant blowing through the film cooling holes II and III was governed by the crossflow "B". The film cooling effectiveness decreased at the same rate as when increasing the distance till the nozzle wake region. The rapid decrease of the film cooling effectiveness on the inner endwall in Figure 21 and Figure 22 was the effect of the strong nozzle wake in the trailing edge region.

The Figure 21 and Figure 22 also show that the film cooling effectiveness could be arranged by using the non-dimensional parameter x/MS. This evidence is very useful for film cooling design of the endwall.

CONCLUSIONS

Measurements of local heat transfer coefficient and film cooling effectiveness were made on the endwall and airfoil surfaces of a fully annular guide vane cascade with an aspect ratio of h/C = 0.5. A surface streamline flow-visualization technique was also adopted to understand the secondary effect on heat transfer phenomena in 3-D flow passage. The following conclusions were obtained through these results:

1. Passage secondary flows strongly affect heat transfer and film cooling on the suction surface of the vane and the endwall.
2. The secondary flow has little influence on the heat transfer and film cooling on the pressure surface of the vane.
3. The horseshoe vortex increases the heat transfer and decreases the film cooling effectiveness near the leading edge on the endwall.
4. The heat transfer and film cooling distribution on the endwall shows a very complex pattern dependent on the passage vortex, crossflow "A", "B", and nozzle wake.
5. The demonstrated influence of the passage secondary flow on heat transfer and film cooling is stronger near the outer endwall than near the inner endwall because the aerodynamic loading on the outer side was greater than that on the inner side.
6. Two dimensional heat transfer and film cooling cascade data were applicable for cooling design of the low aspect ratio vane except the triangular zone near the endwall.

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