AERODYNAMIC DESIGN AND ANALYSIS FOR A GAS TURBINE VANE WITH LARGE PITCH/CHORD RATIO

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ABSTRACT
In order to improve the turbine efficiency and reduce the consumption of cooling air, the first stage of an industrial gas turbine was designed. A type of blade and two types of vanes (XA and XB) were further designed. The pitch/chord ratio is 0.81 and 1.05, respectively, and the number of vanes for the whole circle is 40 and 30, respectively. The aerodynamic performance and internal flow of the two type of vanes were analyzed by numerical simulation. The results show that the efficiency of type XB is higher than that of type XA under the design condition. Furthermore cooling design and analysis of the two types of vanes show that the large pitch/chord ratio with smaller whole circle vane number can not only improve the aerodynamic efficiency, but also reduce the cooling air consumption. It has important application value in the design of gas turbine.

INTRODUCTION
As is well known that gas turbine has the advantages of high power density, fast start/stop and low carbon emission. Improving inlet temperature and pressure is one of the developing trend of gas turbine. For example, Mitsubishi Heavy Industry has developed their J Class gas turbine with the inlet temperature of 1600°C. Because the first stage of turbine generally operates in a dreadful environment, cooling technology, material, processing and coating plays an important role in design and manufacturing of industrial gas turbine.

During recent two decades, the researches of turbine blade mainly focus on the effects of various profiles stacking and flow and heat transfer near asymmetric endwalls. Wang proposed the design method for bending blade, and opened up a novel field of studies on turbomachinery (Han and Wang, 1990). Numerous numerical simulations and experiments show that the flow field at the tip of blade can be improved with bending blade, and the losses of secondary flow can be reduced by reasonably arranging the flow of low energy fluid. Wang studied the effects of bending blade on the evolution of vortex in passage, and demonstrated the effects of vortex structure on losses (Wang et al., 1987). Harrison studied the performance of the plane cascade with incident angle of 20° and compound angle of upper and lower endwalls of 30°, and found out that compound bending could reduce the load and loss at blade tip, increase the load and loss at the middle of the blade, thus could not change the total loss of whole blade obviously (Harrison, 1992). Rosic studied the leakage losses of a turbine blade with low aspect ratio. It is found that the compound bending could induce the change of...
spanwise distribution of load but the total loss of whole blade, and the compound bending has not produced dramatic impacts on the leakage losses of rotor blade (Rosic and Xu, 2012). Pullan et al. studied the influence of swept blades on the performance of a gas turbine, and found that the swept blade increased the profile loss, reduced endwall loss, and made the airflow angle at the outlet uniform (Pullan and Harvey, 2006; Pullan and Harvey, 2008). With the convex pressure side and concave suction side, the flow near pressure side and suction side accelerates and decelerates, respectively. Thus, the accelerated and decelerated flows induce a decreasing transverse pressure gradient. Using the decreasing transverse pressure gradient mentioned above, the evolution of secondary flow can be limited effectively as a result. Ingram carried out endwall modeling to decrease the secondary flow loss using manual adjustment and automatic optimization (Ingram, 2003). Miao et al. controlled the secondary flow through arranging ribs on the endwalls (Miao et al., 2015). Wang et al. developed a design method of high load vane with horn shaped passage. Using this method, the efficiency of blade is improved by 0.34%, and the secondary flow as well as horseshoe vortex is eliminated completely (Wang et al., 2017). In the upgrade of Siemens SGT-800 industrial gas turbine, the number of first stage vanes was reduced by 24% to reduce the area of vanes skins and the trailing edges. As a result, the cooling air consumption is reduced effectively, as well as the manufacturing cost and time of the vanes (Wang et al., 2013).

For the first stage vanes of industrial low power gas turbine, the large trailing edge thickness has great influences on aerodynamic performance. However, few researches have been done on these detailed effects. In this paper, focusing on the characteristics of large trailing edge thickness of turbine first stage vane, numerical simulation is applied to analyze the flow field of vanes with different number of vanes and relative pitches, as well as the influences on efficiency and cooling air consumption.

### DESIGN INPUTS AND KEY PARAMETERS

Table 1 shows the design inputs and the key parameters. In this study, the turbine inlet total pressure is 18bar, total temperature 1500°C, static pressure at the stage outlet 9bar, rotation speed 6600r/min, mass flow rate 151kg/s, and the stage efficiency greater than 92.5%. Low reaction can reduce the temperature of rotor inlet, which is beneficial to cooling design and reducing forward axial thrust, but it will reduce the stage efficiency. Moreover, high reaction will bring adverse effects on efficiency, cooling design and axial thrust. Thus, considering cooling design and stage efficiency, the degree of reaction is set to 0.43 in this study. At the given design rotation speed, blade tip velocity is set to 460m/s, considering stage efficiency, centrifugal force, temperature and material, in order to determine the height of the blade tip.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet total pressure (bar)</td>
<td>18</td>
</tr>
<tr>
<td>Inlet total temperature (°C)</td>
<td>1500</td>
</tr>
<tr>
<td>Static pressure at the stage outlet (bar)</td>
<td>9</td>
</tr>
<tr>
<td>Rotation speed (r/min)</td>
<td>6600</td>
</tr>
<tr>
<td>Mass flow rate (kg/s)</td>
<td>151</td>
</tr>
<tr>
<td>Stage efficiency (%)</td>
<td>&gt;92.5</td>
</tr>
<tr>
<td>Degree of reaction</td>
<td>0.43</td>
</tr>
<tr>
<td>Blade tip velocity (m/s)</td>
<td>460</td>
</tr>
</tbody>
</table>

### RESULTING DESIGN

Figure 1 shows the meridian plane of the first stage, which has only a slight expansion at the hub. This design of meridian plane is based on the following considerations. First, with the design pressure ratio and the meridian plane, the Mach number at the blade outlet does not reach the supersonic speed, so the loss is small and the stage efficiency is high. Second, due to the high temperature, the first stage blades need to be cast using single crystal. Moreover, the height of blade, the difficulty of manufacturing and the centrifugal force can be reduced using this meridian plane.
In order to improve the efficiency, the first stage vanes and blades were twisted. The spanwise distributions of flow angle for vane and blade were obtained by S2 stream surface design. The blade profile design should not only meet the requirements of high efficiency and low losses, but also meet the limitations of strength, vibration, cooling and processing. This design process needs multiple iterations. According to the inlet and outlet airflow angles, the appropriate values of chord length, number of vanes, installation angles, maximum profile thickness, leading and trailing edge circular angles were chosen to complete the blade profile design. In this study, one type of blade and two types of vane (namely, XA and XB) were designed, and the corresponding 3D model is shown in Figure 2. The characteristics of the sections at mid-height of the two types of vane are listed in Table 2. It can be seen that there are giant differences in number of vanes, ratio of maximum thickness to chord length and relative pitch between the two types of vane profiles. Considering the needs of cooling, the thickness of trailing edge was set to 4mm. To keep the meridian plane universal in this study, the axial length of the two types of vanes are nearly the same. It is seen that the axial chord length of types XA and XB are the same at the shroud, and the axial chord length of type XB is slightly less than that of type XA at the hub, as shown in Figure 1. Figure 3 shows the comparison between the profiles of sections at mid-height of types XA and XB. It is seen that compared with type XA, type XB has larger leading edge radius and maximum blade thickness.

**Table 2 Characteristics of Section at Mid-height of Vane**

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Type XA</th>
<th>Type XB</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of vanes</td>
<td>40</td>
<td>30</td>
</tr>
<tr>
<td>Inlet airflow angle (°)</td>
<td>90</td>
<td>90</td>
</tr>
<tr>
<td>Outlet airflow angle (°)</td>
<td>17</td>
<td>17</td>
</tr>
<tr>
<td>Maximum thickness/Chord</td>
<td>0.23</td>
<td>0.35</td>
</tr>
<tr>
<td>Installation angle (°)</td>
<td>51.6</td>
<td>54.3</td>
</tr>
<tr>
<td>Relative pitch</td>
<td>0.81</td>
<td>1.05</td>
</tr>
<tr>
<td>Aspect ratio</td>
<td>0.7</td>
<td>0.68</td>
</tr>
<tr>
<td>Thickness of trailing edge (mm)</td>
<td>4</td>
<td>4</td>
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</table>

![Figure 2 Three-dimensional Model](image)

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<tr>
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<td>90</td>
<td>90</td>
</tr>
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<td>Outlet airflow angle (°)</td>
<td>17</td>
<td>17</td>
</tr>
<tr>
<td>Maximum thickness/Chord</td>
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</tr>
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<tr>
<td>Thickness of trailing edge (mm)</td>
<td>4</td>
<td>4</td>
</tr>
</tbody>
</table>

![Type XA](image) ![Type XB](image) ![Type XA vs Type XB](image)

**Figure 3 Comparison of Profiles of Sections at Mid-height of Vane**

**ANALYSIS OF THREE DIMENSIONAL AERODYNAMIC NUMERICAL SIMULATIONS**

**Computational Method**

Following the aerodynamic design, the three-dimensional viscous numerical analysis was carried out for the two models. Commercial software TurboGrid was employed to generate structured grids. The mesh of computational domain is shown in Figure 4. The total number of grid of computational domain is 889000, and the height of the first layer of grid on the wall is 1μm, so that the y+ on the wall does not exceed 1.

The numerical simulations were carried out using ANSYS CFX. The commercial software uses an element-based finite volume method, which first involves discretizing the spatial domain using a mesh. The mesh is used to construct
finite volumes, which are used to conserve relevant quantities such as mass, momentum, and energy. The governing equations for flows in turbine stage in this study were three-dimensional Reynolds-averaged Navier-Stokes (RANS) equations. To close the RANS equations, the shear-stress Transport (SST) turbulence model which combines the advantages of k-ω and k-ε models, and having good accuracy in solving the boundary layer flow with adverse pressure gradient was applied in the computations.

![Figure 4 Mesh of Computational Domain](image)

**Figure 4 Mesh of Computational Domain**

The boundary conditions for the numerical simulations in this study are listed in Table 3. The working fluid is gas, and the change of gas viscosity with temperature was obtained by fitting.

<table>
<thead>
<tr>
<th>Boundaries</th>
<th>Boundary details</th>
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<tbody>
<tr>
<td>Inlet</td>
<td>Total pressure 18 bar</td>
</tr>
<tr>
<td></td>
<td>Total temperature 1500 °C</td>
</tr>
<tr>
<td>Outlet</td>
<td>Static pressure 9 bar</td>
</tr>
<tr>
<td>Hub, shroud, vane, and blade</td>
<td>No slip adiabatic wall</td>
</tr>
<tr>
<td>Rotation speed</td>
<td>6600 r/min</td>
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</table>

**Table 3 Boundary Conditions**

**Experimental Validation**

An experiment was designed to validate the CFD tool. Pressure distribution on a vane skin was investigated. The test system consists of a blower, a wind tunnel and test section. This experiment was conducted in an open loop wind tunnel driven by a blower, as shown in Figure 5 (Li et al., 2014). The main flow from the blower was first expanded by a diffuser to lower the speed before going through a honeycomb rectifier and then passed a contraction duct, a straight section, a turbulence grid before it went into the test section illustrated in Figure 6. The side walls of the section were made of transparent plastic for optic measurement. This tunnel simulates the main flow path. Figure 7 shows the geometry of test vane and the arrangement of testing points in test section.

![Figure 5 Wind Tunnel of The Experiment](image)  
**Figure 5 Wind Tunnel of The Experiment**

![Figure 6 Test Section of The Experiment](image)  
**Figure 6 Test Section of The Experiment**
The parameters of main flow in the experiment are listed in Table 4. With these parameters, simulation of flow around the test vane was carried out using the numerical method introduced in previous section to gain pressure distribution on vane skin. Figure 8 shows pressure at testing points on the test vane obtained by CFD prediction and experiment. It is demonstrated that the values obtained through CFD prediction and measurement are close to each other. The numerical simulation has supported the feasibility of present numerical method.

### Table 4 Parameters of Main Flow

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
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<tr>
<td>Turbulence intensity</td>
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<tr>
<td>Total pressure (Pa)</td>
<td>104646.5578</td>
</tr>
<tr>
<td>Static pressure (Pa)</td>
<td>104220.6219</td>
</tr>
<tr>
<td>Total temperature (K)</td>
<td>304</td>
</tr>
<tr>
<td>Velocity (m/s)</td>
<td>27</td>
</tr>
</tbody>
</table>

Figure 8 Comparisons of Pressure at Testing Points between CFD Prediction and Experimental Results

**Performance Analysis**

The comparisons between computational results and design objectives are listed in Table 5. The deviation of flow rate for the two models is less than 0.25%, the efficiency for all of the two models is higher than the design value, and the efficiency of type XB is the highest, which reaches 93.4%. The degree of reaction for the two models also meets the design requirements.

### Table 5 Comparisons between Computational Results and Design Objectives

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Design objectives</th>
<th>Type XA</th>
<th>Type XB</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow rate (kg/s)</td>
<td>151</td>
<td>151.24</td>
<td>151.34</td>
</tr>
<tr>
<td>Stage efficiency (%)</td>
<td>&gt;92.5</td>
<td>93.16</td>
<td>93.4</td>
</tr>
<tr>
<td>Degree of reaction</td>
<td>0.43</td>
<td>0.431</td>
<td>0.431</td>
</tr>
</tbody>
</table>

The distributions of reaction and vane outlet flow angle along vane height for the two models are shown in Figures 9 and 10, respectively. The distribution of degree of reaction along vane height are the same for the two models, that is, the degree of reaction changes from 0.35 at the root to 0.48 at the tip. It is seen that the degree of reaction changes slowly along the vane height. The reason of this is that the ratio of root diameter to tip diameter of the first stage is large, and thus the aerodynamic parameters do not change dramatically along the radial direction. In the S2 stream surface design, the two types of vanes are designed with equal outlet airflow angles, but the distributions of outlet airflow angle along vane height for the two vanes are different from each other. At 30% relative vane height, the outlet airflow angle of the vane deflects due to the influence of boundary layer, and the change of the outlet airflow angle of vane for type XA is more severe than that of vane for type XB, which indicates that the vortex intensity at the root of vane for type XA is greater. The deflections of the outlet airflow angle of the two models are similar to each other at the vane tip.
The spanwise distributions of Mach number at outlet of vane and blade for the two models are shown in Figures 11 and 12, respectively. The distributions of Mach number at the outlet of vanes for types XA and XB are similar to each other, but the Mach number for type XB is higher, which is mainly due to the less loss for type XB (shown in Figure 13) and thus makes high actual Mach number at the outlet of type XB. The distributions of Mach number at blade outlet for the two models are the same since the two models share a set of blade model. At the blade outlet, the mean Mach number is 0.75, and the maximum Mach number is 0.78. In this study, the distributions of Mach number at the outlet of the designed first stage vane and blade are reasonable, and the average values are close to each other. Moreover, neither of the Mach number mentioned above reaches transonic or supersonic speed, which is beneficial to improve the stage efficiency.

The spanwise distributions of vane efficiency for the two models are shown in Figure 13. During the 30% to 70% vane height, the change of efficiency is slightly small, and the loss in this part is mainly the profile loss. It is seen that the efficiency of type XB is higher than that of type XA, which indicates that the profile efficiency of type XB is higher. An important reason of higher efficiency for type XB is that the number of vanes is reduced comparing type XA, which consequently reduces the wake loss. Due to the need of cooling at trailing edge, the thickness of trailing edge should be large enough, which may induce large wake loss, and thus the thickness is set to 4mm. The changes of entropy increase of vane in streamwise for the two models are shown in Figure 15. It is can be seen that entropy increase of type XA is larger than that of type XB when gas flows through the outlet of vane, which indicates greater wake loss for type XA. According to Reference (Ainley and Mathieson, 1951), the vane efficiency reaches the optimum value when the relative pitch is between 0.4 and 0.8 (the relative pitch of type XB is 1.05). So the value of relative pitch is often chosen in the above range in design. In this study, the understanding of the effects of relative pitch on blade efficiency under the influence of multiple factors is expanded, and a valuable reference for the design of the first stage vane of gas turbine is provided. It can be also seen in Figure 13 that the passage vortices could affect the efficiency in the parts below 30% vane height and above 70% vane height, and cause larger loss at the root. It should be noted that for the two types of models, XA and XB, share the same efficiency in in the parts below 10% vane height and above 90% vane height. The
reason for this may be that the average total temperature and total pressure, but not the accurate spanwise distributions of these parameters for the vane, are given at the stage inlet in the computation. In the range of 10% to 90% blade height, the efficiency of type XB is higher than that of type XA. The comparison of stage efficiency for the two models is shown in Figure 14, and it can be seen that the distributions for the two models are similar to each other. However, the secondary flow has influences on a greater range in spanwise direction. In the range of 10% to 90% blade height, the stage efficiency of the model with vanes of type XB is higher than that with type XA.

In Figure 16, the comparison of static pressure coefficient distribution on sections at some typical blade heights of types XA and XB is shown. Static pressure coefficient is defined as

$$c_p = \frac{p_{s,j} - \overline{p}_1}{\overline{p}_{01} - \overline{p}_1}$$  \hspace{1cm} (1)

where $p_{s,j}$ is the static pressure on blade skin, $\overline{p}_{01}$ total pressure of the main stream on the section behind the vane, and $\overline{p}_1$ static pressure of the main stream on the section behind the vane.

In Figure 16, it is obvious that the loads of the three typical sections on type XB are higher, due to that the number of vanes is 25% fewer than that of type XA. The distributions of load on the three typical sections of XA are reasonable. On each of the typical sections of type XA, the maximum pressure drop is reached at 80% relative chord length, and thus the pressure expansion section is short, which is consistent with the characteristics of uniformly rear-loaded profile. For all of the typical sections on type XB, the maximum pressure drop is achieved at 60% relative chord length, and thus the pressure expansion section is larger than that of type XA, which indicates that type XB is fore-loaded blade. It can be seen in Figure 15 that the entropy increase of type XA is less than that of type XB, from inlet to trailing edge, which indicates that the friction loss, boundary layer separation loss and blade end loss on the skin of type XA are less. So the load distribution on type XA is more reasonable, comparing that on type XB. As mentioned above, type XB has higher efficiency because of the less number of vanes and less wake loss. Therefore, the efficiency of type XB is higher through the comprehensive considerations.
Flow Field Analysis

Comparisons of Mach number and streamline distributions on the section with mid-height of vane are shown in Figures 17 and 18, respectively. It can be seen that the peak Mach number for type XB is higher than that for type XA, the values of peak Mach number for type XB and type XA are 0.996 and 0.889, respectively, and both of them achieve transonic speed. The distribution of Mach number is similar to that of static pressure coefficient shown in Figure 16. The streamline chart shows that the flows around the two vanes are stable and there is no separation, so the vanes are well designed.

Figure 16 Static Pressure Coefficient Distribution on Sections at Three Typical Vane Heights

(a) 10% vane height  (b) 50% vane height  (c) 90% vane height

Figure 17 Comparison of Mach Number Distributions on The Section with Mid-height of Vane

(a) XA  (b) XB

Figure 18 Comparison of Streamline Distributions on The Section with Mid-height of Vane

(a) XA  (b) XB

The comparison of vortex structures based on Q criterion is shown in Figure 19, and the vortex intensity distributions on sections with different chord length are shown in Figure 20. Moreover, Figure 21 illustrates the streamlines near endwalls for types XA and XB. The horseshoe vortex is formed at the leading edge of the vane, consisting of pressure side and suction side branches. At 10% chord length, the horseshoe vortex intensity for type XB is larger than that for type XA, because type XB is a fore-loaded blade, which could produce a large transverse pressure gradient near the inlet. Type XA is a uniformly rear-loaded blade, and the lateral pressure gradient on pressure and suction sides is small. So the development of secondary flow is slow. At 50% chord length, the vortex intensity is extremely weak, due to the continuous acceleration of flow. Then, the pressure side branch gradually migrates to the suction side of the adjacent vane under the transverse pressure gradient between the pressure and suction sides. The passage vortex attaches to the
suction side and develops downstream continuously, and its position rises in spanwise direction and its scale increases continuously. The influence area at the root is larger than that at the tip. At 90% chord length, the passage vortex intensity increases with the influence of the adverse pressure gradient, and the range of vortex for type XB is larger than that of type XA. Also it can be seen in Figure 15 that there is greater entropy increase and loss for type XB. However, the wake flow after outlet of vane XB is more chaotic near the trailing edge, which illustrated in Figure 19. Therefore the wake loss after outlet of XB is higher than XA near the trailing edge. Due to the less number of vanes, circumferential average entropy increase after outlet of type XB is lower, as shown in Figure 15.

![Figure 19 Vortex Structures for Vanes of Types XA and XB (Q=1x10^7)](image)

![Figure 20 Vortex Intensity(Sections at 10%, 50% and 90% Chord Length)](image)
The comparison of vortex intensity distribution in the outlet section is shown in Figure 22. Due to the large thickness of the trailing edge of the vane, a huge wake loss occurs at the outlet of the vane. It can be seen that the scale and intensity of passage vortex at the root is much larger than that at the tip for both the two models, and the intensity and range of the vortex for type XA are higher than that for type XB as well, which is the same as shown in Figure 15. The entropy increase for type XA at the outlet of the vane and the loss are greater.

The two-dimensional heat transfer calculation program TEXSTAN (Weigand et al., 1997) is adopted to carry out the heat transfer computation for the skin of blade. In the computation, two equation turbulence model with transition is introduced. For the leading edge, the formula of flow around cylinder is used instead of TEXSTAN. The distribution of external heat transfer coefficient on the blade skin is shown in Figure 23. The distributions of the heat transfer coefficient on the two vanes skins are close to each other, and the obvious distinction between the heat transfer coefficient distributions for the two vanes only exists on the leading edge, because of the different radius of leading edge.
Analysis of Cooling Air Consumption

Under the conditions that cooling structures, materials and temperature finally achieved of vanes are consistent, the cooling air consumption of the two types of vanes is compared. Considering the high inlet temperature, the comprehensive cooling structures are adopted for the two types of vanes mentioned above. The arrangements of cooling structures for each vane are as follows. The convection cooling is used to cool the interior of the vane, the film cooling is adopted for cooling the skin, the showerhead is taken to cool the leading edge, the column ribs are employed to cool the trailing edge, and cooling air assigned to trailing edge is discharged from the trailing edge slots. Besides, the skin of vane is covered by the TBC.

According to experience, the relationship between cooling efficiency and heat load parameter is as follow,

\[ \eta = a \cdot \ln \left( \frac{m_c C_p}{h A_g} \right) + b \]  

(2)

where \( m_c \) is the mass flow rate of cooling air, \( C_p \) the specific heat capacity at constant pressures, \( h \) the average heat transfer coefficient of blade skin, \( A_g \) the contact area between blade and gas, and \( a \) and \( b \) constants. In this study, the constant \( a \) is set identical for both types of vanes, and so do the constant \( b \) and the cooling efficiency \( \eta \).

The heat exchange areas of a single vane of types XA and XB are 0.02244m² and 0.02417m², respectively. The average heat transfer coefficients of the two types of vanes (namely, XA and XB) are 3621 W/(m²·K) and 3463 W/(m²·K), respectively. For a single vane, the ratio of the cooling air consumption of the two types of vanes is given in Equation (3).

\[ \frac{m_c^{\text{XB}}}{m_c^{\text{XA}}} = \frac{h^{\text{XB}} A_g^{\text{XB}}}{h^{\text{XA}} A_g^{\text{XA}}} = 1.03 \]  

(3)

CONCLUSIONS

In this study, aerodynamic design of the first stage vanes and blades of a gas turbine with inlet temperature in 1500°C (H Class) were carried out. Two types of vanes (XA and XB) and a type of blade were developed, the one-dimensional cooling design was completed, and the analyses on aerodynamics of the designed vanes and blades were taken in detail. Then, some conclusions are drawn as follows.

(1) Both of the two models have high stage efficiency, and the efficiency meets the design requirement. In particular, the stage efficiency of the model with vane of type XA is 93.16%, and that with type XB is 93.4%. Additionally, the mass flow rates of both models meet the design aim.

(2) Types XA and XB have similar distributions of degree of reaction and Mach number. At the root, the flow deflection and vortex loss of type XA are relatively larger than that of type XB.

(3) The reasons for higher efficiency of type XB are revealed. Although type XA has more reasonable load distribution and less friction loss and vortex loss, type XB has less number of vanes and wake loss, which accounts for a large proportion of the total loss. Therefore, the efficiency of type XB is higher through the comprehensive considerations. This provides guidance for turbine blade design and has significant application value.

(4) The detailed flow field analysis shows that the intensity of horseshoe vortex and passage vortex is larger and flow loss is greater for type XB with the fore-loaded mode. As the number of vanes is less, the wake loss of type XB is smaller, and the intensity and influence range of passage vortex are smaller than that of type XA.

(5) One-dimensional cooling analysis shows that the total cooling air consumption of the full circled vanes of type XB is 22.7% lower than that of type XA.
REFERENCES


