Journal of Thermal Science Vol.22, No.2 (2013) 128−134

DOI: 10.1007/s11630-013-0603-4 **Article ID: 1003-2169(2013)02-0128-07**

Flowfield and Heat Transfer past an Unshrouded Gas Turbine Blade Tip with Different Shapes

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This paper describes the numerical investigations of flow and heat transfer in an unshrouded turbine rotor blade of a heavy duty gas turbine with four tip configurations. By comparing the calculated contours of heat transfer coefficients on the flat tip of the HP turbine rotor blade in the GE-E^3 aircraft engine with the corresponding experimental data, the κ-ω turbulence model was chosen for the present numerical simulations. The inlet and outlet boundary conditions for the turbine rotor blade are specified as the real gas turbine, which were obtained from the 3D full stage simulations. The rotor blade and the hub endwall are rotary and the casing is stationary. The influences of tip configurations on the tip leakage flow and blade tip heat transfer were discussed. It's showed that the different tip configurations changed the leakage flow patterns and the pressure distributions on the suction surface near the blade tip. Compared with the flat tip, the total pressure loss caused by the leakage flow was decreased for the full squealer tip and pressure side squealer tip, while increased for the suction side squealer tip. The suction side squealer tip results in the lowest averaged heat transfer coefficient on the blade tip compared to the other tip configurations.

Keywords: Turbine Rotor Blade, Squealer Blade Tip, Tip Leakage Flow, Aerodynamic Loss, Heat Transfer

Introduction

The turbine inlet temperature of modern gas turbines usually exceeds the material melting temperature by hundreds kelvins. The portions of a turbine vane/blade that exposes to hot gas need to be cooled by sophisticated internal and/or external cooling for a reliable and long-lasting operation. There is a clearance between the rotor blade tip and the stationary casing to allow for the blade to rotate and to expand under thermal and centrifugal stresses. Due to the large pressure difference between the pressure side and the suction side of the blade, the hot gas passes through the tip clearance and forms a leakage flow. In addition to aerodynamic losses, the tip leakage flow results in high heat loading near the blade tip region. Cooling configuration in such a region should be arranged carefully in order to cool the region effectively. On the other hand, efforts have been made to decrease the heat loading on the blade tip by using an appropriate tip shape so that the difficulty of cooling the tip region may be reduced. A recessed tip with sealing rims or squealer tip is often used for an unshrouded turbine blade. Studies on the leakage flows over a flat blade tip or the heat transfer coefficients near a squealer tip have been widely reported.

The studies in [1] indicated that the blade tip heat transfer coefficients are basically independent on the relative motion of the blade and the casing. The blade tip heat transfer experiments maybe carried out on stationary cascades if the similarity of the flow is preserved. The

Received: January 2013 Jian-Jun LIU: Professor, PhD
This investigation was supported by the National 973 Program of China through grant number 2007CB210108.

flow and heat transfer measurements on stationary linear cascade simulated the first stage blade tip of a power generation gas turbine were reported in [2]. Similar features of the blade tip heat transfer coefficient distributions for all the cases studied were noticed. Extensive experiments on the stationary linear cascade with the tip section airfoil of the GE-E^3 aircraft engine high-pressure (HP) turbine rotor blade have been carried out [3-7]. The effects of relative tip clearance height, inlet turbulence intensity and tip shapes on the blade tip heat transfer coefficients and flow near the tip region were measured and compared in detail. The experiments showed that by using a squealer tip, the heat transfer coefficient on the blade tip can be reduced. The overall heat transfer coefficient on the tip for the suction side squealer case was lower than other squealer tip arrangements. Experiments of blade tip heat transfer and flow on a stationary linear cascade with moving endwall were also performed [8]. It was found that although the relative motion between the blade tip and the endwall altered the flow and heat transfer at certain locations, the features of heat transfer coefficient distributions for both flat tip and squealer tip were not significantly affected.

Numerical predictions on the blade tip heat transfer and tip leakage flow were also reported. Simulations of tip leakage flow and heat transfer on the GE-E^3 aircraft engine HP turbine rotor blade with flat tip were reported in [9]. The blade tip heat transfer coefficients were computed for the cases with different tip clearance height and compared with the corresponding experimental data. The effects of turbulence models, including the standard k- ε model, RNG standard k- ε model and RSM model, on the calculated results were also investigated. Calculations of blade tip heat transfer for both a flat tip and a squealer blade tip by using SST turbulence model were performed in [10]. Good agreements between the computed blade tip heat transfer coefficients and the measured data for three cases with different tip clearance height were demonstrated. The rotation effect on the blade tip heat transfer was investigated in [11]. The results showed that the averaged heat transfer coefficient on the blade tip was increased due to the relative motion of the endwall.

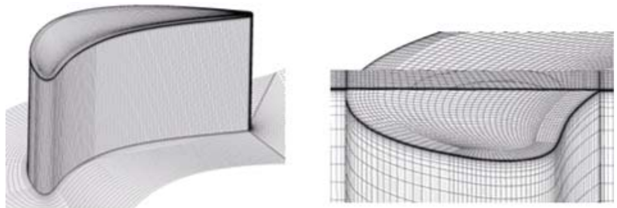
A literature survey on turbine blade tip heat transfer and cooling can be found in [12]. It was noticed that many previous studies focused on the effects of tip clearance height or tip configurations on the tip leakage flow and blade tip heat transfer. Only a few papers investigated both aerodynamic losses and blade tip heat transfer for the different blade tip shapes. In the present paper, numerical simulations for an industrial gas turbine rotor blade with four typical blade tip configurations were performed. The effects of blade tip shapes on the tip leakage flow pattern, blade tip heat transfer coefficient and aerodynamic loss were presented and discussed.

Following this brief introduction, the numerical methods used for the present simulations, the selection of an appropriate turbulence model are described. The computational mesh and boundary conditions used for the simulations are outlined. Results and discussions are presented by comparing the computed flow fields in the blade tip region, blade tip heat transfer coefficient distributions and total pressure losses for the different blade tip configurations. The paper ends with conclusions.

Numerical Method and Validation

The 3D fully implicit coupled Navier-Stokes flow solver CFX is employed for the present numerical simulations. This solver is based on finite-volume method, and the chosen discretization scheme is second-order accurate. It is assumed that the flow in the studied blade row is steady. There are a number of turbulence models available for the RANS calculations, including various *k-ε* turbulence models, *k-ω* model, and SST model.

In order to investigate the sensitivity of turbulence models on the blade tip heat transfer and the flow near the tip regions, and also to validate the flow solver, the calculations for the test case of stationary linear cascade modeled by using the tip section airfoil of GE-E^3 engine HP turbine rotor blade were carried and compared with the corresponding experimental data reported in [13]. The flat tip case with 1.5% tip clearance height was chosen for the present turbulence model sensitivity study and flow solver validation.



(a) (b)

Fig. 1 Computational mesh used for the calculations of stationary linear cascade: (a) mesh on blade surface and endwall, (b) close view of the mesh near tip clearance.

The calculations were carried out for one blade passage. Structured hexahedral mesh was used. The mesh on the blade surface and hub endwall is shown in Fig.1a. The close view of the mesh near the tip clearance was shown in Fig.1b. There are about 10 layers of mesh within the boundary layer in order to well resolve the flow near solid walls. The total mesh elements are of 1.3 million.

According to the experiments in [13], the main flow boundary conditions similar to that in [10] were used for the present simulations. Non-slip wall condition was applied to all of the solid walls. The hub endwall was specified as adiabatic.

In the calculations, k-ω turbulence model and SST turbulence model were used. Convergent solutions were reached while the residuals are less than 1×10^{-5} and the mass flow rate error between the cascade inlet and outlet is less than 0.01%. Figure 2 shows computed and measured inlet total pressure to local static pressure ratio (p_0/p) distributions and blade tip heat transfer coefficient distributions respectively.

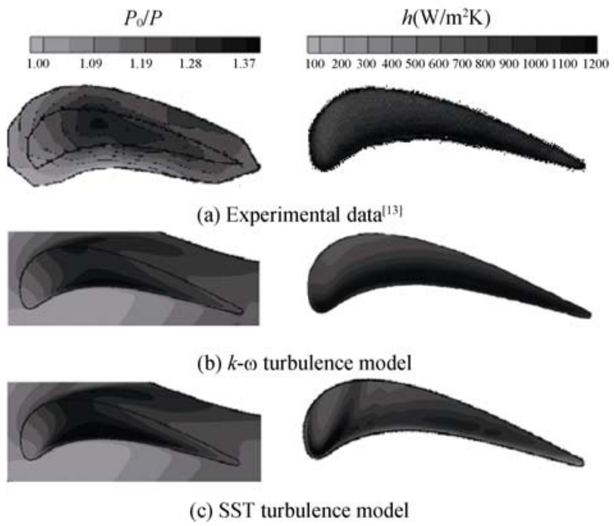


Fig. 2 Calculated and measured pressure ratio p_0/p contours and blade tip heat transfer coefficient h contours.

Since the inlet total pressure p_0 is constant, high ratio p_0/p implies low local static pressure or high local velocity, and vice versa. Both calculated results and experimental result show the acceleration process of tip leakage flow from the blade leading edge to about 1/3 blade chord and the deceleration afterwards. The pattern of pressure ratio (p_0/p) distributions calculated by using k-ω turbulence model is similar to that using SST turbulence model.

Calculations well predicted the features that blade tip heat transfer coefficient decreases from the pressure surface to the suction surface. The values of heat transfer coefficient calculated by using k-ω model well agree with the experiments, as shown in Fig.2b. In contrast, the results calculated by using SST model are lower than the experiments after 20% blade chord, as shown in Fig.2c. It can be seen from the casing pressure ratio distributions and blade tip heat transfer coefficient distributions that k-ω turbulence model behaves better than the SST turbulence model. The k-ω turbulence model is therefore chosen for the following calculations.

Calculation Setup

The blade tip heat transfer and tip leakage flow of the unshrouded first stage turbine rotor blade of a heavy duty gas turbine with different tip shapes were investigated. The tip clearance height for this rotor blade is 1% blade height. Four blade tip configurations as shown in Fig.3 were considered. The squealer height is 2% blade height.

The flow in one rotor blade passage was simulated and periodic boundary conditions were applied to the corresponding periodic boundaries. Structured hexahedral mesh was used in the simulations. The meshes were refined near the solid walls such as blade surfaces and hub/casing endwall and in the tip clearance to well resolve the flow details. The mesh on the blade surface and hub endwall for the full squealer case is shown in Fig.4. The close view of the mesh for the squealer near the blade trailing edge is also shown in this figure. The total mesh elements for the flat tip case and the squealer cases are of 1.25 million and 1.5 million, respectively.

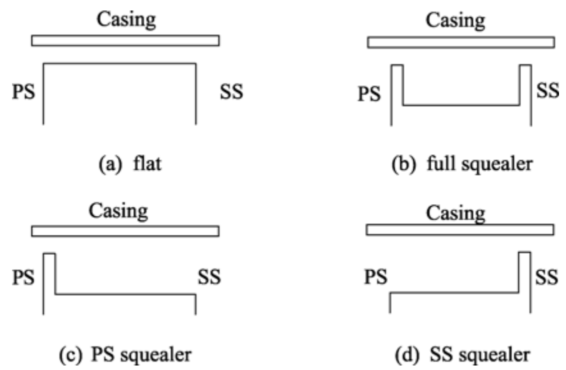


Fig. 3 Four blade tip configurations used for the present investigations.

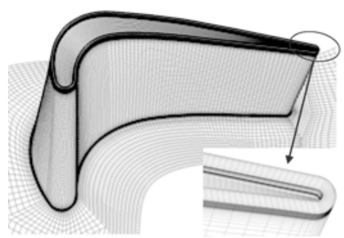


Fig. 4 Computational mesh for the heavy duty gas turbine rotor blade with full squealer tip.

The boundary conditions are taken from the 3D full turbine stage simulations. At the rotor blade inlet, mass flow rate, total temperature, flow angles and turbulence intensity were specified. At the outlet the averaged static pressure was specified. Non-slip condition was applied to all of the solid walls. Isothermal boundary condition (957K) was applied to the rotor blade surface. Adiabatic boundary condition was set on the hub endwall and casing. The casing is stationary. The rotor blade and the hub endwall are rotating with a speed of 3000 rpm. The above mentioned numerical method, turbulence model and convergence criteria were used for the calculations.

Results and Discussions

Tip Leakage Flow

Blade tip shape considerably affects the tip leakage flow patterns. Figure 5 shows streamlines near the tip clearance and stream traces on a typical cross section for the four different tip shapes. For the flat tip case shown in Fig.5a, a small separation bubble is formed on the tip top near the blade pressure surface due to the entrance effect. After passing the separation bubble, the leakage flow reattaches to the blade tip, goes through the tip clearance and forms the tip leakage vortex while leaving the tip gap.

For the full squealer case shown in Fig.5b, a separation vortex is formed in the cavity near the pressure surface squealer. The separation vortex in the cavity develops from the blade leading edge to trailing edge and occupies the full cavity after about 1/3 blade chord. The tip leakage flow is pushed by the separation vortex towards the casing and the formed leakage vortex is similar to that in the flat tip case.

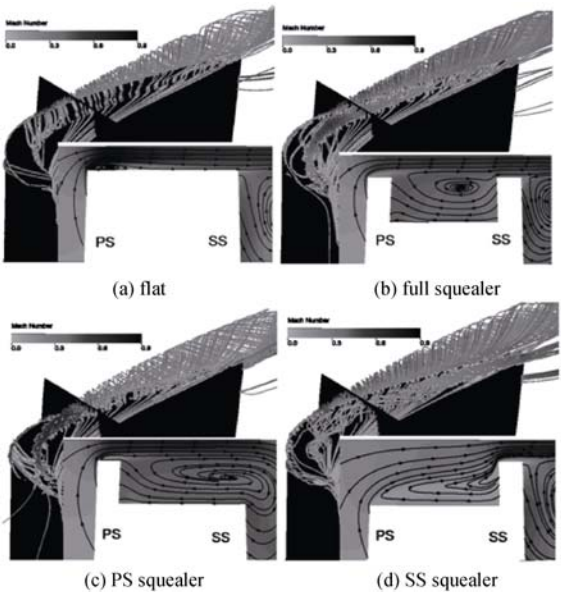


Fig. 5 Streamlines in the tip clearance region and stream traces on the tip gap cross section indicated by the plane.

For the PS squealer case shown in Fig.5c, a separation vortex is also formed on the blade tip top near the pressure surface squealer from the blade leading edge, similar to that in the full squealer case. Due to the absent suction surface squealer in this case, the separation vortex gradually leaves the tip gap with its development from blade leading edge to downstream, and joins the leakage vortex.

For the SS squealer case shown in Fig.5d, a small separation bubble is also formed on the tip top near the blade pressure surface due to the entrance effect, similar to that in the flat tip case. A larger separation vortex is generated and occupies most of the tip clearance with the development of the separation vortex. The leakage flow passes the vortices, leaves the clearance between the suction surface squealer and the casing and forms the tip leakage vortex. Compared with the full squealer case and the PS squealer case, the velocity of the separation vortex for the SS squealer case is relatively low.

Aerodynamic Parameters

Blade tip shape also affects the aerodynamic parameters such as blade surface pressure distributions, total pressure losses and downstream flow angle distributions.

Figure 6 shows the blade surface pressure ratio p/p_0 distributions along the axial chord at 50% and 97% blade height for the four tip configurations studied. Parameter p refers to the local static pressure and p_0 refers to the averaged relative total pressure at the blade inlet. It can be seen in Fig.6a that the change of tip shape rarely affects the static pressure distributions at the 50% blade height,

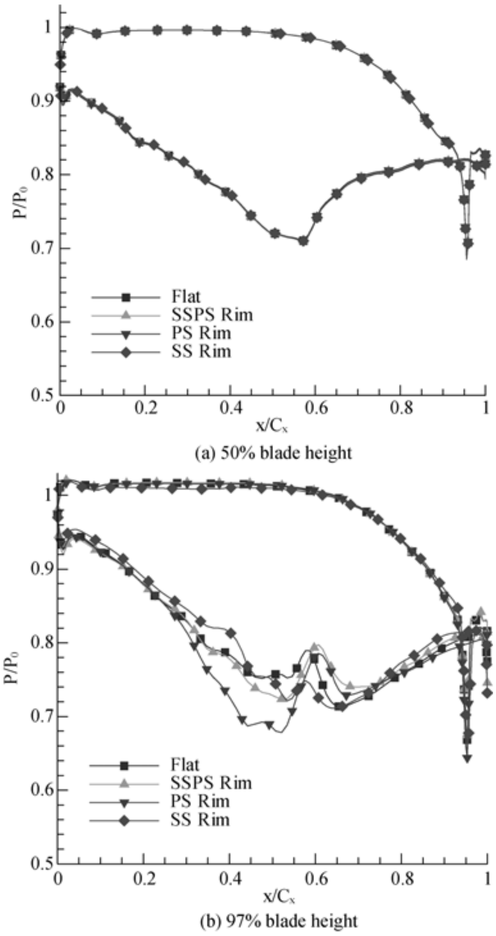


Fig. 6 Static pressure distributions on the blade surface at 50% and 97% blade height for the four blade tip configurations.

which is far away from the blade tip and the tip leakage flow. In contrast, the change of tip shape strongly influences the static pressure distributions at 97% blade height. The large differences of static pressure distributions mainly appear on the suction surface in the region from 30% to 70% axial blade chord, as shown in Fig.7b, where the tip leakage vortex was formed and further developed.

The total pressure loss distributions at the blade exit, i.e., 100% axial blade chord downstream the trailing edge, are also compared. Total pressure loss coefficient c_{tpl} is defined as $c_{tpl} = (p_0\text{-}p_t)/p_0$, where p_0 is the averaged relative total pressure at the blade inlet and p_t is the local relative total pressure.

Figure 7 shows the total pressure loss coefficient contours for the four tip configurations in the blade tip region at the blade exit. The areas of high total pressure loss due to the tip leakage vortex and horse shoe vortex can be clearly seen. In the blade tip region, the total pressure loss caused by the tip leakage vortex is much larger than that by horse shoe vortex. Compared with the flat tip, the full squealer tip weakens the tip leakage flow and reduces the total pressure loss. The pressure surface squealer also reduces the total pressure loss due to the tip leakage flow but enlarges the high loss area due to the horse shoe vortex. The suction surface squealer changes the loss distributions very slightly.

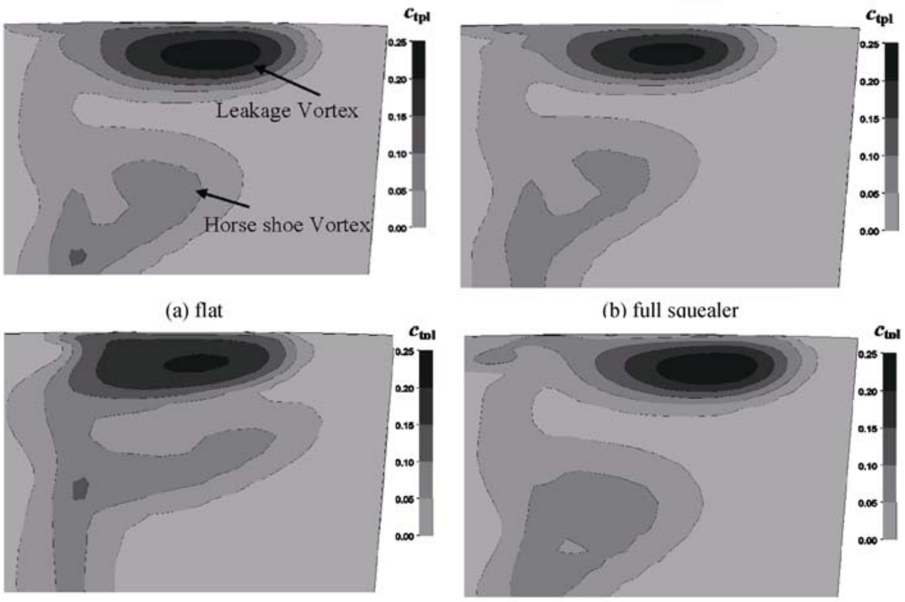


Fig. 7 Contours of total pressure loss coefficient for the four blade tip configurations in the tip region at the blade exit.

The comparisons of averaged total pressure loss coefficient at the blade exit are shown in Fig.8. The averaged total pressure loss for the full squealer case is the lowest and it is the highest for the PS squealer case due to the increase of horse shoe vortex loss. The total pressure loss for the SS squealer case is comparable to the flat tip case.

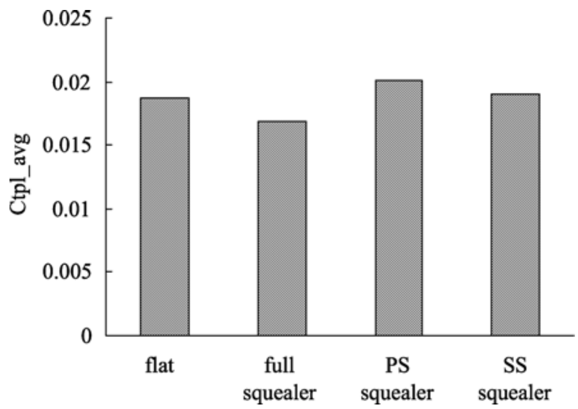


Fig. 8 Comparisons of averaged total pressure loss coefficient at the blade exit for the four blade tip configurations.

Blade Tip Heat Transfer Coefficient

Blade tip shape certainly affects the tip heat transfer coefficient distributions. Heat transfer coefficient h is defined as $h = q_w/(T_w - T_0)$, where q_w refers to wall surface heat flux, T_w is the wall temperature and T_0 is gas total temperature at the blade inlet.

Figure 9 shows the contours of heat transfer coefficient on the blade tip for the four blade tip configurations. For the full squealer, PS squealer and SS squealer cases, the contours on the squealer are also showed. For the flat tip case shown in Fig.9a, the heat transfer coefficient distributions are generally high and about uniform. The regions of low heat transfer coefficients appear near the leading edge suction surface and in the region on the blade top along the pressure surface where the velocity of leakage flow is low.

For the full squealer case shown in Fig.9b, the distributions of heat transfer coefficient are obviously non-uniform. One of the high heat transfer coefficient regions appears on the fore part of the cavity bottom where the leakage flow impinges on the cavity bottom after passing over the separation vortex near the pressure surface squealer. The other high heat transfer coefficient region appears on the mid part of the suction surface squealer where the upstream leakage flow impinges on. The heat transfer coefficients in regions that are not impinged by the leakage flow are relatively low, especially on the aft part of the cavity bottom.

Similar heat transfer coefficient distributions are found for the PS squealer case as shown in Fig.9c. Without the block of the suction surface squealer as in the full squealer case, the high heat transfer coefficient region on the fore part of the cavity bottom for the PS squealer case is slightly enlarged.

For the SS squealer case shown in Fig.9d, the region of high heat transfer coefficient on the fore part of the cavity bottom is minished. The heat transfer coefficients on the squealer top are still high.

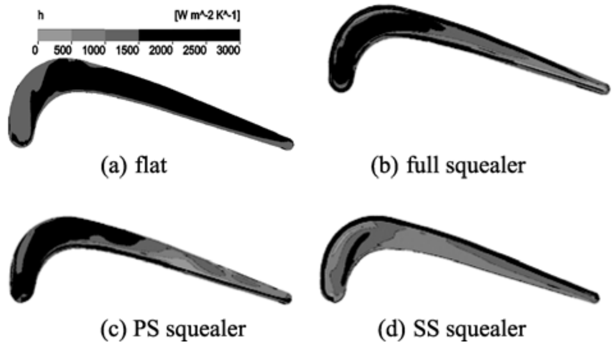


Fig. 9 Contours of heat transfer coefficient on the blade tip for the four blade tip configurations.

The comparisons of averaged heat transfer coefficients on the blade tip top are shown in Fig.10. Excluding the squealer part, the averaged heat transfer coefficient for the flat tip case is the highest. And the averaged heat transfer coefficients for the full, PS and SS squealer case are respectively 86%, 88% and 61% of the flat tip case. For the full blade tip top, i.e., including the squealer part, the averaged heat transfer coefficient for the flat tip case is still the highest. The averaged heat transfer coefficients in this situation for the full squealer case, PS squealer case and SS squealer case are respectively 88%, 86% and 79% of the flat tip case.

Figure 11 shows the contours of heat transfer coefficient on the blade suction and pressure surface for the four tip configurations. It is generally recognized that the heat loads on blade leading edge, trailing edge and suction surface are high. It can be seen in Fig.11a that heat transfer coefficients on the suction surface near the tip region are even higher than that on the leading edge, trailing edge and suction surface, due to the tip leakage flow. The blade tip configurations also influence the heat transfer coefficient distributions on the suction surface near the blade tip by affecting the formation and the structure of the tip leakage flow. In contrast, the differences of heat transfer coefficient distributions on the pressure surface for the four blade tip cases are negligible, as shown in Fig.11b.

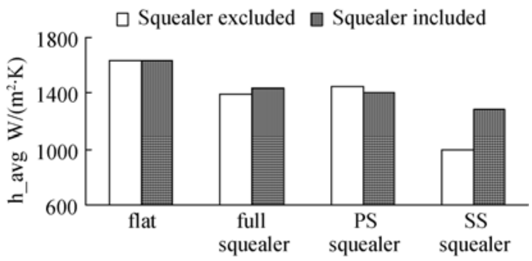


Fig. 10 Comparisons of averaged heat transfer coefficient on the blade tip top for the four blade tip configurations.

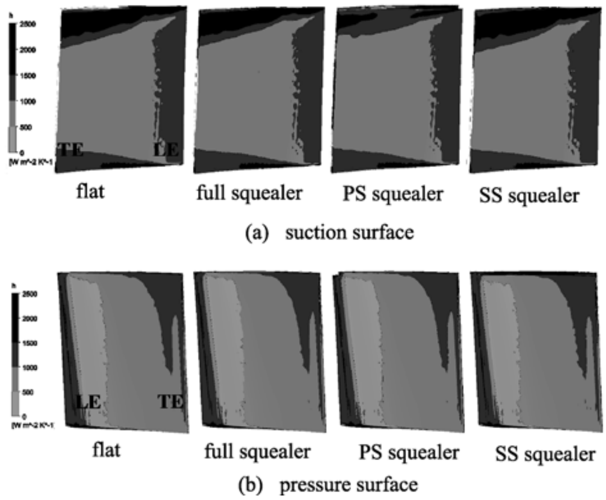


Fig. 11 Contours of heat transfer coefficient on the blade suction and pressure surface for the four blade tip configurations.

Conclusions

The flow and heat transfer in the first stage turbine rotor blade of a heavy-duty gas turbine with four tip configurations have been presented and discussed. The main conclusions are as follows,

The blade tip shape influences the tip leakage flow and blade aerodynamic losses. The full squealer tip results in the lowest total pressure loss and the PS squealer tip results in the highest total pressure loss. The total pressure loss for the SS squealer case is comparable to that of the flat tip case.

All of the squealer tip configurations studied can reduce the averaged heat transfer coefficient on the blade tip, but increase the non-uniformity of heat transfer coefficient. The averaged blade tip heat transfer coefficient for the SS squealer case is the lowest. The heat transfer coefficients on the top of the pressure surface squealer are high.

The tip clearance for a rotor blade leads to tip leakage flow and an extremely high heat transfer coefficient region on the suction surface near the blade tip due to the tip leakage vortex. Special attention should be paid to such a region in the blade cooling design.

Acknowledgement

This investigation was supported by the National 973 Program of China through grant number 2007CB210108.

References

[1] R. E. Mayle, D. E. Metzger: Heat Transfer at the Tip of an Unshrouded Turbine Blade. Proceedings of the Seventh International Heat Transfer Conference, September 6–10, 1982, Munich, West Germany, Vol.3, pp87–92, (1982).

[2] R. S. Bunker, J. C. Bailey, A. A. Ameri: Heat Transfer and Flow on the First Stage Blade Tip of a Power Generation Gas Turbine, Part 1: Experimental Results, NASA/CR 1999-209152, (1999).

[3] Gm. S. Azad, J. C. Han, S. Teng: Heat Transfer and Flow Distribution on a Gas Turbine Blade Tip, ASME Paper No. GT2000-194, Proceedings of ASME TURBO EXPO 2000, May 8-11, 2000, Munich, Germany, (2000).

[4] Gm. S. Azad, J. C. Han, R. Boyle: Heat Transfer and Flow on the Squealer Tip of a Gas Turbine Blade. ASME Journal of Turbomachinery, Vol.122, pp725–732, (2000).

[5] Gm. S. Azad, J. C. Han, R. S. Bunker, C. P. Lee: Effect of Squealer Geometry Arrangement on a Gas Turbine Blade Tip Heat Transfer. ASME Journal of Heat Transfer, Vol.124, pp452–459, (2002).

[6] J. S. Kwak, J. C. Han: Heat Transfer Coefficients on the Squealer Tip and Near Squealer Tip Regions of a Gas Turbine Blade, ASME Journal of Heat Transfer, Vol.125, pp669–677, (2003)

[7] J. S. Kwak, J. Ahn, J. C. Han, C. P. Lee, R. S. Bunker, R. Boyle, R. Gaugler: Heat Transfer Coefficients on the Squealer Tip and near Tip Regions of a Gas Turbine Blade with Single or Double squealer. ASME Paper No. GT2003-38907, Proceedings of ASME TURBO EXPO 2003, June 16-19, 2003, Atlanta, Georgia, USA, (2003).

[8] P. Palafox, M. L. G. Oldfield, P. T. Ireland, T. V. Jones, J. E. LaGraff: Blade Tip Heat Transfer and Aerodynamics in a Large Scale Turbine Cascade With Moving Endwall, ASME Journal of Turbomachinery, Vol.134, pp021020.1-11, (2012).

[9] H. Yang, S. Acharya, S. V. Ekkad, C. Prakash, R. Bunker: Flow and Heat transfer Prediction for a Flat Tip Turbine Blade, ASME Paper No.GT2002-30190, Proceedings of ASME TURBO EXPO 2002, June 3-6, 2002, Amsterdam, The Netherlands, (2002).

[10] F. Mumic, D. Eriksson, B. Sunden: On the Prediction of Tip Leakage Flow and Heat Transfer in a Gas Turbine, ASME Paper No.GT2004-53448, Proceedings of ASME TURBO EXPO 2004, June 14–17, 2004, Vienna, Austria, (2004).

[11] D. L. Yang, X. B. Yu, Z. P. Feng: Investigation of Leakage Flow and Heat Transfer in a Gas Turbine Blade Tip with Emphasis on the Effect of Rotation, ASME Paper No. GT2008-51215, Proceedings of ASME TURBO EXPO 2008, June 9-13, 2008, Berlin, Germany, (2008)

[12] B. Sunden, G. Xie: Gas Turbine Blade Tip Heat Transfer and Cooling: A Literature Survey, Heat Transfer Engineering, Vol.31, No.7, pp527-554, (2010).

[13] J. S. Kwak, J. C. Han: Heat Transfer Coefficient on a Gas Turbine Blade Tip and Near Tip Regions, AIAA Paper No. 2002-3012, Proceedings 8th AIAA/ASME Joint Thermophysics and Heat Transfer Conference, 24-26 June 2002, St. Louis, MO, USA, (2002).