Film Cooling Performance on the Trailing Edge Cutback of Turbine Blade with Various Slot Inner Angles

Mingfeng Chen
College of Mechanical and Power Engineering, Guangdong Ocean University
No.40 East Jiefang Road, Zhanjiang, Guangdong 524025, PRC
mingfeng_chen@163.com

Changming Ling*
* Corresponding author
College of Mechanical and Power Engineering, Guangdong Ocean University
No.40 East Jiefang Road, Zhanjiang, Guangdong 524025, PRC
+86 0759 2382174
ling-cm@163.com

Yuwen Zhang
Department of Mechanical and Aerospace Engineering, University of Missouri
E2412A Lafferre Hall, Columbia, MO 65211, USA
+1-573-884-6936
zhangyu@missouri.edu

ABSTRACT

In order to study the impact of the slot inner angle on the film cooling performance of the trailing edge cutback for turbine blade, physical model and the three-dimensional mathematical model were established. The temperature distribution on the pressure side close to the wall was obtained through numerical simulation method, and the adiabatic temperature difference ratio and actual temperature difference ratio on the trailing edge cutback with various slot inner angles were analyzed. The results showed that, as slot inner angle increases, the average adiabatic temperature difference ratio on the trailing edge cutback and the effect of thermal insulation of gas film are increased. The impact of actual temperature difference ratio on the trailing edge cutback suffers slot inner angle is small, and it is conducive to lower the wall temperature of trailing edge when reduces the slot inner angle. For the study of film cooling performance on the trailing edge cutback of turbine blade, in addition to evaluate the adiabatic temperature difference ratio, it should also assess the actual temperature difference ratio.

CCS Concepts

• Applied computing→Physical sciences and engineering→ Engineering→Computer-aided design.

Keywords

Turbine blade; Trailing edge cutback; Slot inner angle; Film cooling; Numerical simulation.

1. INTRODUCTION

The aerodynamic characteristics of a turbine blade can improved as the thickness of trailing edge of turbine blade decreases [1, 2], and the trailing edge cut back structure as shown in Fig. 1 is adopted to make the thickness as thin as possible [3, 4]. The film cooling performance of the trailing edge cutback is mainly impacted by the slot geometry and the eject flow [5]. Research on trailing edge cutback film cooling has been carried out by some scientists. For example, Sivasegaram and Whitelaw [6] and Burns and Stollery [7] studied trailing edge cutback film cooling through two-dimensional numerical simulation in 1969. Martini et al. [3, 4] studied the trailing edge film cooling by circular coolant wall jets ejected from a slot with internal rib arrays based on both experimental and numerical methods. Ling et al. [8] also numerically analyzed the influence of the ratio of length to width, blowing ratio, and Reynolds number on trailing edge cutback film cooling on the pressure side. Zhu and his co-workers [9, 10] investigated the effects of ratio of slot width to rib width and ratio of lip thickness to slot height on the cooling effectiveness based on both experimental and numerical methods. Shortly after, Zhu et al. [11] studied the influence of blowing ratio and Reynolds number on the cooling effectiveness of trailing edge cutback through experiment method. Yuan et al. [12] adopted liquid crystal measurement technology to investigate the effects of blowing ratio on heat transfer characteristics of the trailing edge cutback.

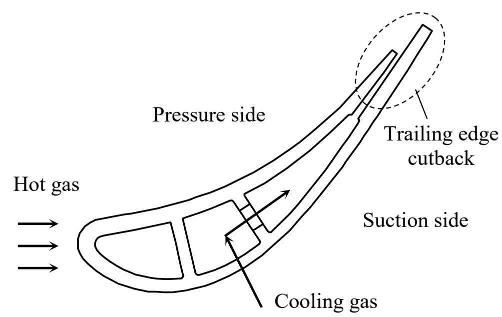

Figure 1. Schematic of turbine blade.

There are two different kinds of wall temperatures on the pressure side of the trailing edge cutback that have been used to judge the film cooling performance: one is adiabatic temperature, and the other one is actual temperature. When adopting the adiabatic temperature [3, 4, 7, 9-13], a dimensionless criterion number called adiabatic temperature difference ratio is used to represent the effect of insulation to the hot air by coolant flow. On the other hand, when adopt actual temperature [8, 14], another dimensionless criterion number called actual temperature difference ratio not only reflect the insulation effect of coolant flow at pressure side, but also reflect the heat conduction effect between pressure side and suction side. However, researches on previous studies mainly focus on the adiabatic temperature difference ratio, less literature about the actual temperature difference ratio. Meanwhile, the comparative analysis between the two difference temperature difference ratios has never reported.

ICCMS 2018, January 8–10, 2018, Sydney, Australia

ACM ISBN 978-1-4503-6339-6/18/01…$15.00
https://doi.org/10.1145/3177457.3191706

In this paper, the physical model and three-dimensional mathematical model of a turbine blade trailing edge cutback are established and a numerical investigation was carried to study the influence of slot inner angles on trailing edge cutback film cooling. The effects of slot inner angles on adiabatic temperature difference ratio and actual temperature difference ratio are discussed.

2. PHYSICAL AND MATHEMATICAL MODEL

2.1 Physical model

A schematic view of the trailing edge cutback of turbine blade is shown in Fig. 2(a). The angle that formed by the upper and lower surfaces of the slot is called slot inner angle which shown in Fig. 2(b). In this paper, the computational model has magnified 20 times base on the prototype of the trailing edge cutback according to the similarity principle, and pressure side wall and suction side wall were considered as plane walls. In order to obtain numerical results close to the actual cases, three rows of staggered pin fins are placed on the inner cooling passage of turbine blade, and the material of turbine blade is 87Al-13Si aluminum alloy. Only half of the slot and half of the rib are treated as computational domain due to symmetry.

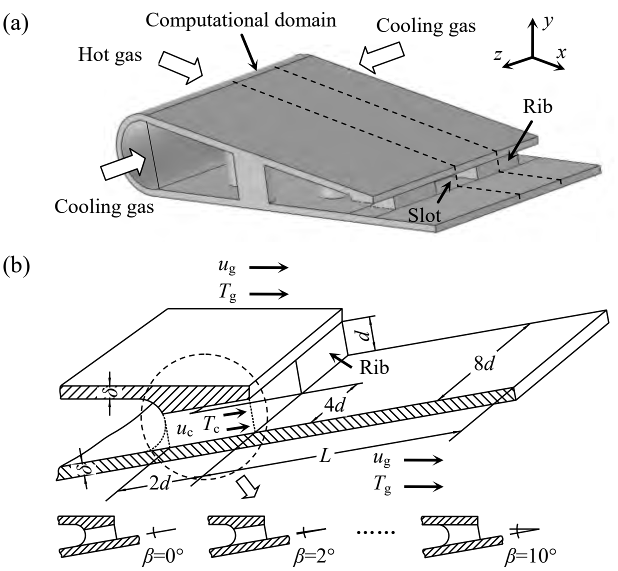

Figure 2. (a) 3D model for trailing edge of turbine blade; (b) Schematic of trailing edge cutback.

Several assumptions have been made to simplify the analysis which are lists as follows: 1) working fluid is incompressible air with constant property; 2) the inlet parameters of fluid is kept unchanged; 3) no-slip fluid flow at solid wall; 4) radiation heat transfer between solid walls is not considered. Numerical simulation parameters are shown in table 1.

An additional 1.5 times of blade length is placed before the inlet of computational domain to make sure that the fluid is fully developed, and another 5 times of blade length is placed after the outlet of computational domain to avoid backflow phenomenon in the fluid outlet boundary [15]. Moreover, 5 times of blade height is placed in the upper and lower of computation domain, respectively. The actual computational domain is shown in Fig. 3.

Table 1. Numerical simulation parameters.

Slot height d/mm	**Cutback length L/mm**	**Wall thickness δ/mm**	**Blade turning angle α/ °**
10	60	5.6	10
Hot gas temperature T_g/K	**Cooling gas temperature T_c/K**	**Blowing ratio M**	**Reynolds number Re_g**
363	303	1.0	3×105

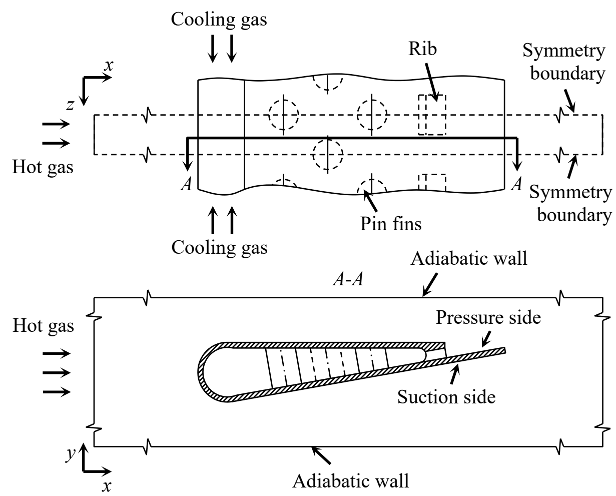

Figure 3. Computational domain.

2.2 Governing equations

The three-dimensional, steady-state, incompressible, turbulent flow and heat transfer problem in this study can be analyzed by employing the standard k-ε model, which can be expressed as follows [16]:

Continuity equation:

$$\frac{\partial}{\partial x_i}\left(\rho u_i\right)=0 \tag{1}$$

Momentum equation:

$$\frac{\partial}{\partial x_i}\left(\rho u_i u_j\right)=\frac{\partial}{\partial x_i}\left[\left(\mu+\mu_t\right)\frac{u_j}{\partial x_i}\right]-\frac{\partial p}{\partial x_j} \tag{2}$$

Energy equation:

$$\frac{\partial}{\partial x_i}\left(\rho u_i T\right)=\frac{\partial}{\partial x_i}\left[\left(\frac{\mu}{Pr}+\frac{\mu_t}{\sigma_T}\right)\frac{\partial T}{\partial x_i}\right] \tag{3}$$

Turbulent kinetic energy equation:

$$\frac{\partial}{\partial x_i}\left(\rho u_i k\right)=\frac{\partial}{\partial x_i}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_i}\right]+G_k-\rho\varepsilon \tag{4}$$

Specific dissipation rate equation:

$$\frac{\partial}{\partial x_i}\left(\rho u_i \varepsilon\right)=\frac{\partial}{\partial x_i}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_i}\right]+\frac{\varepsilon}{k}\left(C_{1\varepsilon}G_k-C_{2\varepsilon}\rho\varepsilon\right) \tag{5}$$

where the turbulent viscosity μ_t and the production of turbulent viscosity Gk are given as:

$$\mu_t = \rho C_\mu k^2 / \varepsilon \tag{6}$$

$$G_k = \mu_t \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \tag{7}$$

The empirical constants in k-ε model are given in Table 2.

Table 2. Empirical constants in k-ε model.

σ_T	σ_k	σ_ε	C_μ	$C_{1\varepsilon}$	$C_{2\varepsilon}$
0.85	1.0	1.3	0.09	1.44	1.92

2.3 Boundary conditions

Inlet boundary [17]:

$u = \text{const}$, $v = w = 0$, $T = \text{const}$, $k = 1.5u^2I^2$,

$$\varepsilon = C_\mu^{3/4} k^{3/2} / (0.07 d_e) \tag{8}$$

where I is turbulence intensity which can be expressed as $I = 0.16 Re^{1/8}$, d_e is hydraulic diameter.

Symmetrical boundary:

$$\frac{\partial u}{\partial z} = \frac{\partial v}{\partial z} = \frac{\partial k}{\partial z} = \frac{\partial \varepsilon}{\partial z} = \frac{\partial T}{\partial z} = 0,\quad w = 0 \tag{9}$$

Outlet boundary:

$$\frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial w}{\partial x} = \frac{\partial k}{\partial z} = \frac{\partial \varepsilon}{\partial z} = \frac{\partial T}{\partial x} = 0 \tag{10}$$

Solid wall region:

$$u = v = w = 0 \tag{11}$$

The method of standard wall function is employed to solve the boundary conditions near the solid walls [16].

2.4 Parameter definitions

Some parameters are defined in order to analyze the simulation works. The Reynolds number Re, adiabatic temperature difference ratio η, actual temperature difference ratio ζ, dimensionless height Y and dimensionless temperature Θ are defined as follows:

$$Re_g = \rho_g u_g D / \mu \tag{12}$$

$$\eta = (T_g - T_{aw})/(T_g - T_c) \tag{13}$$

$$\zeta = (T_g - T_w)/(T_g - T_c) \tag{14}$$

$$Y = H / d \tag{15}$$

$$\Theta = (T - T_c)/(T_g - T_c) \tag{16}$$

where ρ_g is hot air density, u_g is hot air inlet velocity, D is hot air inlet hydraulic diameter, μ is kinetic viscosity, T_g is hot air inlet temperature, T_c is cool air inlet temperature, T_{aw} is adiabatic wall temperature, T_w is actual wall temperature, H is vertical distance from wall surface and d is slot height.

3. NUMERICAL METHODS

The governing equations in this study are solved by using a finite volume method (FVM). The commercial software FLUENT 6.3 was adopted in the simulation works to solve the governing equations. The SIMPLEC algorithm was used to couple pressure and velocity. The convective terms and diffusion terms in governing equations are discretized by the second upwind scheme and the central difference scheme, respectively. The heat flux between the solid walls and air flow equal to zero when analyzed the adiabatic temperature difference ratio. In addition, when analyzed the actual temperature difference ratio, the fluid-solid coupled method is used to solve heat conduction between solid walls and air flow [16].

The solution is considered to be converged as the normalized residual of continuity equation is below 5.0×10^{-6} and when the target achieve, the normalized residual of momentum, turbulent kinetic energy and specific dissipation rate equations are all below 1.0×10^{-6}.The normalized residual of energy equation is below 1.0×10^{-10} and the temperature field will not change any more.

The three-dimensional geometric model is created and meshed using the commercial software GAMBIT 2.4. The whole computational domain was split into several small pieces and each piece is adopted structured grid. In order to well obtain the drastic changes of fluid parameters near the solid walls, grid meshing become more intensive from all around to solid walls in both the x-axis and y-axis directions. The width ratio of adjacent control volume in the same direction is limited within the range of 0.9 to 1.2. The grid distribution is shown in Fig. 4.

(a)

(b)

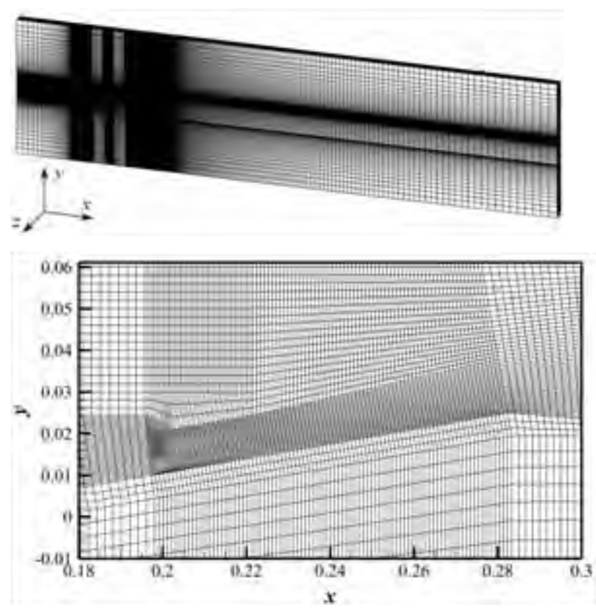

Figure 4. Grid distribution. (a)The whole mesh; (b)Grid distribution of the cutback area in section xy.

The grid-independent assessment has been made as slot inner angle β=0°, so that the computational results can be deemed enough to replicate the physical phenomenon. Results of grid assessment are shown in Table 3. It was found that the maximum difference in average adiabatic temperature difference ratio or average actual temperature difference ratio between the results of grid systems of 442×108×20 and 442×114×30 is below 1.0 percent, and finally the grid system of 442×108×20 is regarded as the one for proving grid independence solutions.

4. RESULTS AND DISCUSSIONS

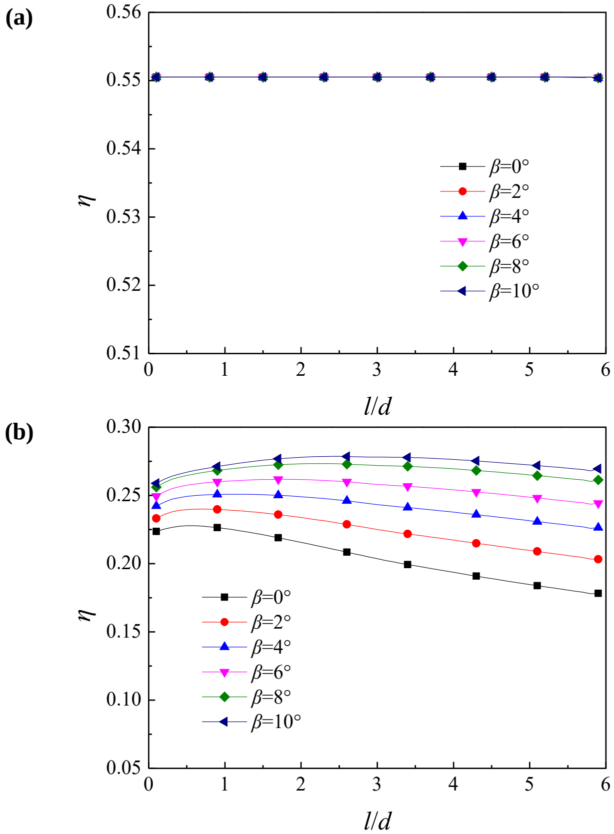

Figure 5. ***η*** **along with the change of** ***l*****/*****d*****. (a)Center line of slot downstream; (b)Center line of rib downstream.**

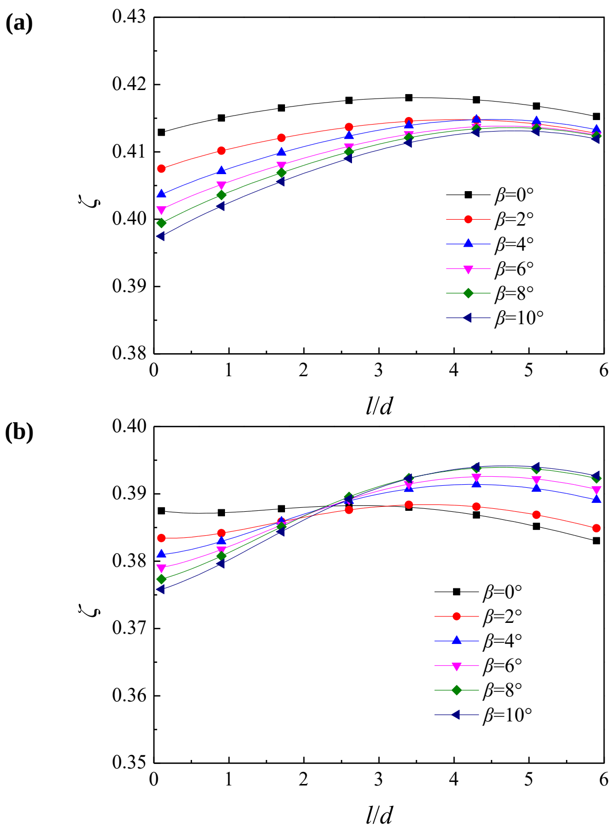

Figure 6. ***ζ*** **along with the change of** ***l*****/*****d*****. (a)Center line of slot downstream; (b)Center line of rib downstream.**

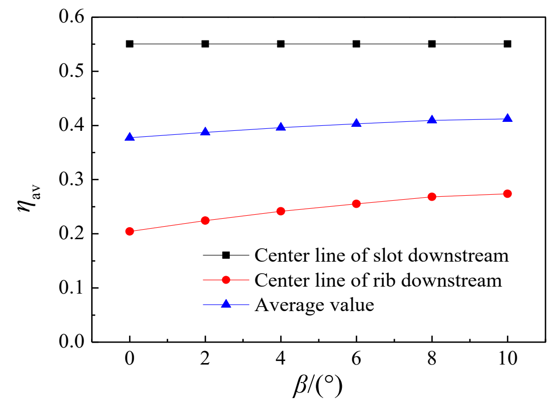

Figure 7. $\boldsymbol{\eta_{av}}$ **along with the change of** $\boldsymbol{\beta}$**.**

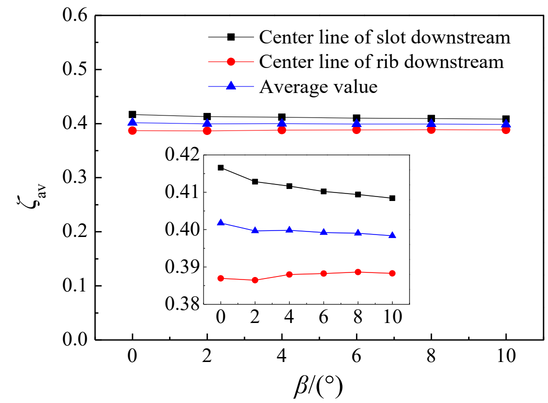

Figure 8. $\boldsymbol{\zeta_{av}}$ **along with the change of** $\boldsymbol{\beta}$**.**

Figure 5 shows the adiabatic temperature difference ratio along with the change of *l*/*d* under different slot inner angles. It can be seen from Fig. 5(a) that the adiabatic temperature difference ratio is almost constant on the downstream of slot centre line while slot inner angle change from 0° to 10°. It is because film cooling layers in slot downstream are not disturbed and remains intact, and the change of the thickness of film cooling layers are small as the slot inner angles have changed so that insulation effect is stable. It can be seen in Fig. 5(b) that the adiabatic temperature difference ratio on the downstream of rib centre line increases as the slot inner angle increase; it means that insulation effect is enhance when the slot inner angle raise.

Figure 6 shows the actual temperature difference ratio along with varying *l*/*d* under different slot inner angles. It can be seen that the actual temperature difference ratio on the downstream of slot centre line decrease as the slot inner angle increase. Meanwhile, on the downstream of rib centre line, the actual temperature difference ratio which near the rib wall tends to decrease while the far away area tends to increase.

Figures 7 and 8 show the average adiabatic temperature difference ratio and average actual temperature difference ratio versus the slot inner angle, respectively. It can be seen from Fig. 7 that the average adiabatic temperature difference ratio increases while the slot inner angle increases. When the slot inner angle change from 0° to 10°, the average adiabatic temperature difference ratio increases by 9.2% while the other parameters remain constants. It can be seen from Fig. 8 that the average actual temperature difference ratio tends to decrease while the slot inner angle increases. However, the change is small that the average actual temperature difference ratio increases by only 0.85% as the slot inner angle changes from 0° to 10° while the other parameters remain constants.

The computational results show that the trailing edge cutback wall temperature did not reduce although the raise of slot inner angle can enhance the film cooling insulation effect. Conversely, the trailing edge cutback wall temperature drop when the slot inner angle decreases. That means it is not enough that merely assessing the adiabatic temperature difference ratio in studying of the trailing edge cutback film cooling, and the actual temperature difference ratio should be taken into account.

Table 3. Results of grid assessment.

No.	Grid number ($x \times y \times z$)	Total element number	η_{av}	ζ_{av}
1	324×102×10	325412	0.3582	0.3855
2	324×102×20	630472	0.3706	0.4018
3	442×108×20	908380	0.3756	0.4071
4	442×114×30	1418010	0.3787	0.4095

5. CONCLUSION

In this paper, a numerical investigation was carried to study the influence of slot inner angles on the trailing edge cutback film cooling. The following conclusions can be made:

(1)The raise of slot inner angle can be useful in improving the performance of trailing edge cutback film cooling.

(2)The effect is small to the trailing edge cutback film cooling under different slot inner angle, however, the larger slot inner angle is, the lower cutback wall temperature will be.

(3)It is not enough that merely assessing the adiabatic temperature difference ratio in studying of the trailing edge cutback film cooling, and the actual temperature difference ratio should be taken into account.

6. ACKNOWLEDGMENTS

This work is supported by the Key Disciplines Construction Project of Power Engineering and Engineering Thermophysics of Guangdong Ocean University (No. 52130600801)

7. REFERENCES

[1] Uzol, O., Camci, C., & Glezer, B. 2001. Aerodynamic loss characteristics of a turbine blade with trailing edge coolant ejection: part 1—effect of cut-back length, spanwise rib spacing, free-stream reynolds number, and chordwise rib length on discharge coefficients. Journal of Turbomachinery, 123, 2(May 8–11, 2000), 238-248. DOI= http://doi.acm.org/10.1115/2000-GT-0258.

[2] Uzol, O., & Camci, C. 2001. Aerodynamic loss characteristics of a turbine blade with trailing edge coolant ejection: part 2—external aerodynamics, total pressure losses, and predictions. *Journal of Turbomachinery*, 123, 2(February 01, 2000), 249-257. DOI= http://doi.acm.org/10.1115/1.1351817.

[3] Martini, P., Schulz, A., Whitney, C. F., & Lutum, E. 2003. Experimental and numerical investigation of trailing edge film cooling downstream of a slot with internal rib arrays. *Proceedings of the Institution of Mechanical Engineers Part A Journal of Power & Energy*, 217, 4(January 2003), 393-401. DOI= http://doi.acm.org/10.1243/095765003322315450.

[4] Martini P, Schulz A. 2004. Experimental and Numerical Investigation of Trailing Edge Film Cooling by Circular Coolant Wall Jets Ejected From a Slot With Internal Rib Arrays. *Journal of Turbomachinery*, 2004, 126(June 2003), 229-236. DOI= http://doi.acm.org/10.1115/1.1645531.

[5] Han J C, Dutta S, Ekkad S. 2000. Gas turbine heat transfer and cooling technology. *CRC Press*.

[6] Sivasegaram S, Whitelaw J H. 1969. Film cooling slots: the importance of lip thickness and injection angle. *Journal of Mechanical Engineering Science*, 11, 1(February 1, 1969), 22-27. DOI= http://doi.acm.org/10.1243/JMES_JOUR_1969_011_005_02.

[7] Burns W K, Stollery J L. 1969. The influence of foreign gas injection and slot geometry on film cooling effectiveness. *International Journal of Heat and Mass Transfer*, 12, 8(August 1969), 935-951. DOI= http://doi.acm.org/10.1016/0017-9310(69)90156-2.

[8] LING Chang-ming, ZHI Zhong-pei, GUAN Zhi-qiang, et al. 2004. Two Dimensional Model Study on Cooling Configuration for the Trailing Edge of Turbine Vane. *Industrial Heating*, 33, 3(March 2004), 20-22. DOI= http://doi.acm.org/10.3969/j.issn.1002-1639.2004.03.005.

[9] Zhu Hui-ren, Yuan He-peng, Zhou Zhi-qiang, et al. 2006. Effect of geometry of back-step slots on film cooling. *Journal of Propulsion Technology*, 27, 4(August 2006), 312-315. DOI= http://doi.acm.org/10.3321/j.issn:1001-4055.2006.04.006.

[10] YUAN He-peng, ZHU Hui-ren, KONG Man-zhao, et al. 2006. Effect of geometry of back-step on heat transfer coefficients in three dimensional slots. *Journal of Thermal Science and Technology*, 5,1(march 2006), 17-21. DOI= http://doi.acm.org/10.13738/j.issn.1671-8097.2006.01.004.

[11] ZHU Hui-ren, YUAN He-peng, 2006. ZHOU Zhi-qiang, et al. Effect of Aerodynamic Parameters of Backward-Step Three Dimensional Slots on Film Cooling Effectiveness. *Journal of Aerospace Power*, 21, 2(April 2006), 315-319. DOI= http://doi.acm.org/10.13224/j.cnki.jasp.2006.02.016.

[12] Yuan H, Zhu H, Kong M. 2008. Effects of blowing ratio measured by liquid crystal on heat transfer characteristics of trailing edge cutback. *Chinese Journal of Aeronautics*, 21, 6(December 2008), 488-495. DOI= http://doi.acm.org/10.1016/S1000-9361(08)60165-X.

[13] Tian Li-ting. 2004. Study on Flow Resistance and Adiabatic Effectiveness of Film Cooling Slot in Trailing Edge of Turbine Blade. Zhanjiang: Zhanjiang Ocean University.

[14] Min Chun-hua. 2003. Study on Flow and Heat Transfer of Cooling Configuration for the Trailing Edge of Turbine Vane. Zhanjiang: Zhanjiang Ocean University.

[15] Qu Z G, Tao W Q, He Y L. 2004. Three-dimensional numerical simulation on laminar heat transfer and fluid flow characteristics of strip fin surface with X-arrangements of strips. *ASME Journal of Heat Transfer*, 126, 5(November 16, 2004), 697-707. DOI= http://doi.acm.org/10.1115/1.1798971.

[16] Tao Wen-quan. Numerical Heat Transfer(2nd Edition). 1988. Xi'an: Xi'an Jiaotong University Press.

[17] W. Rodi. 1984. Turbulence models and their application in hydraulics: a state of the art review. *International Association for Hydraulic Research*.