Research and development of asymmetrical heat transfer augmentation method in radial channels of blades for high temperature gas turbines

I V Shevchenko1, A N Rogalev1, I V Garanin2, A N Vegera2 and V O Kindra2

1 Department of Innovative Technologies of High-Tech Industries, National Research University “Moscow Power Engineering Institute”, Krasnokazarmennaya 14, Moscow, 111250 Russia
2 Innovation Department, National Research University “Moscow Power Engineering Institute”, Krasnokazarmennaya 14, Moscow, 111250 Russia

E-mail: kindra.vladimir@yandex.ru

Abstract. The serpentine-like one and half-pass cooling channel systems are primarily used in blades fabricated by the lost-wax casting process. The heat transfer turbulators like cross-sectional or angled ribs used in channels of the midchord region failed to eliminate the temperature irregularity from the suction and pressure sides, which is reaching 200°C for a first stage blade of the high-pressure turbine for an aircraft engine. This paper presents the results of a numerical and experimental test of an advanced heat transfer augmentation system in radial channels developed for alignment of the temperature field from the suction and pressure sides. A numerical simulation of three-dimensional coolant flow for a wide range of Reynolds numbers was carried out using ANSYS CFX software. Effect of geometrical parameters on the heat removal asymmetry was determined. The test results of a blade with the proposed intensification system conducted in a liquid-metal thermostat confirmed the accuracy of calculations. Based on the experimental data, the dependencies for calculation of heat transfer coefficients to the cooling air in the blade studied were obtained.

1. Introduction
The serpentine-like one and half-pass cooling channel systems are broadly used in cooling of the gas turbine blades. The midchord region of these blades has the radial channels with the centrifugal and radial-inward cooling air flow. The cross-sectional or angled ribs used in channels of the feather median sections [1, 2, 3, 4] failed to eliminate the temperature irregularity from the suction and pressure sides, which is reaching 200°C for a first stage blade of the high-pressure turbine for an aircraft engine. This irregularity derived from distribution of gas velocities in the cascade blade channel, while in blade systems with a low-aspect ratio it is derived from the effect of a vortex pair from the suction side. Difference in temperature of the suction and pressure sides causes additional thermal stresses decreasing of blade strength coefficients. Research and development of asymmetrical heat transfer augmentation methods in radial cooling channels of turbine blades allowing to reduce the
irregularity of temperature field in an airfoil cross section without the use of film cooling was the purpose of this article.

2. Numerical modelling of heat and mass transfer processes in cooling channels

Software tool ANSYS CFX was used for simulation of heat and mass transfer processes in cooling channels. The $k-\omega$ turbulence model with an automatic wall function was used to close Reynolds-averaged Navier–Stokes equations. An unstructured tetrahedral computational grid with prismatic layer for flow was created in ICEM. Maximum linear dimension is 0.0025 m, minimal linear dimension of grid elements is 0.0005 m. A near-wall region has detailed resolution using the prisms with the following parameters: number of layers – 11, prism initial height – 0.000002 m, growth law – WB-exponential. Total element number – 20–30 mln.

Model of the midchord region consisting of two square cross section channels ($4 \times 4$ mm) was the object of research. The channel walls had a cross-sectional ribbing. Ribs were installed at an angle of 90° to the air flow direction. Rib height 0.5 mm, width – 0.5 mm, pitch – 5 mm. On the opposite walls the ribs were offsetted by half a pitch. A general view of the model is shown in figure 1. The inlet channel located on the right side is connected with the outlet channel through the holes made in a separation rib on the wall simulating the pressure side. Ribs in the inlet channel are installed downstream of the holes and in the outlet channel – upstream of the holes. The hole height was equal to the rib height of 0.5 mm, hole length was set at 0.5, 1.0 and 1.5 mm. The separation rib had a bend on its ends for simulation of the open flow area restriction at the inlet of outlet channel and at the outlet of inlet channel.

![Figure 1](image)

**Figure 1.** The model studied: (a) – view of model with removed wall; (b) – model cross section.

First kind boundary conditions were set on the channel external surface – wall temperature was 419°C, air temperature at the model inlet was 20 °C, range of Reynolds numbers – from 6 000 to 55 000. Thermal conductivity of the model walls – 16 W/m·K. Ten cases of the channels were studied; their geometric parameters are given in table 1. The inlet and outlet channel restrictions for each case were equal. Case 1 (two parallel channels without restrictions and with two-side ribbing) was taken as a basic case.

The distribution of local heat transfer coefficients lengthwise of the channels from the suction and pressure sides, values of Nusselt numbers ($Nu$), relative Nusselt numbers (intensification coefficient) $C = Nu/Nu_0$ were determined based on the performed calculation data. $Nu_0$ was calculated by the equation (1).

$$Nu_0 = 0.021Re^{0.8}Pr^{0.43},$$  \hspace{1cm} (1)$$

where $Re$ – Reynolds number;
$Pr$ – Prandtl number, relative coefficient of linear hydraulic resistance $f/f_0$. 


Table 1. Model geometric dimensions.

| Case | Hole dimension, mm | Hole area, mm² | Restriction dimension, mm | Restriction area, mm² |
|------|--------------------|----------------|--------------------------|----------------------|
| Case 1 | 0                  | 0              | 4                        | 16                   |
| Case 2 | 1                  | 1              | 1                        | 4                    |
| Case 3 | 1                  | 1              | 2                        | 8                    |
| Case 4 | 1                  | 1              | 3                        | 12                   |
| Case 5 | 0.5                | 0.5            | 1                        | 4                    |
| Case 6 | 0.5                | 0.5            | 2                        | 8                    |
| Case 7 | 0.5                | 0.5            | 3                        | 12                   |
| Case 8 | 1.5                | 1.5            | 1                        | 4                    |
| Case 9 | 1.5                | 1.5            | 2                        | 8                    |
| Case 10 | 1.5                | 1.5           | 3                        | 12                   |

Nu/Nu₀ and f/f₀ values averaged lengthwise of the channels were also determined on the basis of which a preliminary data analysis was performed. Intensification coefficients for Reynolds number of 20 000 in channels are given in Table 2.

Table 2. Thermohydraulic parameters of models studied.

| Case | IP | IS | OP | OS | IP | IS | OP | OS |
|------|----|----|----|----|----|----|----|----|
| Case 1 | 1.95 | 1.95 | 1.95 | 1.95 | 7.03 | 7.96 | 8.00 | 7.84 |
| Case 2 | 2.25 | 1.80 | 4.46 | 3.20 | 4.82 | 6.16 | 7.95 | 11.70 |
| Case 3 | 1.93 | 1.64 | 3.37 | 2.29 | 4.73 | 5.80 | 9.53 | 9.88 |
| Case 4 | 1.75 | 1.59 | 2.31 | 2.06 | 4.67 | 5.69 | 7.15 | 7.93 |
| Case 5 | 2.22 | 1.94 | 3.99 | 3.09 | 4.56 | 7.14 | 8.32 | 8.32 |
| Case 6 | 2.01 | 1.84 | 2.90 | 2.27 | 4.75 | 7.23 | 7.69 | 7.95 |
| Case 7 | 2.19 | 2.05 | 2.57 | 2.20 | 6.22 | 8.64 | 7.62 | 8.40 |
| Case 8 | 2.31 | 1.80 | 3.83 | 2.68 | 6.71 | 8.68 | 12.00 | 13.14 |
| Case 9 | 2.02 | 1.65 | 3.18 | 2.24 | 5.74 | 6.98 | 9.06 | 9.83 |
| Case 10 | 1.85 | 1.60 | 2.44 | 1.94 | 5.69 | 7.05 | 7.61 | 8.39 |

As it is shown in Table 2, cases 5 and 6 are most preferred as compared to the basic case 1. They ensure retention of the intensification level from the suction side and its increasing from the pressure side. If intensification coefficient given will be used as a criterion, than in this case reference should be made to the cases with a larger hole area (cases 2, 3, 4 and 8, 9, 10).

The channel system forms a complex three-dimensional flow restricting drawing of criterion dependencies for calculation of local heat transfer coefficients considering all geometrical parameters. Examination of the geometry of each channel with determination of the heat-transfer intensification coefficients is more reasonable.

The proposed channel system operates in the following way. The air flows through the inlet channel, decelerates by guiding ribs installed on the pressure side wall and then flows in the outlet channel. The overshadowing ribs in the outlet channel installed on the pressure side wall form the detached flow regions, in which the air streams pass from the through holes. The air flow through the holes formed by the guiding ribs provides a one-side rundown of the boundary layer on the inlet channel wall from the pressure side, while the overshadowing ribs – one-side jet intensification of the heat transfer on the outlet channel wall from the suction side.
3. Experimental investigation of heat transfer processes in cooling channels

Examination of a blade with the serpentine-like cooling channel system has been conducted to prove the simulation results. The blade was fabricated using selective laser sintering method. The airfoil cross section is shown in figure 2. Channels K3, K4, K5 and K6 were provided with the proposed intensification system.

![Figure 2. The midsection of the airfoil.](image)

Tests were carried out using the calorimetric measurement method in a liquid-metal thermostat [5, 6]. The blade was tested at five different pressure differences $P/P_0 = 1.49; 1.68; 1.78; 1.97; 2.37$. Three experiments were performed for each mode. The local coefficients of heat transfer $\alpha$ to the cooling air along the internal surface coordinate $S$ performed at figure 3. The difference of heat transfer coefficients in the adjacent radial channels were due to a different rate of the heat transfer augmentation and a different value of local temperatures used for calculation of heat transfer coefficients. The heat transfer augmentation in radial channels from the pressure side is higher than from the suction side by 1.8–2.0 times. Nusselt number values ($\text{Nu}_i$) were calculated using the obtained values of heat transfer coefficients for each test mode in every point of the internal surface. Thereafter, the criterion dependencies were drawn using method of least squares (equation (2)).

$$\text{Nu}_i = A_i \cdot \text{Re}_i \cdot 0.8.$$  

(2)

Also, the heat transfer coefficients were calculated using dependence $\text{Nu}_i/\text{Nu}_{0i}$. Values of intensification coefficients $C_i$ are given in table 3. As seen from table 3, for example, for channel K4, the intensification coefficient from the pressure side is higher by 1.65 times. Obtained experimental data proves the efficiency of developed method for a one-side heat transfer augmentation.

4. Conclusions

Method for the heat transfer augmentation in radial cooling channels of gas turbine blades was developed. It is grounded in establishing of near-wall jet flow on the channel walls from the pressure side. Variations in the rib hole sizes and areas of the inlet and outlet cross sections of the radial channels allow to change the heat transfer augmentation rate on the channel opposite walls over a wide range. Using this system for cooling the blade midsection allows to reduce the temperature field irregularity in the feather cross section without film cooling of the pressure side.
Figure 3. Distribution of heat transfer coefficients to the cooling air along the internal surface of the feather midsection depending on a pressure difference.

Table 3. Values of intensification coefficient along the blade outline.

| S  | C   | S  | C   | S  | C   | S  | C   |
|----|-----|----|-----|----|-----|----|-----|
| 32 | 3.624 | 17 | 8.584 | -3 | 2.177 | -23 | 5.165 |
| 31 | 7.325 | 16 | 3.977 | -4 | 2.429 | -24 | 5.079 |
| 30 | 7.051 | 10 | 9.265 | -5 | 2.452 | -25 | 5.088 |
| 29 | 7.117 | 9  | 9.425 | -6 | 2.519 | -26 | 5.309 |
| 28 | 7.310 | 8  | 4.625 | -12| 2.232 | -27 | 5.625 |
| 27 | 7.521 | 7  | 4.633 | -13| 4.395 | -28 | 2.102 |
| 26 | 7.786 | 6  | 4.605 | -14| 4.352 | -34 | 6.466 |
| 25 | 8.034 | 5  | 4.631 | -15| 4.325 | -35 | 6.658 |
| 24 | 3.352 | 4  | 4.779 | -16| 4.362 | -36 | 6.712 |
| 23 | 3.380 | 3  | 5.562 | -17| 5.233 | -37 | 6.726 |
| 22 | 3.371 | 2  | 10.94 | -18| 2.166 | -38 | 6.610 |
| 21 | 3.354 | 1  | 7.136 | -19| 2.255 | -39 | 6.467 |
| 20 | 9.221 | 0  | 4.825 | -20| 2.371 | -40 | 2.850 |
| 19 | 9.039 | -1 | 1.974 | -21| 2.454 | -41 | 2.783 |
| 18 | 8.770 | -2 | 1.953 | -22| 2.498 | -42 | 2.712 |

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