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Numerical analysis of the influence of turbulators constructive features on heat transfer in gas turbine blade cooling channels

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Abstract

In the present study, the influence of turbulators constructive features on heat transfer in gas turbine blade cooling channels has been investigated by analyzing the results of numerical modelling. The numerical model has been verified by experimental data. The radial channel of the leading edge with the heat transfer turbulators in the form of ribs was the first object of a numerical study. Three options of the rib design were investigated: with the sharp edges of all rib surfaces, with the rounded end surfaces of the ribs and with end rounding of the top and side rib surfaces. Additionally, an option with ribs installed at an angle of 45° relatively to the flow direction was studied. A convergent channel of the blade trailing edge with one and half-pass cooling channel was the second object of a numerical investigation. Effect of an angle of the coolant flow incidence on heat transfer for staggered pin fin array has been analyzed. A semicircular cross-section channel with the coolant radial flow, in which the air supply was carried out through circular holes was the last research object. A cross flow influence on thermal and hydraulic characteristics has been investigated.

Keywords: rib turbulated cooling, pin fin cooling, impingement cooling, heat transfer, gas turbine blade, trailing edge, leading edge, internal cooling passages

INTRODUCTION

An increase in the cycle initial temperature is one of the key focus areas in the development of gas turbines. Increases in the inlet parameters of gas turbines are mainly limited by the efficiency of a blade cooling system. A precision investment casting is the main technological process in fabrication of cooled blades for the high-temperature gas turbines. Cooling channels are formed with ceramic rods (Fig. 1) that are fabricated by moulding process using special moulds.

This technological process limits the geometry of turbulators in the cooling channels. Specifically, for blades of modern gas turbine engines the height of wall ribs is at least 0.3 mm, ribs and pins of vortex matrices have a tilt relatively to the feather walls and normal cross section of radial channels differs from rectangular shape. In addition, we must take into account an effect of standard deviations on the geometry of turbulators (Fig. 2).

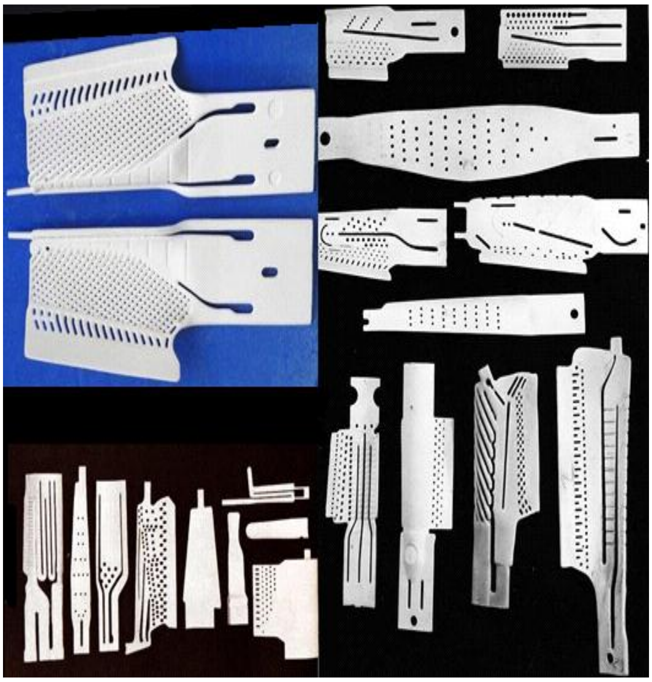


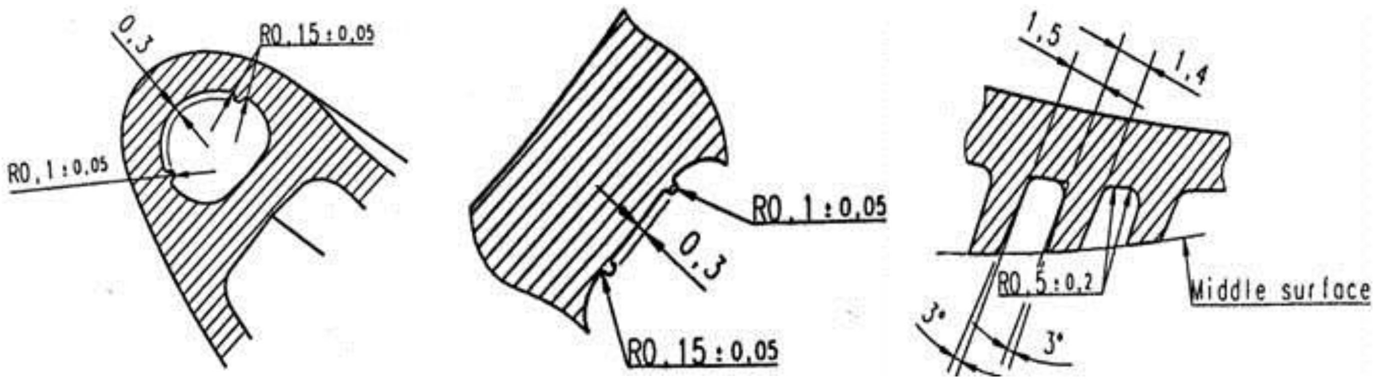
Figure 1: Ceramic rods used to form the blade inner cavity with different cooling systems

For the positive deviation limits of the rounding radii of edges the cross-sections of ribs have the semicircle form and cross-sections of matrix channels are contracting to the external feather wall and have almost semicircle bottom (Fig. 2).

Nowadays, the heat transfer problems are solved to a high precision with any design, however accuracy of results obtained depends on the correctness of specifications for boundary conditions of heat transfer, which are determined by the efficacy of a thermal model. In a thermal model of the blade cooling system the criterion dependences obtained by different authors from the experimental tests are used for calculation of heat transfer coefficients.Generally, a design engineer makes choice for the dependence. For example, the average heat transfer values in paths with the coplanar channels having an intersection angle 2β = 90° obtained by different authors differ almost twice. Also it should be noted that in practice all experimental tests of the hydraulic and thermal characteristics of the heat transfer turbulators were conducted on the rectangular shape channels with a strongly defined air flow direction.

All of the factors described above may contribute to the blade thermohydraulic model being developed failing to fully simulate the air flow and the heat exchange processes in some sections of the cooling path. Hence, the design temperature field will differ from the temperature field of an actual blade in full-scale conditions of a turbine operation.

In this article the numerical research was being conducted on the effects of several design and technological factors on the heat transfer rate in the turbine blade cooling channels. Software ANSYS CFX were used for simulation of the heat-mass exchange process in the cooling channels. Because of high Reynolds numbers observed in cooling channels of gas turbine blades, k–ω turbulence model with the scalable wall function were used for simulation of the heat-mass exchange process [1, 2, 3].



a) b) c)

a – ribs in a leading edge channel; b – ribs in a radial channel; c – ribs of the vortex matrix from the blade pressure side

Figure 2: Design of turbulators in inner cavity channels of cast blades

VERIFICATION OF NUMERICAL MODELS

Verification of numerical model was carried out using a two-dimensional channel of the length 120 mm with channel cross-section of 2 × 10 mm on one of the wall at 50 mm distance from the inlet of 0.5 mm width with an angle 90° to the flow direction. The relative height of a rib was 0.25, all rib edges were sharp. Distribution of the local heat transfer coefficients lengthwise of the channel was obtained based on the data of channel test conducted in a liquid-metal thermostat [4, 5, 6, 7] that was compared with the results of a numerical simulation performed for the test conditions including the geometry of a section of the air supply to the channel.

Comparison results are presented in Fig. 3 and Fig. 4 in a form of the relationship between Nusselt number Nu and number Nu_0 calculated for a smooth channel by the equation (1).

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

where Re – Reynolds number;
Pr – Prandtl number.

The deviation of numerical heat transfer values from the experimental values of Nu/Nu_0 less than 10% confirming the accuracy of selected parameters of the grid and turbulence model.

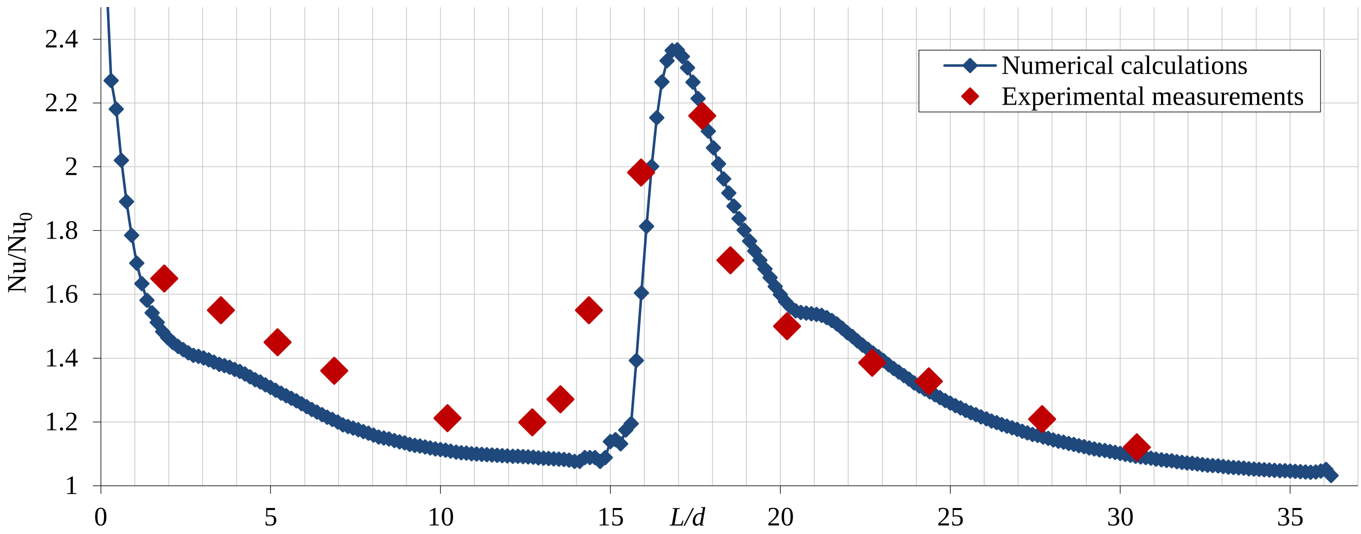


Figure 3: Distribution of Nu/Nu_0 along the channel length with a single rib on the ribbed wall at Re = 36000

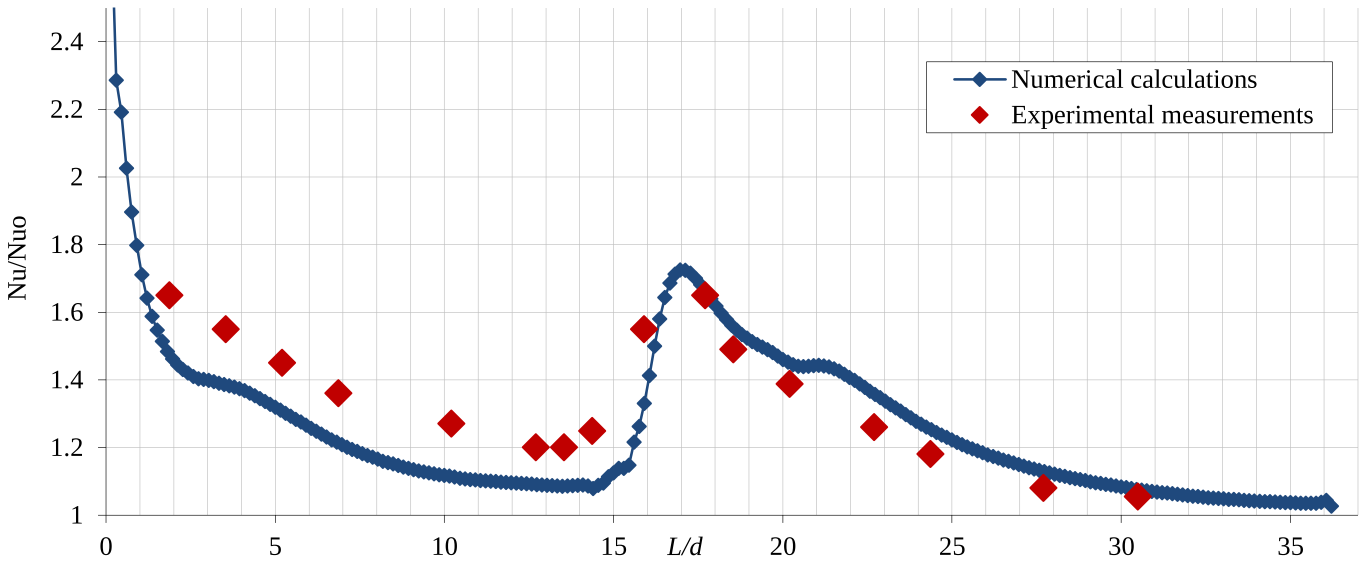


Figure 4: Distribution of Nu/Nu_0 along the channel length with a single rib on the smooth wall at Re = 36000

THE INFLUENCE OF SHAPE AND INCLINE OF RIBS ON THERMAL AND HYDRAULIC CHARACTERICTICS OF THE RADIAL CHANNEL OF THE LEADING EDGE

The radial channel of the leading edge with the heat transfer turbulators in the form of ribs shown in Fig. 5 was the first object of a numerical study [8, 9].

Geometrical parameters of the ribs are presented in Table 1 and geometrical parameters of the ribbed channel – in Table 2.

Figure 5: 3-D model of a ribbed channel

Table 1: Geometrical parameters of the ribs

Parameter	Value
Rib height, mm	0.3
Rib width, mm	0.3
Ribs pitch, mm	2.85
Angle of the rib setting relatively to the flow direction, °	90

Table 2: Geometrical parameters of ribbed channel

Parameter	Value
Channel hydraulic diameter *d*, mm	1.6
Channel length *L*, mm	48
Ribs number	17

Three options of the rib design were studied: with the sharp edges of all rib surfaces (Fig. 6a), with the rounded end surfaces of the ribs (Fig. 6b) and with additional end rounding of the top and side rib surfaces (Fig. 6c). An option with ribs installed at an angle of 45° relatively to the flow direction (Fig. 6d) was studied additionally. The air flow was supplied at the channel entrance. The channel wall temperature was 800°C, the intake air temperature was 50°C, the intake air pressure was 16 bar.

The results of solving the heat transfer problems for different options of the leading edge channel with the ribs perpendicular to the flow are presented in Fig. 7, 8 and 9 in the form of a relationship between Nusselt number and Nusselt number calculated for a smooth channel with the same cross-section by the equation (1).

Data analysis shows that the ribs with sharp edges have maximum efficiency and the ribs with all rounded edges – minimum.

As can be seen from the charts an influence of the rib geometry on the heat transfer rate is demonstrated at the distance of 15 hydraulic diameters from the inlet cross-section that is due to the initial flow turbulence. Rounding of all rib edges results in decreasing of Nu/Nu_0 by 15-20% (Re = 40000, L/d = 20-27)

With increase of Re number up to 10^5 the intensification coefficient is not more than 1.5 throughout the channel length and the effect of rib geometry on the heat transfer is not observed. The edge rounding allows for reducing the channel hydraulic resistance in the number Re range studied by 17-20% in comparison with a channel having all sharp edges.

The results of comparison of the thermal and hydraulic characteristics of channels with ribs installed at angles of 90° and 45° are presented in Table 3. The rib edges were sharp. The rib inclination tilt at an angle of 45° to the flow direction results in increasing of the Nusselt number by 17-22% as compared with the transverse ribs. Apart from that the coefficient of linear hydraulic resistance increases by 2.3 times..

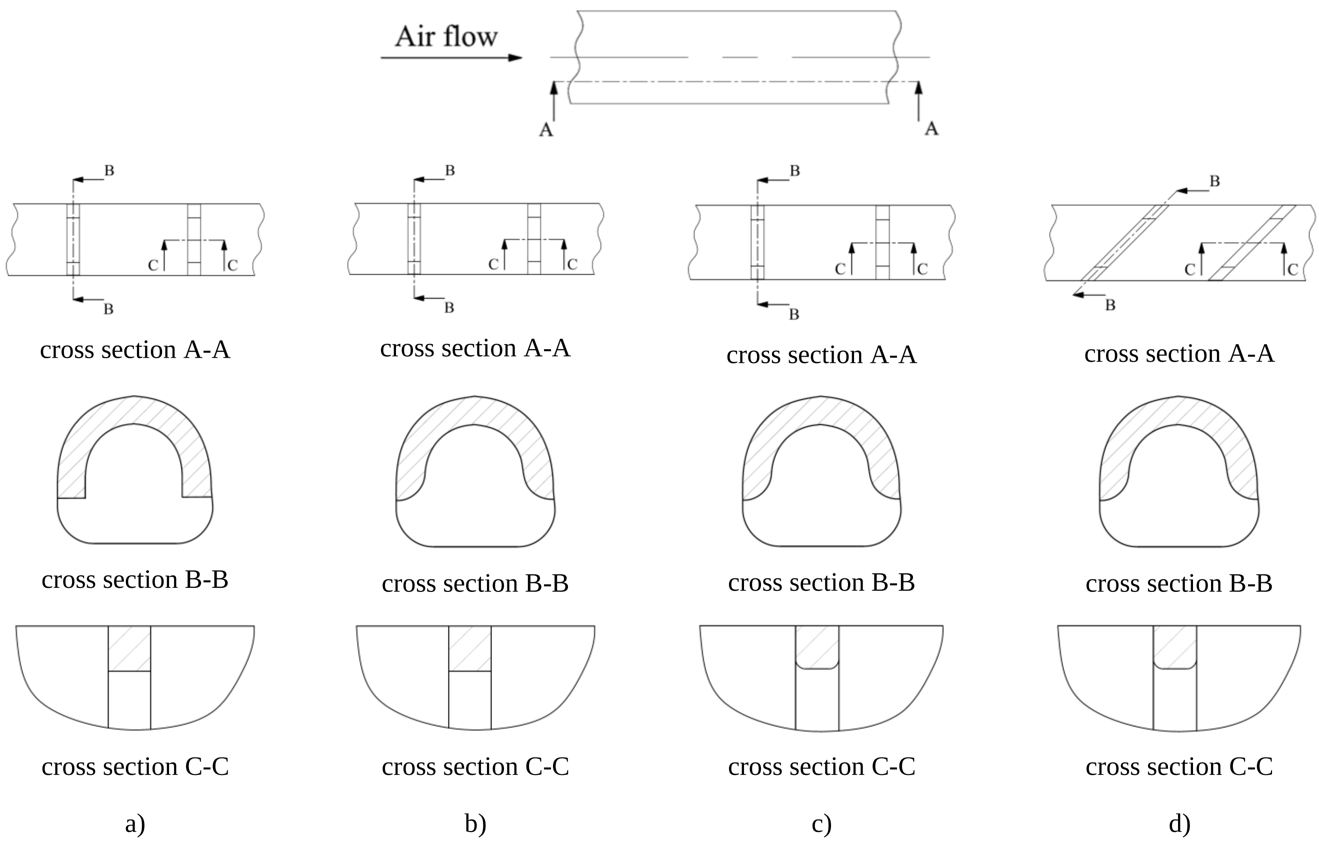


Figure 6: Ribs geometry

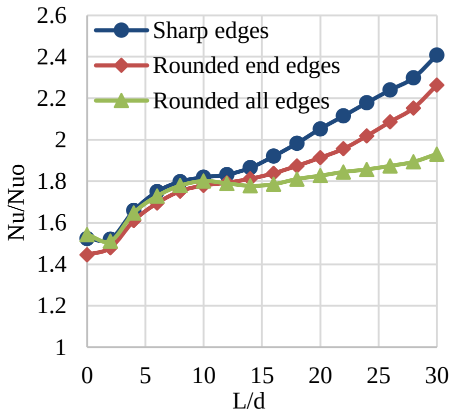


Figure 7: Distribution of the parameter value Nu/Nu_0 lengthwise the channel (Re = 40000)

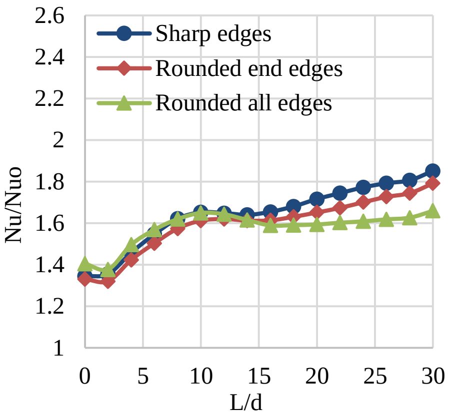


Figure 8: Distribution of the parameter value Nu/Nu_0 lengthwise the channel (Re = 70000)

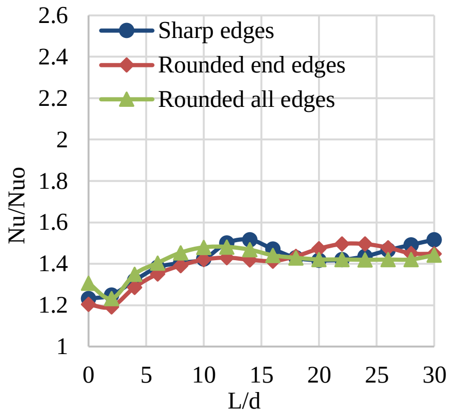


Figure 9: Distribution of the parameter value Nu/Nu_0 lengthwise the channel (Re = 100000)

Table 3: The numerical study data on installation of ribs

N	Model	$Re \cdot 10^{-3}$	Nu	Nu/Nu_0	f
1	Ribbed channel with ribs installed at an angle of 90°	40	121	1.9	34
2	Ribbed channel with ribs installed at an angle of 45°		148	2.3	80
3	Ribbed channel with ribs installed at an angle of 90°	70	159	1.6	35
4	Ribbed channel with ribs installed at an angle of 45°		185	1.9	81
5	Ribbed channel with ribs installed at an angle of 90°	100	172	1.4	36
6	Ribbed channel with ribs installed at an angle of 45°		197	1.6	83

EFFECT OF AN ANGLE OF THE COOLANT FLOW INCIDENCE ON HEAT TRANSFER ON THE INNER SURFACE OF A COOLING CHANNEL IN AREA OF THE GAS TURBINE BLADE TRAILING EDGE

A convergent channel of the blade trailing edge with one and half-pass cooling channel was the second object of a numerical study [10]. Four rows of the staggered pins were installed in the channel for the heat transfer intensification. A geometrical model of the trailing edge channel element is shown in Fig. 10. Pin diameter was set to D = 2 mm, longitudinal pin pitch to S_1 = 5.2 mm and transverse pin pitch – to S_2 = 5.2 mm. The stagnation pressure and temperature was set at the model entry according to Reynolds number at the channel entry equal to 50000. An effect of design features of the blade back cavity on the velocity vector of the flow incoming to the pin zone and, accordingly, on the heat transfer in the pin rows was simulated.

Calculations of the heat transfer coefficients and Nu numbers were carried out for four values of an entry angle of the air flow in to the trailing edge channel: 0°, 15°, 30° and 45°.

The diagrams presented in Fig.11 show different Nu number distribution on the concave and convex blade sides.

A decrease in the heat transfer rate with an increase in an entry angle of the flow relatively to the basic option (0°) are caused by the fact that the array resembles in in-line arrangement. This is especially prominent in the first rows of pins. Starting from around third row the streamlines are straightened (Fig. 12).

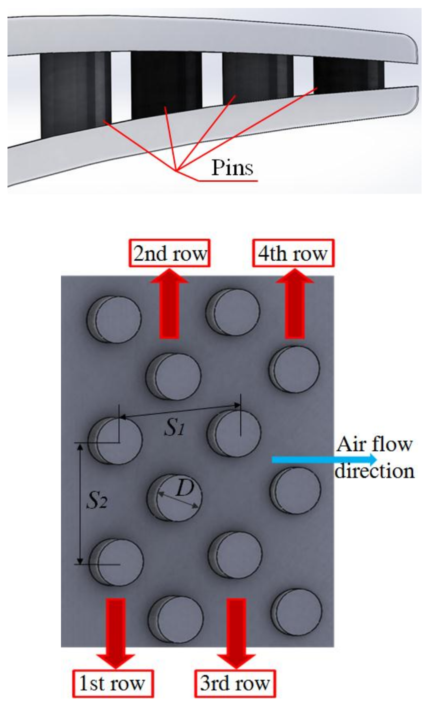


Figure 10: Geometric model of the trailing edge cooling channel

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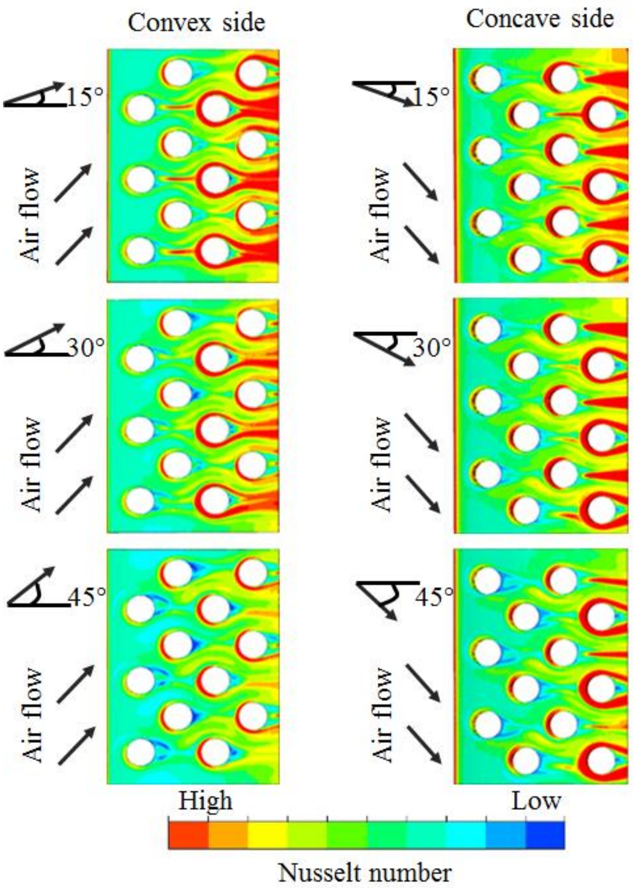


Figure 11: Effect of an angle of the coolant flow incidence on distribution of heat transfer coefficient on the inner surface of a cooling channel in area of the gas turbine blade trailing edge with ribs installed

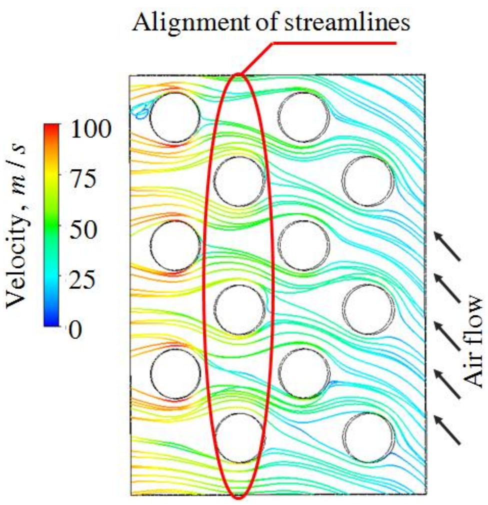


Figure 12: The coolant streamlines in a cooling channel with pins at an angle of the flow incidence of 45° and Re = 50000

The low-pressure areas in the "shadow" part of the pins was essentially increased at an angle of 45°, while the heat transfer rate at the last rows of pins was significantly reduced.

The flow deviation from axis direction by 15° had almost no effect on the heat transfer level in the pin area (Fig. 13). In other words, the flow typical for staggered pin arrangement was retained. Nusselt numbers were decreased by 10% across all pin rows with increase in an inlet angle up to 30°. At the initial flow angle of 45° a local decrease of the Nusselt number by 25% is shown in the area of third row of pins. This might be explained by the flow deflection at the segment of third row of pins as it's clearly seen from the calculation data of streamlines (Fig. 12).

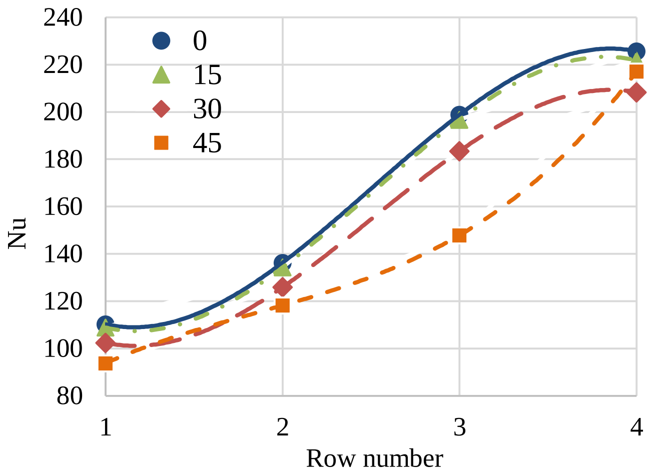


Figure 13: Effect of an angle of the coolant flow incidence on distribution of Nu across the pin rows at Re = 50000

Quantitative estimation of the effect of an angle of the coolant flow incidence on Nusselt number Nu for the staggered pin array is presented in graphic form in Fig. 14. This characteristic might be used as a correction for the heat transfer values in the cooling channel in case of deflection of an angle of the flow incidence.

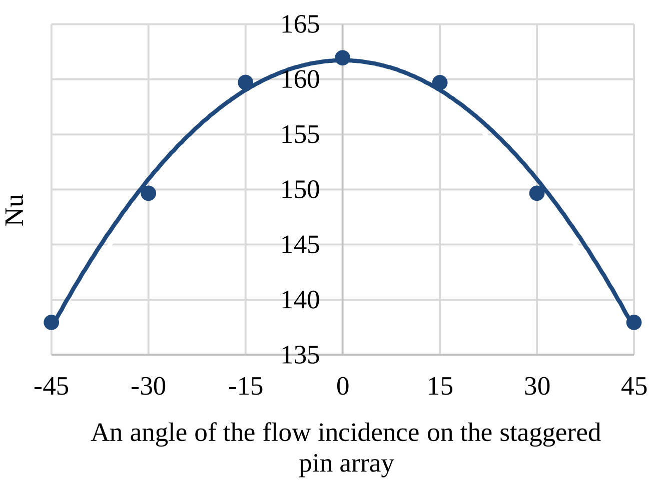


Figure 14: Effect of an angle of the coolant flow incidence for the cooling channel at Re = 50000

CROSS FLOW INFLUENCE ON THERMAL AND HYDRAULIC CHARACTERISTICS

An impingement cooling is one of the heat transfer intensification methods in a radial cooling channel of the leading edge [11]. The research object was a semicircular cross-section channel with the coolant radial flow, in which the air supply from a parallel collector channel was carried out through circular holes having diameter *d* (Fig. 15). Geometric parameters of the model are presented in Table 4.

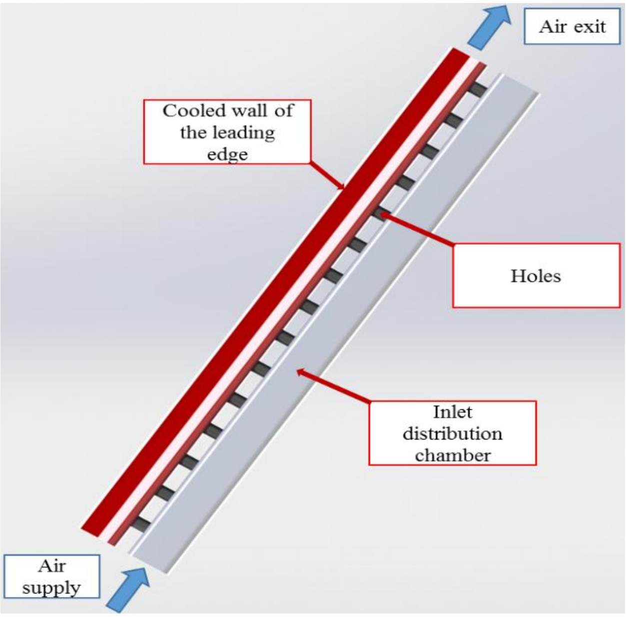


Figure 15: Model of leading edge cooling

Table 4: Channel geometric dimensions

Parameter	Value
Hole diameter, mm	2.05
Hole pitch, mm	0; 6; 15
Maximum distance from the hole surface to the leading edge, mm	6.15
Hole length, mm	2.05
Inside radius of the leading edge, mm	4.1
Total number of holes in a model, pcs	15

The cross section with computational grid showed in Fig. 16. Static flow pressure of 16 bar and air temperature of 350°C was set at the model exit. Cooled surface of the leading edge was simulated by the viscous isothermal wall (temperature of 800°C). Other walls of the model were assumed viscous, adiabatic.

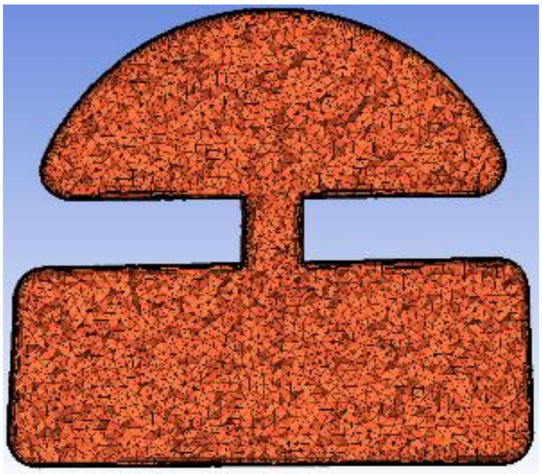


Figure 16: Channel cross section with computational grid

The model study was performed for three conditions: Re_{hole} = 5000, Re_{hole} = 10000, Re_{hole} = 15000. Diameter of holes was used as a hydraulic diameter for the calculation of Re_{hole}. Based on these data, we can conclude that the efficiency of jet cooling is sufficiently high especially for a condition with large Reynolds numbers: intensification is 3.2. Moreover, the hydraulic resistance of the considered cooling system is increased trice in comparison with the smooth radial channel, which is not critical.

The thermal and hydraulic characteristics of the radial channel with impingement cooling are presented in Table 5.

Table 5: The numerical study data on impingement cooling

N	$Re \cdot 10^{-3}$	Nu	Nu/Nu_0	*f*
1	5	22	2.4	1.8
2	10	38	2.4	1.9
3	15	52	3.2	2.0

Overall, a jet cooling system of the leading edge provides an essential heat transfer intensification (Fig. 17) with increase in hydraulic resistance by three times. An efficiency coefficient of such system is more than two. However, it should be taken into account that in actual blade there will be the air heating in an entrance channel and consequently a decrease in amount of removed heat in the feather peripheral sections. This intensification method will be best utilized as a local in the blade sections with the maximum gas flow temperature.

The model for numerical investigation of cross flow influence on thermal and hydraulic characteristics is illustrated in the Fig. 18.

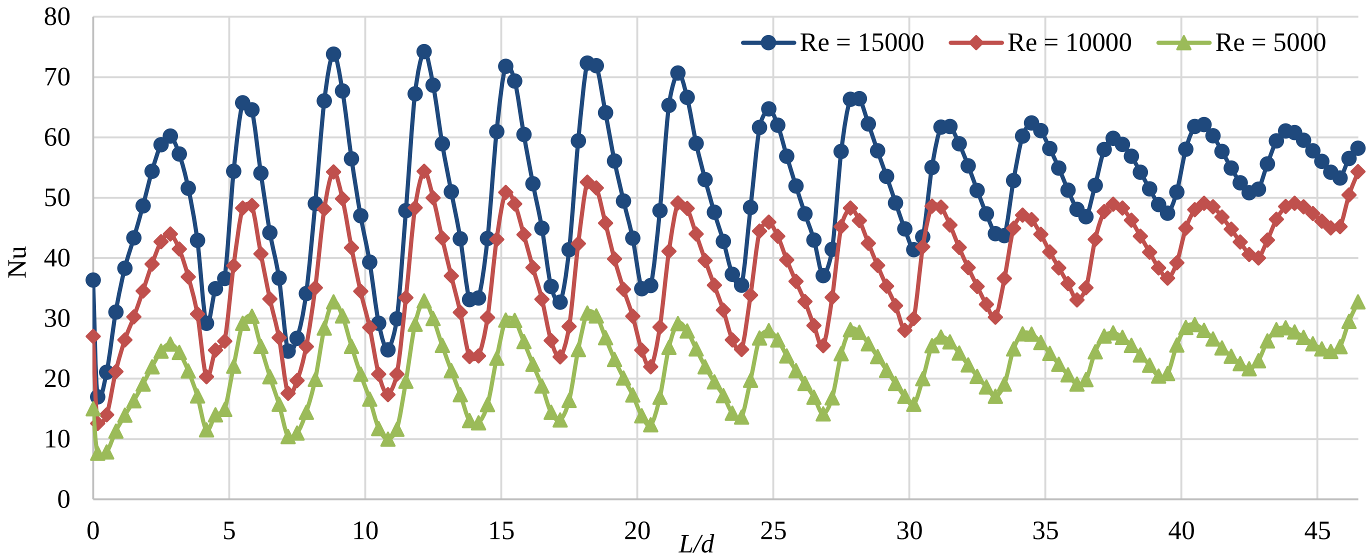


Figure 17: Distribution of Nusselt number along the channel length in the leading edge at different Reynolds numbers

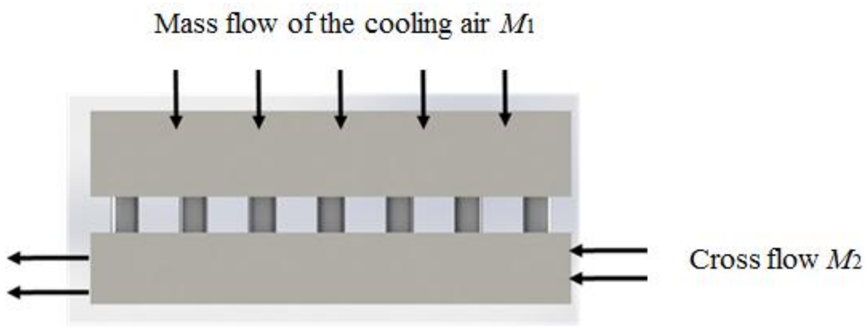


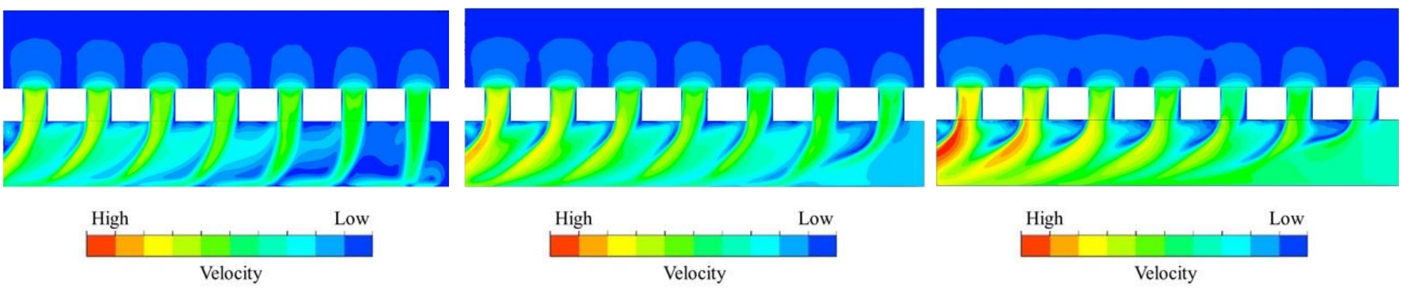
Figure 18: The three-dimensional model for the investigation of the cross flow influence on thermal and hydraulic characteristics

The method of study was as follows: the mass flow of the cooling air M_1 was fixed (to ensure constant Re_{hole}) and the cross flow M_2 was varied. The main parameters for the numerical modelling are presented in Table 6.

The distribution of the velocity in channel is illustrated in the Fig. 19. The higher cross flow, the more the area of region with no jets of cooling air and the lower the heat tarnsfer process in the system.

Table 6: The main parameters for the numerical modelling

Parameter	Value		
M_2, g/s	0	5	10
M_1, g/s	10		
Inlet air temperature, °C	350		
Outlet air pressure, bar	16		
Wall temperature, °C	800		



a) $M_2 = 0$ b) $M_2 = 0.5M_1$ c) $M_2 = M_1$

Figure 19: Cross flow influence on the velocity in channel

CONCLUSION

1) An influence of the rib geometry on the heat transfer rate manifests at the distance of 15 hydraulic diameters from the inlet cross-section due to the initial flow turbulence.

2) Rounding of all rib edges results in decreasing of Nu/Nu_0 by 15-20% at Reynolds numbers 40000.

As the Reynolds number is further increased the effect of rib geometry on the heat transfer is not observed.

3) The edge rounding allows for reducing the channel hydraulic resistance in the Reynolds number range studied by 17-20% in comparison with a channel having all sharp edges.

4) The rib inclination tilt at an angle of 45° to the flow direction results in increasing of the Nusselt number by 17-22% and increasing of the coefficient of linear hydraulic resistance by 2.3 times compare to the transverse ribs.

5) An increase of the coolant flow incidence from 0° to 45° in the area of blade trailing edge with staggered pins leads to a decrease of Nusselt number by 15%. It caused by the fact that the array resembles in in-line arrangement.

6) The cross flow has a bing influence on thermal and hydraulic characterictics in channel with impingement cooling.

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