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Numerical simulations of a conceptual blade cooling with a working medium

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The development of steam power units aims to increase the working steam parameters as they are the main factors that determine the efficiency of energy conversion. Most state of the art units are designed for supercritical steam parameters. However, the temperature level of steam feeding the turbine is limited by thermal strength of the material used to make the machine components. In this situation, using nickel alloys or cooling the elements exposed to the impact of high temperatures could be the appropriate solution. The former is rather expensive and the latter – technically difficult. The cooling option would require that the cooled element should be fed by a steam with a very high pressure and with a lower temperature than the temperature in the machine flow system. This paper presents the concept of using working steam as the cooling medium after it is expanded in a convergent-divergent nozzle. In such a case, the cooling system is very simple and the performed simulations indicate, for example, that the turbine blades may be cooled in this way.

Keywords: blade cooling, convergent-divergent nozzle, Laval nozzle, shock wave.

1. Introduction

One of the main directions in the development of modern power engineering is to maximize the efficiency of power plants. Better energy generation efficiency brings advantages such as reduced fuel consumption, reduced emissions of pollutants and lower costs of production. The building/development of supercritical power units is one of the most important directions in the development of conventional power plants. Due to the increase in steam parameters (temperature and pressure), it is possible to obtain efficiencies exceeding 50%. However, the turbines operating in supercritical systems have to meet the requirements resulting from much higher thermal loads. The ultimate goal is to achieve temperatures of the order of 700°C for live steam and reheated steam. Such temperatures will necessitate the use of state of the art materials (nickel alloys) or the implementation of cooling of the elements which are most exposed to thermal loads (first stages of the high and intermediate pressure parts). The first solution will increase investment costs considerably but the second one may make it possible to use currently applied materials at the expense of greater structural complexity of the machinery components. The present blade cooling systems are applied to gas turbines only. These systems have their origins in the 1960s. Since then, a wide range of solutions using the air as the cooling medium has been developed, from straight convective cooling channels to complex structures with film cooling [1]. Research is also pursued on solutions based on water vapour being the cooling medium [2] and channels with additional ribs [3]. These are open-system solutions (cooling involves mixing the coolant with the working medium). Closed systems, due to their much higher complexity, are not used at the present. Some solutions used in gas turbines could be adapted for application in the steam turbine blade cooling. Systems operating based on water vapour which is used as the cooling medium are especially interesting. Preliminary analyses of the possibility of the blade convective cooling using steam in a closed system have already been conducted in [4], and different variants to optimize the shape of cooling channels have been considered in [5]. Due to the cooling medium's low pressure, both the inflow and outflow would have to take place through the blade root. Such solutions are complex and may have a negative impact on the turbine availability. Open systems are better in this respect. Owing to their simplified structure they should not affect the machine availability. In the case of steam turbines, open systems are limited solely to water vapour being the coolant (negative impact of an inert gas on the condensation process in the condenser). However, the problem is that using open-system cooling requires a cooling medium with the inlet pressure higher than that prevailing in the flow system downstream the cooled blade ring. Preparing steam for cooling purposes would be problematic and expensive even for subcritical turbines. For supercritical ones, in which pressure can reach 30 MPa, the same would be irrational if not unfeasible because steam would first have to be generated with a pressure at least equal to that of live steam but with a lower temperature, and then it would have to be delivered through feeding channels into the blade. In order to avoid these problems, a concept is put forward that involves reversing the direction of the flow through cooling channels, where the cooling medium is an expanded working steam.

2. Self-cooling concept

The self-cooling concept is based on using the turbine working medium to cool the most thermally loaded components (i.e., the stator and rotor blades in the first stages of the high and intermediate pressure parts).

This, in fact, is convective cooling in an open system with the reversed coolant flow direction, where the steam is extracted from the turbine flow system. For this purpose, special systems consisting of a de Laval nozzle connected to outlet passages are placed inside the blades (Fig. 1).

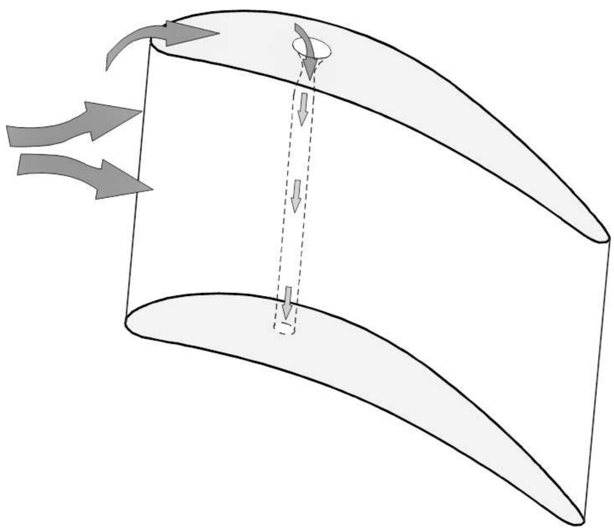

Fig. 1. Diagram of the blade cooling system.

The nozzle is fed with a steam from the turbine flow system, e.g., from above the blade. The nozzle located at the channel's beginning allows for steam expansion and this involves a reduction in the steam temperature. The expanded steam flows through cooling channels along the blade height, where heat is collected by means of convection. Finally, the cooling steam is brought out through the blade root.

3. The problem physics

3.1. Flow through a de Laval nozzle

The first step should be to verify whether the concept of steam expansion in a de Laval nozzle is capable of reducing the fluid temperature to a level that ensures effective cooling. According to [6], the change in the channel cross-section (A) compared to the critical cross-section (A^*), as a function of the Mach number, can be described by Eq. (1):

$$\frac{A^*}{A} = (\mathrm{Ma})\left(\frac{2}{\kappa+1} + \frac{\kappa-1}{\kappa+1}(\mathrm{Ma})^2\right)^{-\frac{\kappa+1}{2(\kappa-1)}} . \tag{1}$$

As it can be seen, for each ratio of surface areas different from 1, two positive Mach number solutions may occur. The analysis of Eq. (1) indicates that after the critical cross-section is reached, even a slight increase in the cross-section surface area results in a supersonic flow. The relationships that describe the relative change in temperature and pressure depending on the Mach number can be quoted according to [6].

Relative change in temperature is

$$\frac{T}{T_1} = \frac{1}{1 + \frac{\kappa-1}{2}(\mathrm{Ma})^2} \tag{2}$$

and relative change in pressure is

$$\frac{p}{p_1} = \frac{1}{\left(1 + \frac{\kappa-1}{2}(\mathrm{Ma})^2\right)^{\frac{\kappa}{\kappa-1}}}, \tag{3}$$

where index 1 stands for the inlet plane.

Equations (2) and (3) are presented in the chart (Fig. 2) as the curves which illustrate changes in the static temperature and static pressure, respectively, as a function of the Mach number.

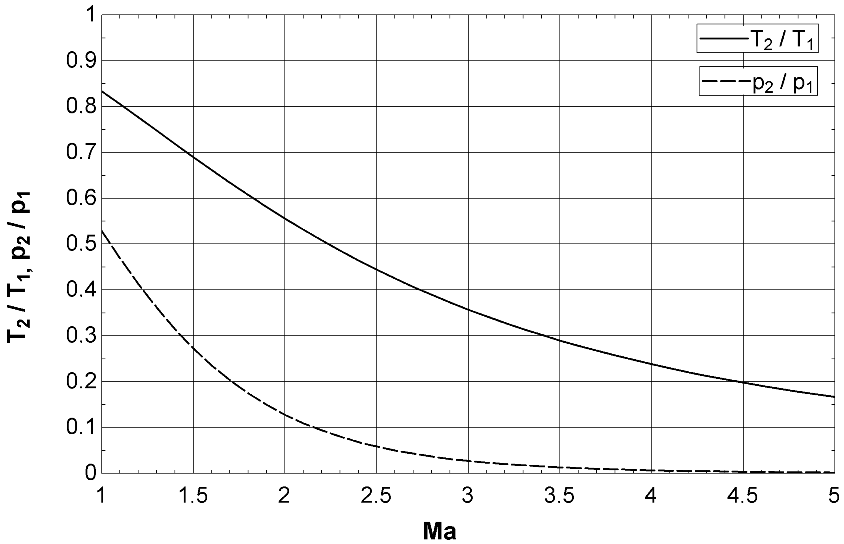

Fig. 2. Relative change in temperature and pressure depending on the Mach number (subscripts: 1 – inlet plane, 2 – outlet plane).

The considerations presented above concern the flow of an ideal gas in an unheated nozzle. At this stage it can be seen that the gas expanded in the nozzle may have the potential to cool the blade.

In the case of self-cooling, the nozzle would be heated additionally with a heat flux density varying along the profile. Therefore, in order to determine the real temperatures of the fluid, a decision was made to perform a numerical analysis using the Ansys CFX software package. The analysis may have a verifying nature for the computational model with the nozzle heating excluded.

3.2. Flow in the cooling channel

The channel is heated and has a constant cross-section along its entire length. By additionally assuming that there is no friction on the outlet channel walls, the flow satisfies the Rayleigh flow conditions [6].

At this stage, the volume of changes in the gas parameters due to the heating in the channel should be verified. A special emphasis is given to determining the temperature and the Mach number in the outlet cross-section.

By writing the energy balance for the control volume comprising the entire channel, the following is obtained:

$$T_1 + \frac{w_1^2}{2c_p} + q_{1-2} = T_2 + \frac{w_2^2}{2c_p}. \tag{4}$$

The outlet velocity w2 may be found directly from the Mach number definition:

$$w_2 = (\mathrm{Ma})_2\sqrt{\kappa R T_2}. \tag{5}$$

Using the momentum equation for the ideal gas, the following ratio can be written for the outlet and inlet cross-sections:

$$\frac{p_2}{p_1} = \frac{1+\kappa(\mathrm{Ma})_1^2}{1+\kappa(\mathrm{Ma})_2^2}. \tag{6}$$

Similarly, the expression for changes in the static temperature between the cross-sections is obtained from the continuity equation:

$$\frac{T_2}{T_1} = \frac{p_2^2(\mathrm{Ma})_2^2}{p_1^2(\mathrm{Ma})_1^2}. \tag{7}$$

Substituting Eq. (6) into Eq. (7), the relationship is obtained that describes changes in the static temperature depending only on gas and the Mach number in boundary cross-sections:

$$\frac{T_2}{T_1} = \left(\frac{1+\kappa(\mathrm{Ma})_1^2}{1+\kappa(\mathrm{Ma})_2^2}\right)^2 \frac{(\mathrm{Ma})_2^2}{(\mathrm{Ma})_1^2}. \tag{8}$$

Due to the heat transfer, the Rayleigh flow is not isentropic. Equation (9) defines the change in the gas entropy as the function of the Mach number:

$$\Delta S = ln\left[(\mathrm{Ma})^2\left(\frac{\kappa+1}{1+\kappa(\mathrm{Ma})^2}\right)^{\frac{\kappa+1}{\kappa}}\right]. \tag{9}$$

Introducing the additional dimensionless critical temperature ratio:

$$\frac{T}{T^*} = \left[\frac{(\kappa+1)\,(\mathrm{Ma})}{1+\kappa(\mathrm{Ma})^2}\right]^2 \tag{10}$$

it can be seen that for each ratio different from 1, two solutions are obtained: a subsonic and a supersonic one. As it can be noticed, the fluid heating should involve an increase in the static temperature and a drop in the Mach number. If the channel is sufficiently long, it should approach the value of 1.

Equations (4), (5) and (8) constitute a three-equation system with four unknowns in the form of the heat flux q1-2, temperature T2, velocity w2 and the Mach number M2. The missing equation defines the amount of heat supplied into the channel (q1-2). Due to complex conditions of the heat transfer, it is very difficult to precisely determine the density of the heat flux. Therefore, a decision was made to perform the numerical computations of the heat transfer and flow in the channel. For this purpose, a computational model was prepared to solve the conjugate heat transfer (CHT) problem involving the flow of cooling steam and heat in the blade material.

To extract the heat from the metal, the adiabatic wall temperature should be lower than the actual wall temperature. The adiabatic wall temperature in gases is always lower than the free-stream stagnation temperature [6] and its value is defined with a quantity known as recovery factor. Recovery factor coefficient describes the real value of the adiabatic wall temperature according to a free-stream stagnation temperature, and highly depends on the thermal properties of the gas. For the flow with the Prandtl number close to 1 (e.g., in a steam flow inside the cooling channel) recovery factor could be assumed as the function of the cube root of Prandtl number.

4. Numerical model

4.1. Geometry

The analyzed model represents a quarter of the channel volume, which is connected to the part of the blade wall through which/where steam flows pass. The model is composed of a short section of a straight channel after which a convergent-divergent nozzle is placed. Further on is the cooling channel, where heat is collected from the blade (Fig. 3).

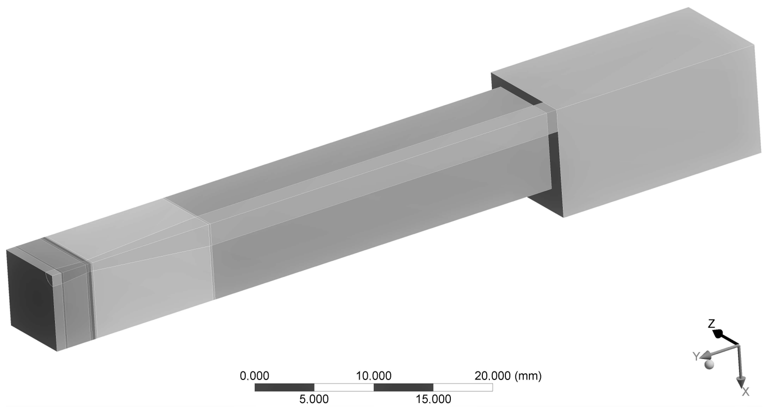

Fig. 3. Geometrical model under analysis.

As the possibility of using a de Laval nozzle for the blade cooling has already been identified, and due to the early stage of our research, such a model may be considered as sufficient. Additionally, a collective chamber with rectangular walls is placed at the nozzle outlet to shift the plane of the outlet boundary condition away from the nozzle edge (to avoid the impact of shock waves generated due to secondary expansion of the medium downstream the nozzle).

4.2. Model discretization

The mesh (Fig. 4) of the investigated geometrical model was made using the ANSYS ICEM CFD program. In the convergent-divergent nozzle area and in the outlet channel region the hex dominant mesh was generated.

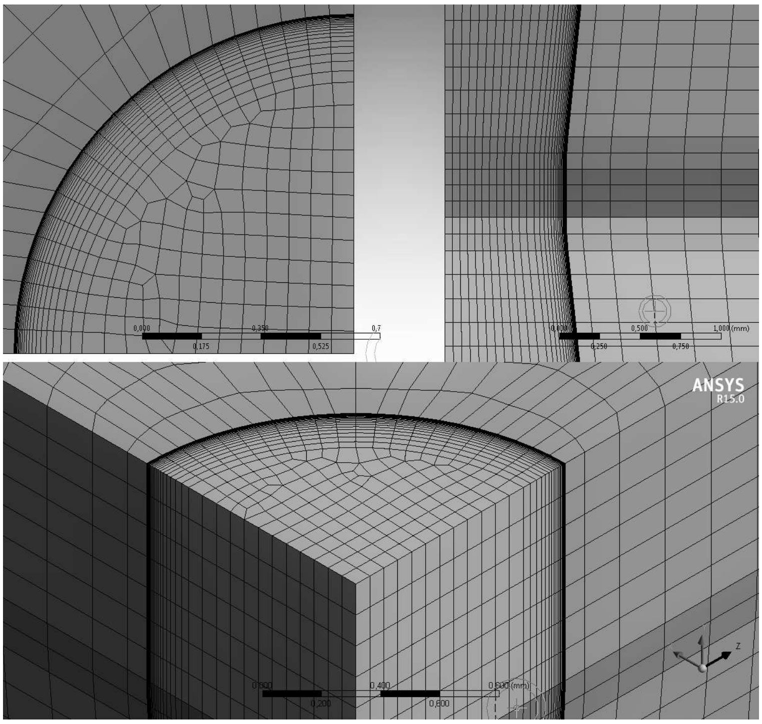

Fig. 4. Model discretization.

In order to determine the temperature profile in the boundary layer as precisely as possible, the area is described by 22 layers of elements with a thickness increment of 1.2 and a total thickness of 0.2 mm. The total number of elements in the area is 208 760. The blade area is modelled using the hex-dominant mesh with the element dimension of 0.2 mm. The mesh generated in this way is composed of 399 480 elements. The mean skewness of the mesh elements is 0.18, the mean shape factor – 5.09 and the mean orthogonality – 0.89. The parameter y+ assumes the values of about 3.5. By taking into account the supersonic character of the flow and water vapour as the medium, this value is considered to be sufficient.

4.3. Boundary conditions and model definition

Solving the problem required creation of a model composed of the following equations:

- the energy conservation equation for a solid,
- the mass, momentum and energy conservation equation for the fluid,
- the turbulence model equations,
- the gas state equation,
- the equations defining the material properties depending on temperature.

The computations were performed using the SST (shear stress transport) turbulence model [7]. This model is an extension of the two-equation k–ω model, which takes into account the transport of tangential stress in the fluid. Moreover, it allows a precise determination of the pressure distribution on the channel surface, the distribution of tangential stresses and the velocity profile for flows with an inverse pressure gradient. Owing to that, the model also allows a precise forecast of the stream separation from the channel wall, which is of great importance in computations focused on the heat transfer.

Special attention was given to modelling the heat exchange. Due to the predicted occurrence of the flow supersonic velocity, the full form of the Fourier-Kirchhoff equation (11) is used [8]. Because the model is solved in the steady state, the time terms of the equation are omitted.

$$\frac{\partial\left(\rho h_{\mathrm{tot}}\right)}{\partial t}-\frac{\partial p}{\partial t}+\nabla\cdot\left(\rho h_{\mathrm{tot}}\right)=\nabla\cdot\left(\lambda\nabla T\right)+\nabla\cdot\left(U\cdot\tau\right). \tag{11}$$

In the case of flows with the Mach number higher than approximately 0.3, the kinetic energy resulting from the flow velocity has a substantial impact on the fluid enthalpy. Therefore, total enthalpy, defined by Eq. (12), is used in the computations:

$$h_{\mathrm{tot}}=h+\frac{1}{2}U^{2}. \tag{12}$$

Another effect of the high velocity of the flow is the heat generation due to tangential stresses in the fluid (the impact of fluid viscosity). This is taken into account in the computations as the last term of Eq. (11). The parameters of steam as a real gas are determined based on the IAPWS-97 formulation.

In the considered numerical case, the boundary conditions were as follows. For the steam flow through the nozzle, the inlet overpressure of 30 MPa (with reference pressure of 1 atm) and the stationary temperature of 700°C are assumed. Zero overpressure is taken in the outlet plane (to avoid impact of the numerical boundary condition on expansion inside the nozzle). The computations are carried out with the symmetry condition on the other surfaces.

Convective heat transfer is considered on the blade outside surface, assuming the fluid temperature of 700°C. As there are no correlations describing the Nusselt number for supercritical parameters of a steam flowing past the steam turbine blade, a decision was made to use the results of the analysis of such a flow presented in the literature. According to [4], an averaged value of the heat transfer coefficient of 45 kW/m^{2}K was used. The symmetry condition is adopted in the other planes (Fig. 5).

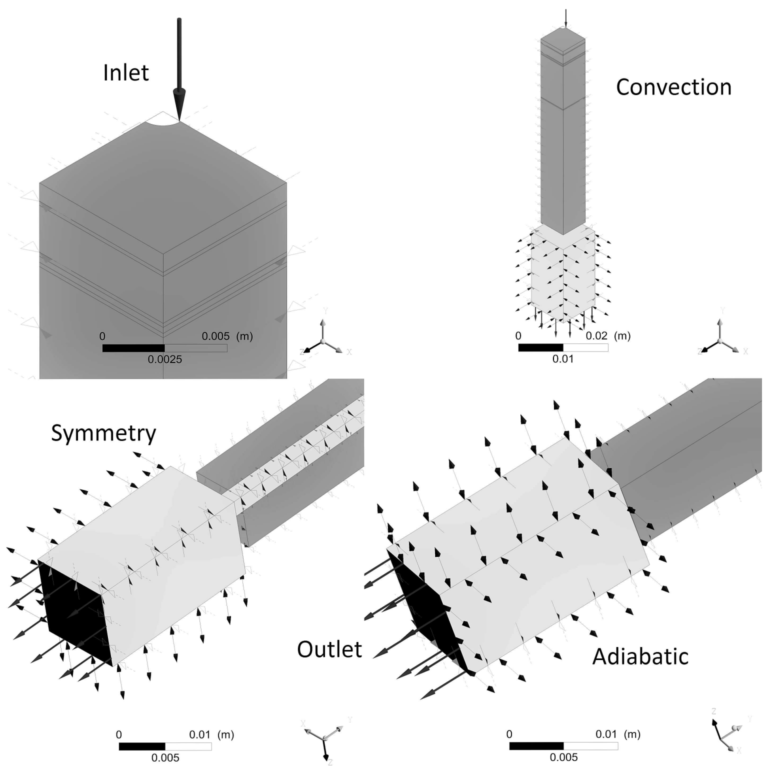

Fig. 5. Distribution of boundary conditions on the model surface.

5. Analysis of the results

The numerical simulations made it possible to determine the steam flow parameters along the entire length of the channel, as well as the temperature field of a cooled element. Analyzing the distribution of the Mach number and static pressure along the cooling channel axis (Fig. 6), it can be stated that the obtained histories are close to the expected theoretical curve [6].

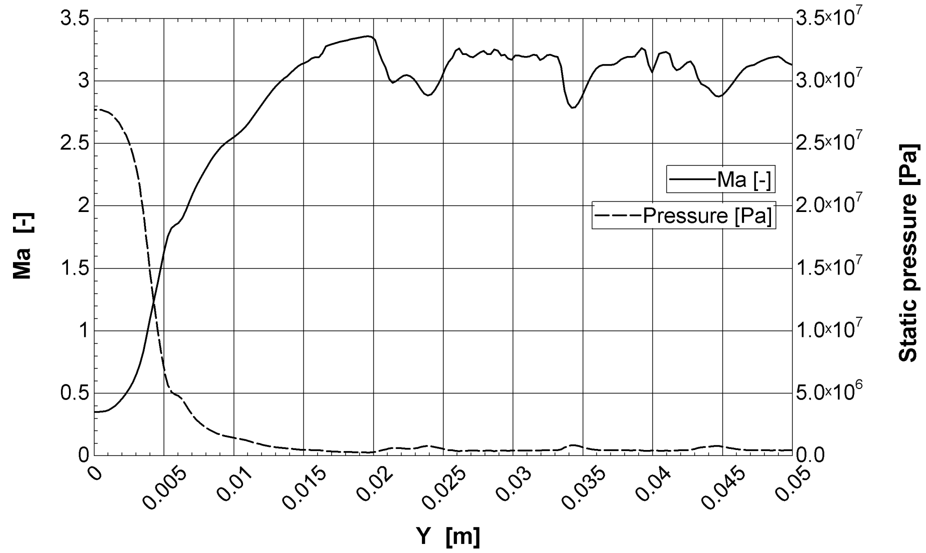

Fig. 6. Distribution of the Mach number and static pressure along the cooling channel axis.

In the inlet area, the medium flows at a subsonic speed. Acceleration to the speed of sound (the Mach number is equal to 1) can be seen in the convergent part. This point falls on the area of the nozzle's smallest cross-section, which proves that the analysis is correct. In the nozzle divergent part, the medium is accelerated further until it reaches the maximum (supersonic) speed, which occurs directly downstream the nozzle outlet. In the further area of the channel, the Mach number demonstrates a falling trend, which agrees with the physics of Rayleigh flow.

The static pressure distribution (Fig. 6) also agrees with the expected analytical curve. The medium is expanded abruptly inside the nozzle. The lowest pressure value occurs in the outlet cross-section of the divergent part. However, slight pressure variations can be noticed in the cooling channel area, which is the effect of a high level of flow turbulence and the complex processes of heat transfer.

The curve illustrating changes in the Mach number results from changes in static temperature presented in the chart (Fig. 7), which are inversely proportional to changes in the Mach number. The gas acceleration in the nozzle to a supersonic speed is accompanied by a distinct drop in static temperature. The static temperature lowest value occurs at the nozzle divergent part outlet. Downstream the nozzle the temperature tends to rise. This is the effect of both the heat collection from the cooled channel and the reduction in the medium velocity.

The chart (Fig. 7) also presents changes in the medium velocity in the channel. As it can be seen, they are inversely proportional to changes in static temperature, and close to the history of the Mach number values. The velocity fluctuations are much gentler compared to the changes in the Mach number. The medium is strongly accelerated inside the nozzle. Downstream the nozzle, velocity demonstrates a tendency to slightly drop.

With respect to the distributions of static temperature, the flow velocity (Fig. 7) and the Mach number (Fig. 6), it can be observed that the model is extremely sensitive to changes in pressure. Slight variations in static pressure have a considerable impact on changes in temperature, velocity and the local Mach number.

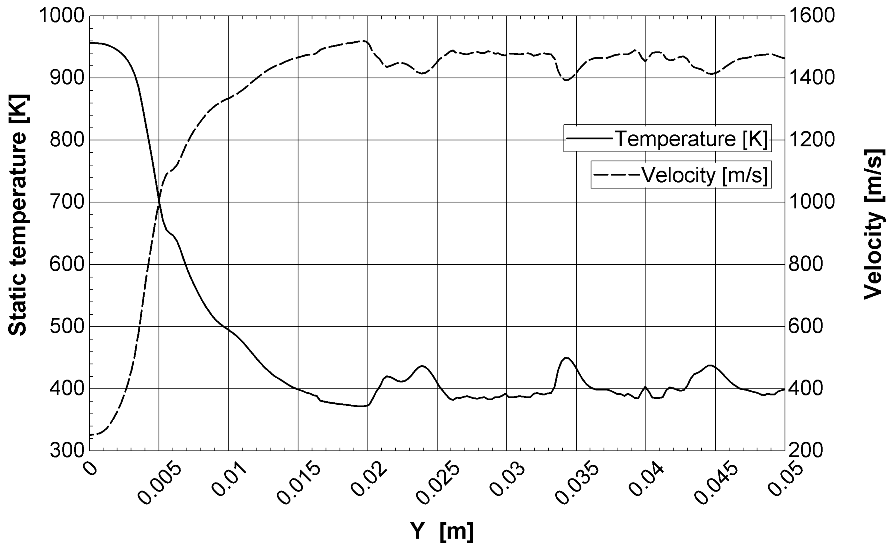

Fig. 7. Distribution of static temperature and velocity along the cooling channel axis.

The curves illustrating changes in individual parameters of the gas are similar to the known analytical solution. Characteristic points, such as the critical cross-section, are mapped correctly, which indicates that the performed computations of flow dynamics are correct, too.

Analyzing the heat transfer, attention should be drawn to the heat flux distribution on the steam-metal interface (Fig. 8). The medium expansion is rather abrupt and the process involves slight pressure gradients (Fig. 6) with variations in static temperature and the medium flow velocity corresponding to them (Fig. 7). This causes differences in the heat flux along the channel length (Fig. 9) and in the direction perpendicular to the flow axis.

The chart (Fig. 9) presents the distribution of the unit heat flux on the channel surface. It can be seen that in the nozzle convergent area the heat flux is negative, which means that the blade

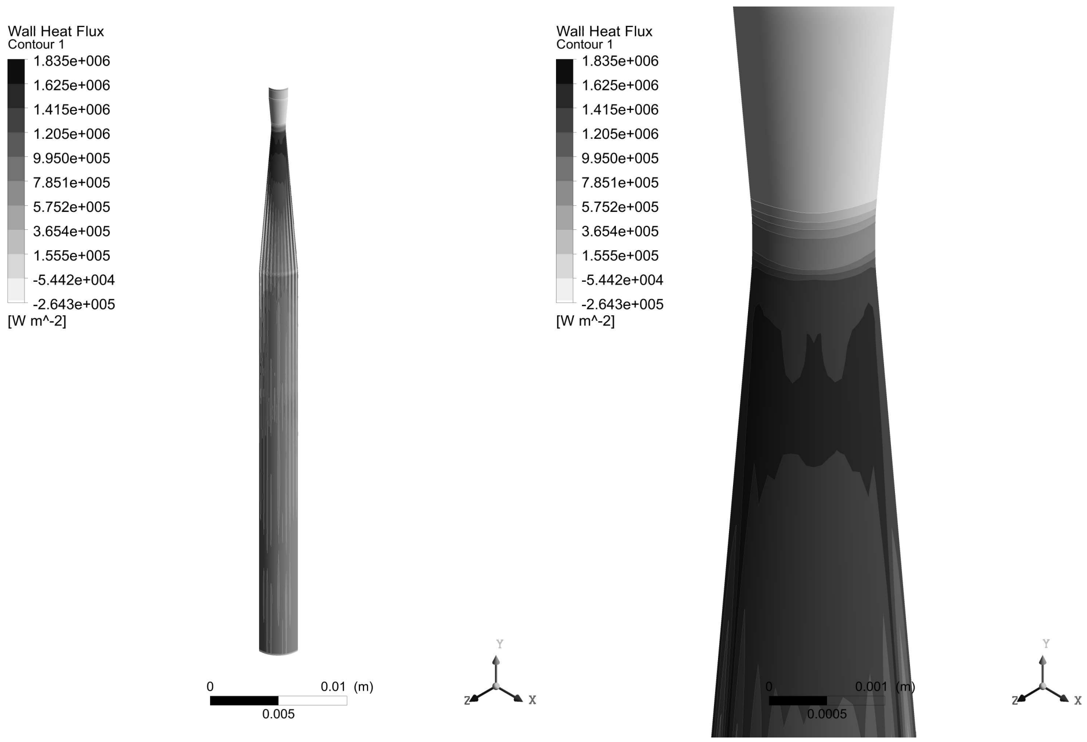

Fig. 8. Heat flux on the steam-metal interface.

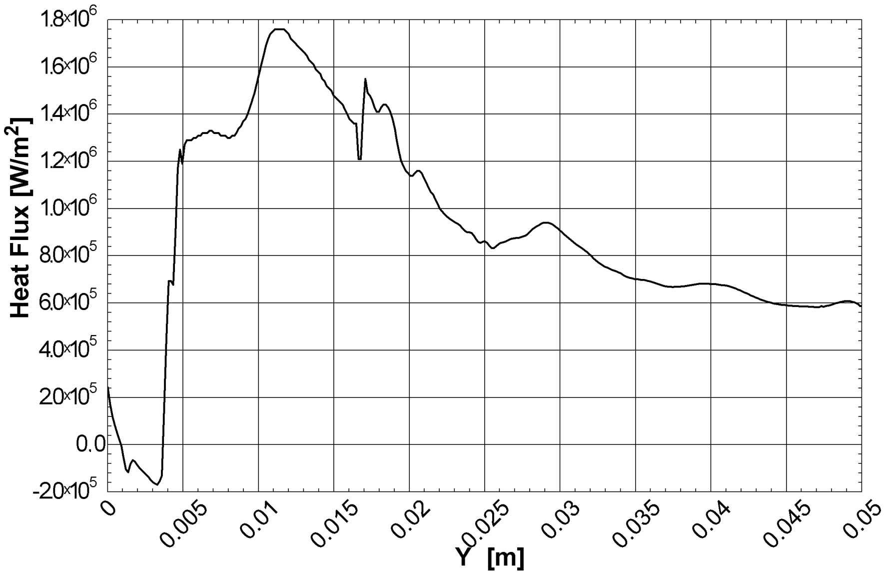

Fig. 9. Unit heat flux along the steam-metal interface.

is heated by inflowing steam. This is the result of insufficient expansion of steam combined with a considerable increase of the flow velocity in the subsonic region (intense forced convection). Downstream the critical cross-section, the heat flux becomes positive, i.e., steam collects heat from the blade material. The maximum value of the heat flux falls on the areas close to the divergent part of nozzle outlet, where the cooling medium temperature is the lowest.

The chart (Fig. 10 – left side) presents the distribution of temperature on the outside surface of the model representing the blade surface. The occurrence of a positive heat flux in the cooling channel area has an impact on the blade surface temperature. A fall in temperature can be observed

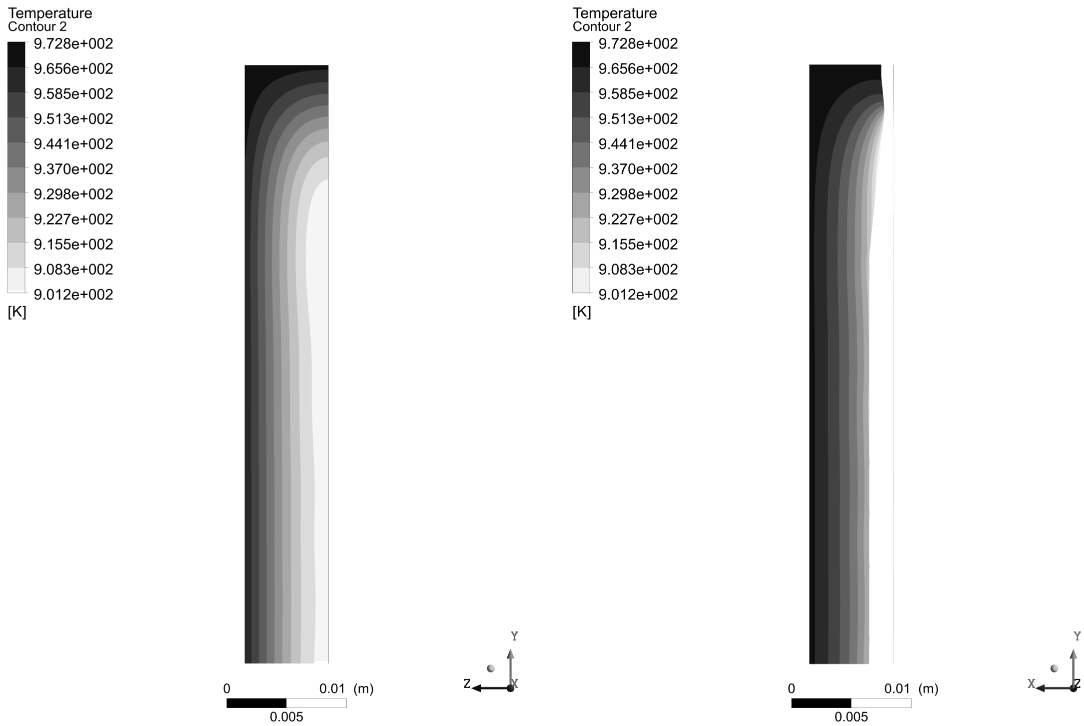

Fig. 10. Distribution of temperature on the surface representing the blade surface and in the cooling channel cross-section.

in the area where a cooling channel is located under the surface. Figure 10 (right side) shows the distribution of metal temperature in the cross-section going through the cooling channel centre.

The assumed intense heat transfer on the outside surface of the blade leads to a big temperature gradient across the wall, which may produce considerable thermal loads of the blade material. This indicates that there is a need for a more precise determination of the heat transfer conditions on the outside surface of the blade, or for a solution of the conjugate heat transfer problem, comprising also of the flow past the blade profile.

The chart (Fig. 11) presents the temperature profile in the direction perpendicular to the flow axis, determined halfway through the channel length. A large temperature gradient occurs in the boundary layer area. This results from the fact that in this flow region the medium is slowed down, which restricts the heat transfer. Inside the metal, the temperature profile is linear.

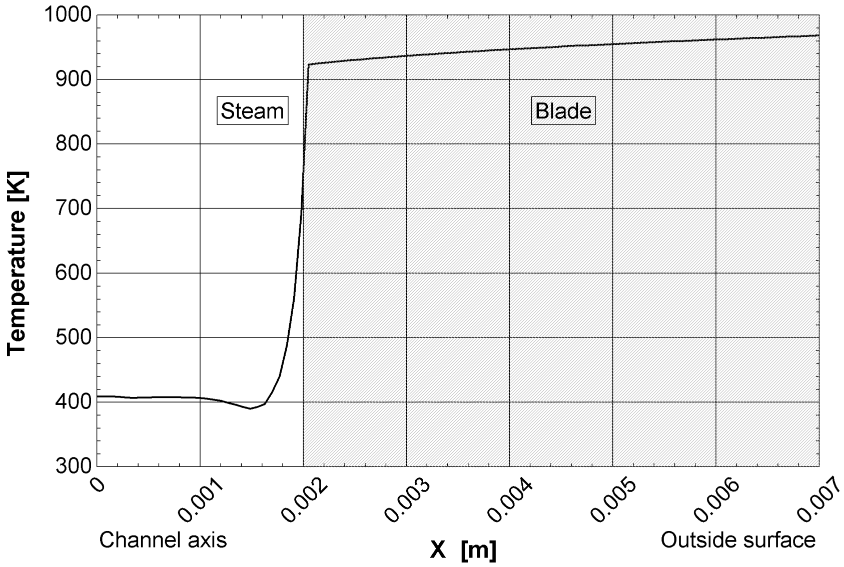

Fig. 11. Distribution of static temperature in the cross-section, halfway through the channel length.

6. Conclusions

The computation results indicate that the presented cooling system has the potential to reduce the turbine blade temperature. However, at this stage of research, the ultimate reduction in the blade temperature cannot be specified unequivocally and the results presented herein have a qualitative character only. Taking into account the small diameter of the channel and the blade intense heating from the working medium, it may be concluded that in order to effectively lower the supercritical steam turbine blade temperature, the cooling channels would have to be located relatively densely, at a short distance from the outside surface. Such a concept is presented in [9], but for conventional cooling of the gas turbine blades only.

In the analyzed case, high velocity of the cooling medium (of order of 1300 m/s) substantially limits the cooling potential (due to secondary heating of the medium as it is slowed down in the boundary layer). The optimal geometry would make it possible to achieve a low static temperature of the medium together with a flow velocity causing no excessive disturbance to the heat exchange process.

Another aspect worth developing is the heat transfer on the working medium side. The value of the heat transfer coefficient assumed for the computations is not good enough to appropriately represent the conditions of the blade operation. In reality, the heat transfer along the blade profile is more complex and the heat flux varies in a wide range. Therefore, the computations would have to be made for the flow past a full-scale blade with cooling channels located inside.

Acknowledgments

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