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NUMERICAL MODEL OF THE AXIAL TURBINE BLADE COOLING PROCESS

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ABSTRACT

The article describes influence of finite-element mesh flow parameters in the blade passage of cooled turbine stator blades on results of film cooling efficiency factor calculation. Recommendations for value selection of these parameters to build simplified numerical models of flow in stator blade rows in Numeca Fine Turbo software package with simulation of coolant blowing out from cells on blade surfaces are offered.

INTRODUCTION

The development of advanced gas turbine engines (GTEs) inevitably faces the task of increasing the aerodynamic efficiency of turbines, since they have a direct impact on the specific fuel consumption, thereby determining the competitiveness of GTEs. Reduction of specific consumption is achieved first of all by increasing efficiency of thermos-aerodynamic cycle of the engine - by increasing gas temperature before the turbine, which allows applying higher values of pressure ratio (Fig. 1). Overall, this results in an increase in specific cycle work and an increase in effective efficiency [1].

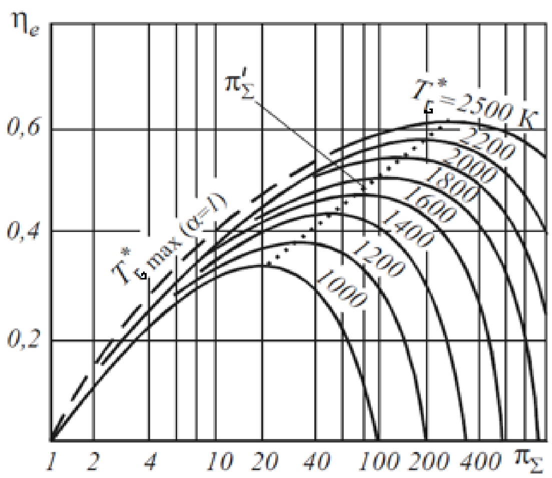


Figure 1 – Dependence of the effective efficiency of a gas turbine engine η_e on the compressor pressure ratio (π_Σ) at different gas temperatures before the turbine T_g^*

However, an increase in gas temperature before the turbine is limited by acceptable temperatures of alloys from which turbine parts and, first of all, stator blades and working blades are made. Therefore, in order to provide operability and necessary resource of turbines, systems of convective and convective-film cooling of their blades are applied [2]. The convective cooling of blades provides an opportunity to increase the inlet gas temperature to T_g^*=1300 K, and the convective-film cooling - to T_g^*=1800 ... 2000 K [1]. At the same time film cooling allows to increase the efficiency of blades cooling up to two times [3].

In order to avoid the blades burnout the cooling air veil must be distributed efficiently over the surface and should reliably protect against overheating, which is quite a complex task for the engineer.

The mentioned problem is currently solved by means of numerical simulation of film cooling [4]. There are various ways of simulation of turbine blade cooling, presented, for example, in papers [5-8]. However, all of them require increased computational and time resources. This causes the need for simplified film cooling models, providing, on the one hand, acceleration of calculations and, on the other hand, the results satisfactorily coinciding with the experimental data.

The Ansys CFX [9] and Numeca Fine Turbo [10] software packages allow such simplified numerical models to be created. They simulate the blowing of cooling air through perforations to provide film cooling, like the blowing onto the blade surface from the mesh model cells used to estimate the aerodynamic parameters of uncooled blade rows.

Unfortunately, so far there is insufficient information in the technical literature available to the authors to build such numerical models that are suitable for the physical pattern of blade film cooling. In particular, there are no recommendations for the configuration of the mesh model in the area of cooling air perforations. In this connection the authors of this work have carried out investigations of influence of finite-element mesh flow parameters on numerical simulation of film cooling of stator blades, on the basis of which recommendations on selection of mesh model configuration in the area of cooling air supply perforations are offered.

EXPERIMENTAL DATA ON FILM COOLING EFFICIENCY OF STATOR BLADES

The results of the experimental determination of the film cooling efficiency of the stator blades, provided by Kuznetsov PJSC, were used to work out the configuration of the mesh flow models in the area of the cooling air supply perforations.

Tests were carried out on three stator blades (SB) units, consisting of 3 blades (Fig. 2). Temperature measurements were taken on the middle blade of the unit. Cooling system of each blade included subsystems:

- convective-film cooling of the leading edge, pressure and suction side;
- convective cooling of the trailing edge.

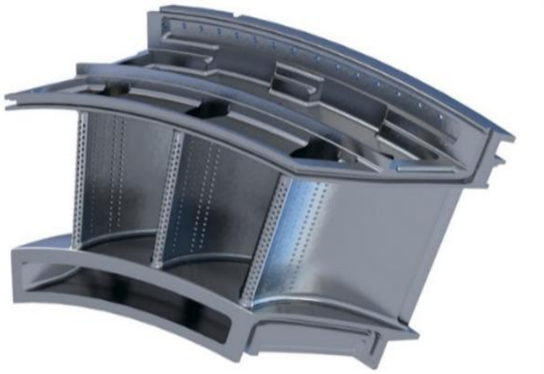


Figure 2 – Experimental stator blade unit

The convective-film cooling subsystem consisted of a blade front cavity with two deflectors in the front cavity and ten lines of cooling air blowing perforations (Fig. 3). The air supply to the front cavity was via the hub and peripheral (shroud) end of the blade.

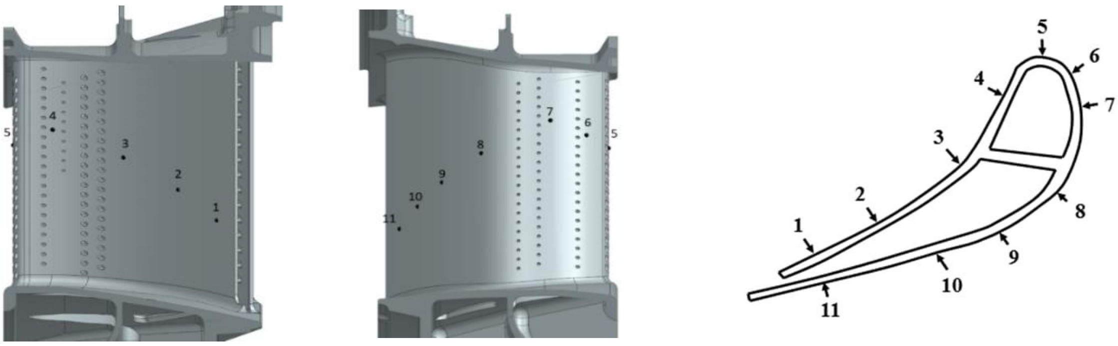


Figure 3 – Blade views of pressure side (a) and suction side (b), with thermocouple arrangement (c)

The convective cooling subsystem of the trailing edge was designed as a back cavity of the blade, a vortex matrix and cooling air blowing slots in the trailing edge (Fig. 3). Air supply to the back cavity was also provided via the hub and peripheral (shroud) end of the blade.

The experiment investigated the effectiveness of only film cooling of the leading edge, pressure and suction sides, while the back cavity was completely filled with asbestos.

The SB sector was blown with hot air with inlet temperature $t_G^* = 850°C$ (1123 K) and overpressure $p_{G\ over}^* = 0.7$ atm (68 646 Pa). Static pressure at the SB sector outlet corresponded to atmospheric pressure and was approximately 1 atm (98066 Pa). Moreover, isentropic flow velocity at the SB outlet was equal $\lambda_{1s} = 0.92$, and Reynolds criteria Re = $3.5 \cdot 10^6$.

The cooling air inlet temperature at the front cavity of the blade was 380°C (653 K) and overpressure $p_{cool.over}^* =$ 0.734 atm (71 980 Pa).

During the tests, surface temperatures were measured on the middle blades of the three SB units at 11 points, as shown in Figure 3.

By measuring the temperatures at each of the 11 points on the surface of the three blades, as well as the hot and cold air temperatures, the cooling efficiency factor was determined according to the formula:

$$\theta = \frac{T_G^* - T_{bl}}{T_G^* - T_{cool.}^*}, \qquad (1)$$

where T_G^*, K – total temperature of the main flow at the SB inlet; T_{bl} , K– flow temperature at the blade surface; $T_{cool.}^*$, K – total temperature of cooling air at the blade cavity inlet.

These coefficient values θ are shown in Figure 4. The mean values of the coefficient θ_{mid} and their confidence limits are also given here, at a confidence probability of 95%. The error value was calculated by the authors using standard data processing techniques on the basis of available measurement results. Unfortunately, the error was significant due to the small number of measurements, differences in the blade geometry and probably unsteady effects during the experiment.

In addition, it was not possible to measure temperatures at points 1 and 5 on one of the SB units. Therefore, these points had to be eliminated from further analysis.

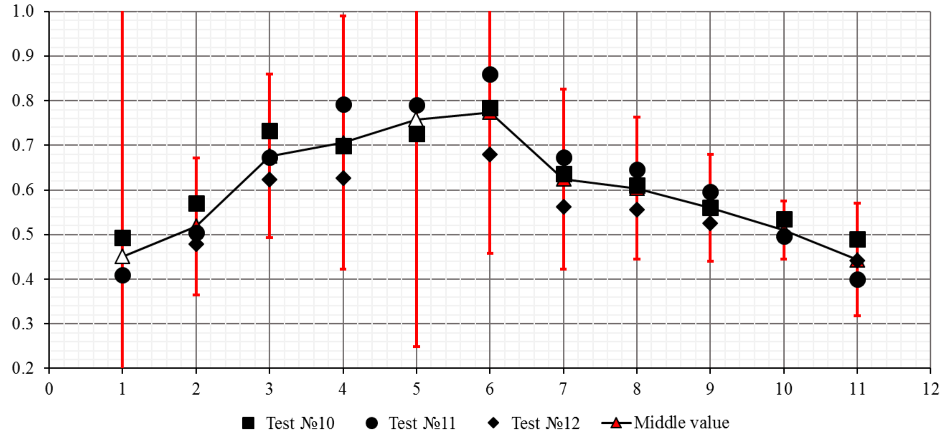


Figure 4 – Results of experimental determination of the cooling efficiency coefficient

NUMERICAL SIMULATION OF FILM COOLING WITH ANSYS CFX

Numerical modelling of film cooling of blades of SB block (see Fig. 2) in software complex Ansys CFX [9] was conducted in order to define coolant flow rate through blowing perforations, necessary for further investigations of models with simulation of coolant blowing from cells on blades surface, and also coefficient values θ at thermocouple installation points on surface of stator blades.

Initially, a solid model of the SB blade with front cavity and cooler blowout perforations was prepared. A finite element mesh model was then generated using the Ansys Workbench software to calculate the flow around the blade, as well as the flow in the front cavity of the blade and the blowout perforations.

The numerical model was created in the ANSYS CFX software package (Figure 5). Turbulence model is SST with Fully Turbulent function. The total pressure and temperature were set at the inlet and the static pressure at the outlet. For cooling air of film and convective cooling, flow rate and temperature were set at the inlet. Working fluid is an ideal gas with specific heat capacity and dynamic viscosity as a function of temperature, the blade material is ZhS6U. The numerical values of the boundary conditions corresponded to the experimental conditions.

The appearance of the computational area with the given boundary conditions is shown in Figure 5. It should be noted that the CFX numerical model turned out to be very "heavy", the computational area consisted of 34 million elements and the computation time was approximately 18 hours on 80 processors.

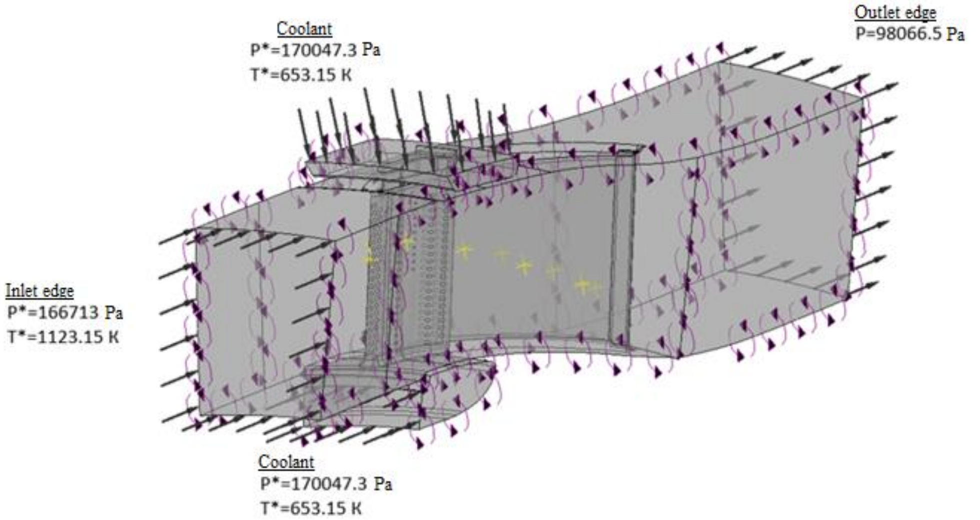


Figure 5 – Appearance of the calculation area with boundary conditions

As a result of the CFX model calculation, the coolant flow rates through all the blowout perforations (Fig. 6) and the factor values θ (Fig. 8) were obtained. As can be seen, practically all (with the exception of the value at point 6) calculated coefficient values θ are within the confidence limits of the experimental results θ_{mid}.

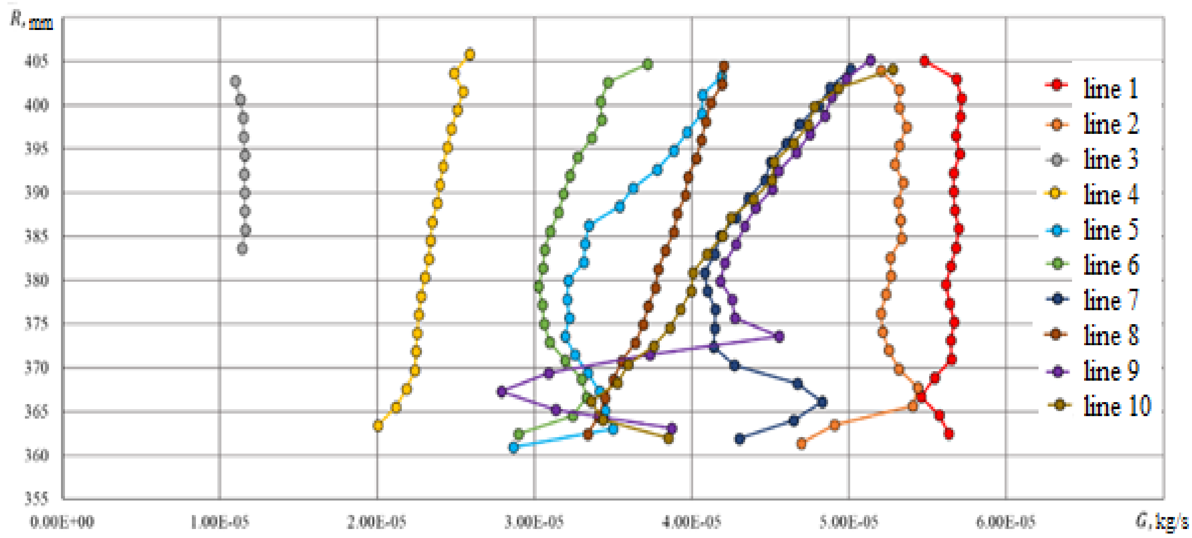


Figure 6 - Radial distribution of the coolant flow rates in the blowout hole lines (The numbering of the film cooling holes is shown in Figure 7)

The radial distributions of the cooling air flow rates (Fig. 8) were subsequently used to set the boundary conditions in a simplified numerical film cooling model in the Numeca FineTurbo software package [10].

STUDY OF THE INFLUENCE OF FINITE ELEMENT MESH PARAMETERS OF SIMPLIFIED GAS FLOW MODELS, SURROUNDING THE COOLED STATOR BLADES, ON THE CALCULATED FACTOR VALUES θ

Initially, the influence of the parameters recommended in [11] for forming finite element meshes of flows in uncooled blade rows was investigated, specifically, the influence of the parameters B2B, MR, ER.

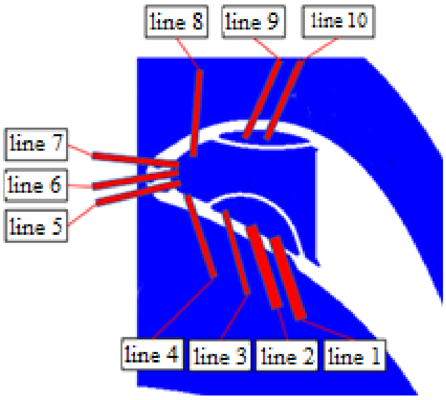


Figure 7 - Numbering of film cooling hole lines

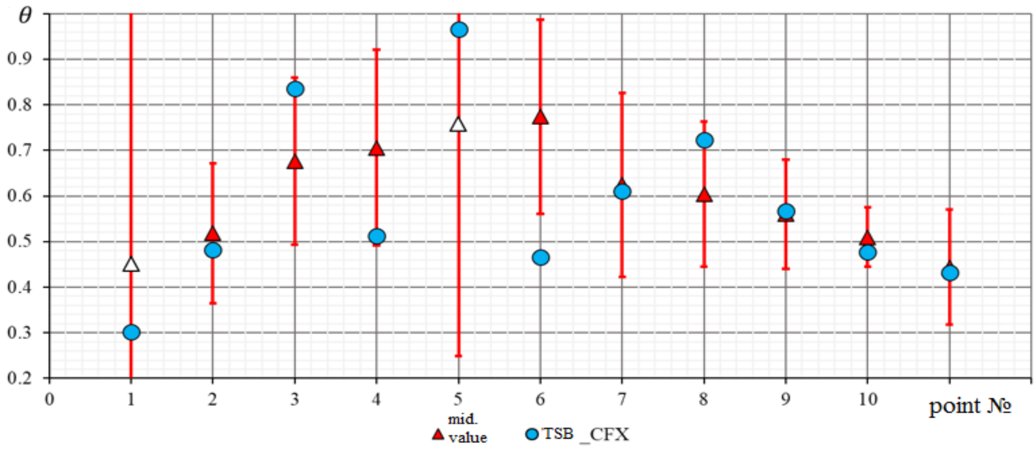


Figure 8 – Comparison of the results of the film cooling efficiency calculation with experimental data

The B2B parameter describes the average number of elements in the cylindrical sections of the blade passage. The MR and ER parameters define the mesh configuration along the height of the flow path. The MR parameter is the ratio of the highest height $y_{FP\ max}$ of the element located along the radius in the middle of the flow path to the minimum height $y_{FP\ 1}$, nearest to the wall end (торец) of the element (Figure 9):

$$MR = \frac{y_{FP\ max}}{y_{FP\ 1}} \qquad (2)$$

The ER parameter (element growth coefficient from the end wall) characterises the densification of the mesh over the height of the flow path (Figure 9):

$$ER = \frac{y_{FP\ i}}{y_{FP\ i-1}} \qquad (3)$$

In addition, the recommendation that the height of the first element at the end wall $y_{FP\ 1}$ is equal to the width of the first element at the blade profile $y_{B2B\ 1}$ has been used in the construction of the meshes, which generally provides the equality $y^+{}_{FP\ 1} = y^+{}_{B2B\ 1}$.

In total, 14 variants of mesh models were prepared, with B2B varying from -2 to 2, MR from 125 to 2000, and ER equal to 1,2.

Simplified numerical flow models in the Numeca FineTurbo software have been made with the 14 mesh options and values have been determined at eleven points on the surface of the stator blade mentioned above. It turned out that the results of calculation with different meshes do not differ much from each other. The values obtained with the three meshes are shown in Fig. 10. The first one has B2B = -2 and MR = 2000, the second one has B2B = 0 and MR = 500 and the third one has B2B = 2 and MR = 125.

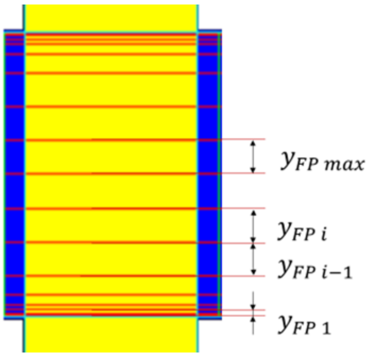


Figure 95 – MR and ER parameterization schematic

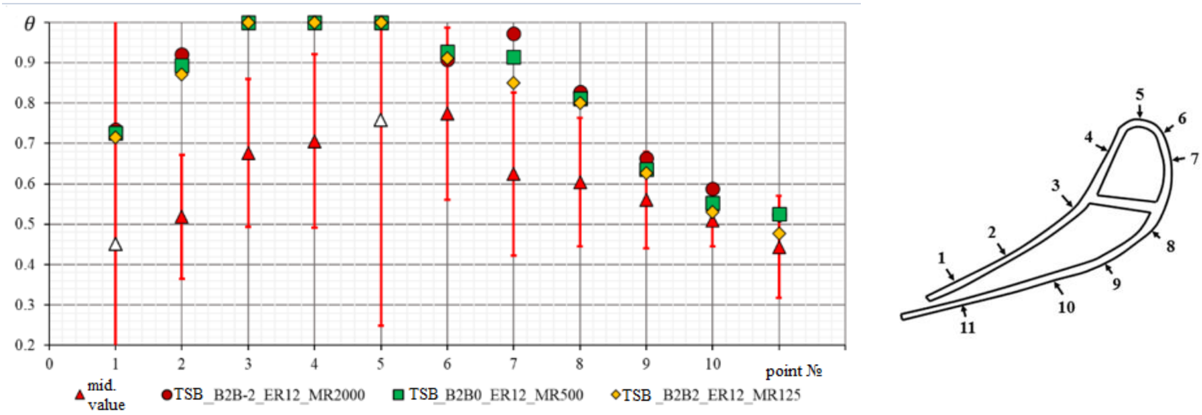


Figure 10 – Comparison of coefficient θ values calculated by simplified models with experimental data and results of calculations in the Ansys CFX software package

It should be noted that the values θ found using all 14 mesh variants are larger than the experimental values of the factor θ. Most of the values θ obtained using the 14 mesh variants were outside the confidence limits of the experimental values θ_{mid}. This indicates insufficient accuracy obtained by calculation of the flow pattern surrounding the cooled stator blades.

Therefore, it was decided to adjust the configuration of the flow mesh models by increasing its densification in the area where the cooling air perforations are located.

For this purpose, two additional mesh parameters were introduced into the consideration: the NPW parameter, which was equal to the number of elements per perforation in the B2B section, and the NPH parameter, equal to the number of elements per perforation along the height of the flow path (Figure 11).

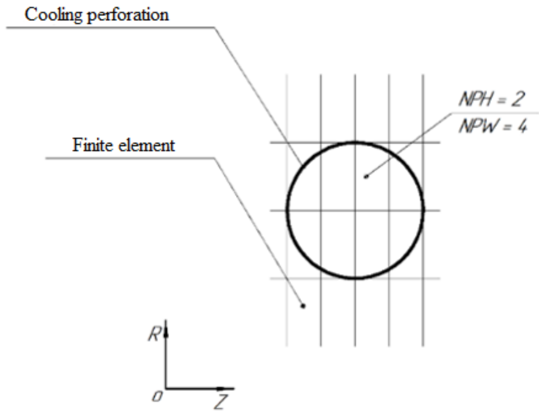


Figure 61 – NPW and NPH identification schematic

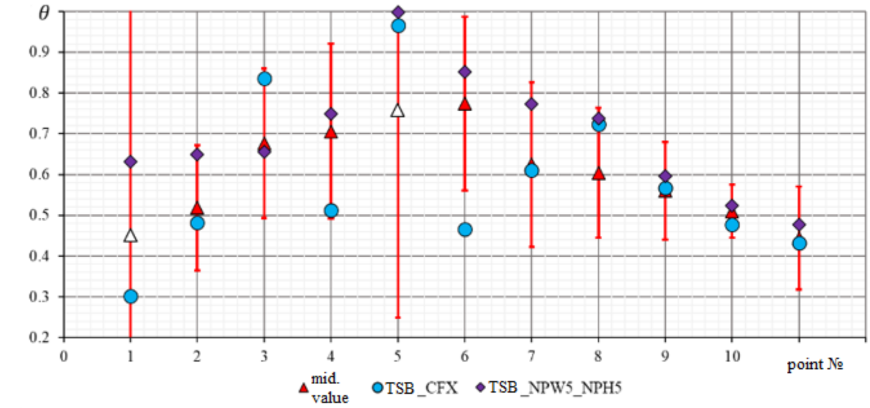
Additional parameters were used in the construction of new finite element meshes of the flow in the blade passage of the SB to determine the values of the factor θ. The NPW parameter took values 1, 3 and 5 and the NPH parameter took values 1, 3, 5, 7 and 9.

According to the results of the calculation studies, a change in the NPW parameter has practically no effect on the calculated value θ.

Increasing the NPH parameter from 1 to 5 leads to a reduction in the calculated value θ, especially noticeable at the blade pressure side (at points 1-4). For example, at point 2 the value θ reduces by 0.25 and at point 3 by 0.35. A further increase in NPN from 5 to 9 causes almost no change θ.

The calculated values θ obtained with the help of a mesh with NPH parameter taken as 5 (at NPW = 5), at 11 points on the surface of the stator blade, are shown in Fig. 12. As can be seen from this figure, all the values obtained in the calculation are within the confidence limits of the experimental results. Apparently, densification of the finite-element mesh in the area of the coolant blow-out allows to reproduce the flow pattern in this place more accurately and to receive calculation results θ appropriate to the experimental data.

It should also be noted that the resulting numerical flow model contains practically 6 times fewer elements than the "heavy" model in the Ansys CFX software package considered above, and the calculation time can be reduced by 5 to 7 times.



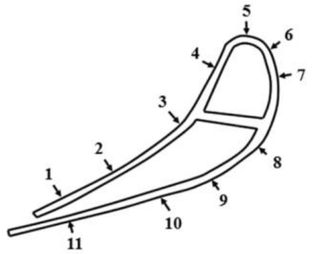


Figure 12 – Comparison of calculated values for film cooling efficiency factor at NPH = 5 and NPW = 5 with experimental data

CONCLUSIONS

As a result of this research it was found that it is not enough to use recommendations for selecting values of finite element mesh parameters to form numerical models of flow in the blade rings of uncooled turbines to obtain results of calculation of factor values corresponding to the experimental data, using a simplified numerical flow model in the software package Numeca Fine Turbo.

It is advisable to use a mesh with densification in the area of the blow out perforations. The number of cells per perforation along the blade height must be at least 5 (NPH $\geq$ 5) and along the blade profile at least 1 (NPW $\geq$ 1).

It should be noted that the use of simplified numerical flow models with simulated coolant blow-out from the cells on the surface of the blades can reduce the calculation time by about 5 to 7 times.

NOMENCLATURE

B2B – parameter that describe number of finite element at blade passage, that uses for calculation of uncooled turbines
MR – parameter that describe number of finite elements at height that uses for calculation of uncooled turbines
ER – parameter that describe of finite element expansion ratio at height channel
SB – stator blade
NPH – parameter that describe number of finite elements at height that uses for calculation of cooled turbines
NPW – parameter that describe number of finite element at blade passage, that uses for calculation of cooled turbines.

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