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OPTIMAL DESIGN OF HIGH PRESSURE STEAM TURBINE STAGE USING COMPUTATIONAL FLUID DYNAMICS

A. V. Boiko, D. I. Maksiuta

Turbine Projection Chair
National Technical University «Kharkiv Polytechnical Institute»
Kharkiv, Ukraine
anatoliboiko@yahoo.com, studkolo@gmail.com

ABSTRACT

This paper presents the results of the development and application of the shape optimization method of a high pressure steam turbine stage, which takes into account the nature of the flow around the turbine profiles and leakages through the radial seals. Computational fluid dynamic methods (ANSYS CFX) were used to calculate the stage flow characteristics. A comprehensive three-step verification of the computational model was carried out before its optimization. As a result of the optimization a new stage with an efficiency increase of 0.48 % compared to the original design was obtained. The reasons leading to this performance boost were analyzed and are presented here.

NOMENCLATURE

Symbols

A - matrix of formal metamodel coefficients
b - blade chord
d - mean diameter
G - mass flow rate
l - blade height
$\bar{l}$ - relative blade height
M - Mach number
m - varied parameter in the optimization
N - rotor speed
n - factors number of formal metamodel
P^* - total pressure
P - pressure
q - normalized varied parameters
r - radius
Re - Reynolds number
T^* - total temperature
t - cascade pitch
Y - response function
z - blades number
α - outlet flow angle in absolute motion
α_{st}, β_{st} - stagger angles of nozzle and blade cascades
β - outlet flow angle in relative motion
$\Delta\beta_g$ - inlet metal angle adjustment of blade cascade
δ - axial gap
ρ - degree of reaction

Indices

0 - stage inlet parameters
1 - parameters in the axial gap; nozzle wheel
2 - stage outlet parameters; blade wheel
inc - incidence
ij - integer values for indexation
init - initial
opt1 - first stage of the optimization
opt2 - second stage of the optimization

INTRODUCTION

Earlier the axial turbine stage shape optimization task considering leakages was solved in a two-dimensional axisymmetric formulation taking into account streamline's slope and curvature. The control parameters were mainly twist laws of the nozzle and blade cascades. Such a formulation allows to significantly increase stage efficiency (Boiko and Garkusha, 1999).

To simulate and make detailed analyses of the complex physical phenomena in turbine stage flow is impossible using a two-dimensional formulation. Therefore, in this investigation, 3D computational fluid dynamics were used to increase the accuracy of the calculation of the complex turbine stage flow field. However, before being able to start the optimization task numerical simulation and experimental results must be reconciled.

This paper presents the results of the complex work which contains the following: 3D calculation model building, verification of this model, development of the turbine stage shape optimization method (considering the nature of the flow around the turbine profiles and leakages through the radial seals) and application of the developed method.

MODEL TURBINE STAGE

A prototype of the third high pressure steam turbine stage of the turbine K500-65/3000 was selected as the studied stage in this research. The profiles used in the nozzles are TC-1A with extension, and in the blades they are active profile R2. The drawing of the model turbine stage is shown in figure 1.

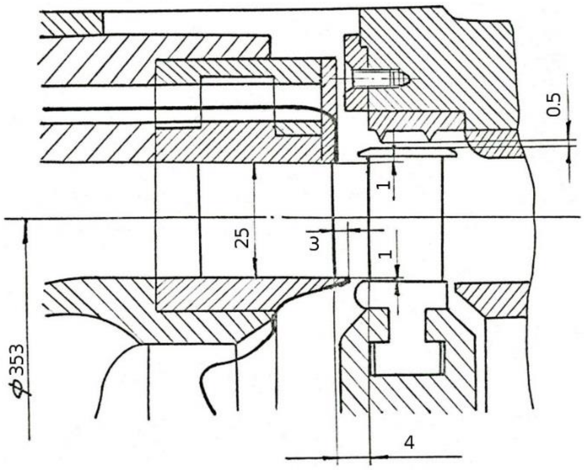

Figure 1: **The studied turbine stage drawing (all dimensions are in mm)**

The working fluid in the model turbine was air and the boundary conditions are presented hereafter: $P_0^* = 145500$ Pa, $T_0^* = 387$ ºK, $P_2 = 100300$ Pa. Such boundary conditions provide a subsonic flow in the entire stage (the Mach number does not exceed M = 0.78). The mass flow in the turbine was G = 1.63 kg/sec, Reynolds number at the mean radius Re_{c1} = 3.5*10^5. Flow characteristics were measured: in the axial gap – behind the nozzle cascade; behind the stage – at a distance of 16 mm from the blade cascade trailing edge. All measurements were done by traversing the flow by Pitot tube. The efficiency of the stage was defined by the measured torque on the rotor (Goncharenko, 1979). This method was used in all aerodynamic experiments with turbine stages at the Turbine Projection Chair which are mentioned in this paper. Therefore, while CFD simulating, the same measurements were done.

The geometrical characteristics of the stage can be seen in table 1.

Table 1: **Geometrical characteristics of the initial stage**

Parameter	**Nozzle**	**Blade**
Mean diameter (*d)*	353 mm	353 mm
Blade chord (*b*)	21.83 mm	16 mm
Stagger angle (α_{st}, β_{st})	55.1806º	13.17º
Relative cascade pitch (*t/b*)	0.747	0.722
Relative length (*l/b*)	1.145	1.69
Blades number (*z*)	68	96
Blade height (*l*)	25 mm	27 mm
Rotor speed (*N*)	8145 rpm	
Axial gap (δ)	4 mm; 9.7 mm	
Radial gap	0.5 mm	

NUMERICAL DATA VERIFICATION

Before the formulation of the shape optimization task, the verification of the computational model was done in three phases:

1) Verification of the 2D calculation of the isolated blade profile with different incidence angles;

2) Verification of the 3D calculation results of the blade incidence angle influence in the turbine stage;

3) Verification of the 3D calculation results of the turbine stage while working with the leakages at the radial seals.

All calculations for verification and future optimization were done using computational fluid dynamics (ANSYS CFX.)

Verification of the 2D calculation of the isolated blade profile with different incidence angles.

In the process of turbine stage optimizing both the nozzle and the blade cascades are changed. At the intermediate points of the experimental plan the nozzle cascade with outlet flow angles different from the originals can be obtained. This leads to a non-nominal flow around the blades (with significant incidence angles.) Obviously, it is necessary to research the incidence angle influence on the blade losses and to compare the obtained numerical results with the real experimental data.

To perform this task, the experimental data of the flow around the R2 turbine profiles were considered. The parameters of the profile and the boundary conditions were: $t/b = 0.722$, $b = 28.88$ mm, $P_1^* = 1.194219$ atm, $T_1^* = 20$ ºC, $P_2 = 1$ atm, working fluid – air (Meltiuhov, 1986). Such boundary conditions provide a subsonic flow with $M = 0.51$, and $Re = 4.3 \cdot 10^5$.

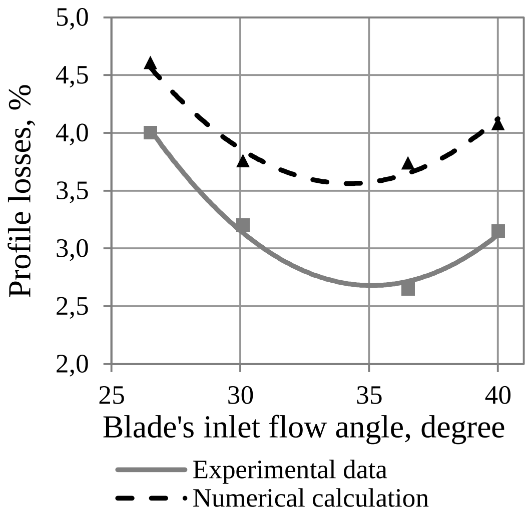

Figure 2: **Profile losses for the experimental data and the numerical calculations**

Experiments were carried out for the four different flow angles: 26.5º; 30.1º; 36.5º; 40º. The inlet metal angle for this profile is 29.5º. Numerical calculation results can be seen in figure 2. Evidently, the value of the profile losses coefficient in numerical calculations is overestimated. However, the numerical and the experimental dependencies are approximately equidistant lines. This means that the minimal profile losses points in both the numerical and the experimental calculations lean towards the same value of the blade profile flow angle, which is more important in the optimization task.

If it is needed to obtain the accurate absolute values of the profile loses, the coefficients in the turbulence model should be varied.

Verification of the 3D calculation results of the blade incidence angle influence in the turbine stage.

The next verification phase is the study of the influence of the blade’s cascade incidence angle on the flow characteristics in the working turbine stage.

In this verification phase numerical calculations of the initial turbine stage without leakages were performed. The stage losses were calculated and the blade inlet flow angle dependency at a distance of 1 mm from the blades cascade leading edge was also obtained (see figure 3.)

To create the shape of the blade which provides a zero incidence angle to the entire blade is technologically not practical because of the sophisticated form of the inlet flow angle curve from

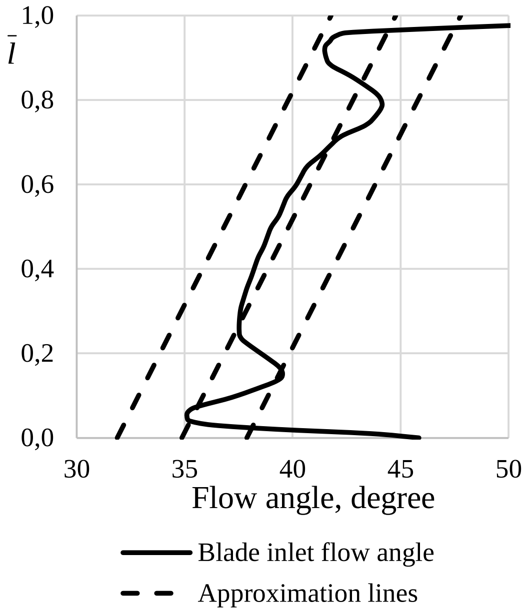

Figure 3: **Inlet flow angle distribution on the blade and approximation lines for different incidence angles**

the hub to the shroud. It was decided to approximate this curve by a linear law ignoring the complex flow near the hub and the shroud boundaries. The obtained, approximate line of the inlet flow angle height distribution has the smallest standard deviation (cf. figure 3.) To provide a zero incidence angle the blade profile was changed from the hub to the shroud according to the obtained approximate inlet flow angle distribution.

When the boundary conditions are specified, the geometry of the nozzle cascade is fixed, the rotor angular velocity and the blade throat are also unaltered, and the only opportunity to affect the blade incidence angle is to change the form of the blade profile by varying its inlet metal angle. To obtain the blade profile with the inlet metal angle which provides the specified incidence angle, the approximation line was moved to the value of the incidence angle without changing the angle of its inclination. Thus, the needed incidence angle for the entire blade was produced. The following incidence angles of interest are presented hereafter: –3º, 0º and +3º (a negative value corresponds to the blowing happening on the blade low pressure side while a positive value means the blowing was done on the blade high pressure side.) Approximation lines for all the considered incidence angles are shown in figure 3.

Table 2: **Stage losses while flowing around blades with different incidence angles**

Losses	**–3º**	**0º**	**+3º**
In nozzles	4.87022%	4.87055%	4.87410%
In blades	7.69816%	7.49868%	7.84988%
Total	12.56838%	12.36923%	12.72398%

The blade profile was modified by defining it with two Bezier curves on the low pressure side and three circles (leading edge, trailing edge and high pressure side). The control parameters of these curves allow to make all the described modifications. After changing the newly obtained profiles with the various inlet metal angles these are not significantly different from the original profiles (see figures 4, 5.)

The results of the numerical calculations are presented in table 2. This table shows that the changing of the inlet metal angle has a significant influence on the stage losses. The lowest value of the stage loss coefficient was found with a zero degree incidence angle which corresponds to common representations of the nature of the flow around turbine profiles.

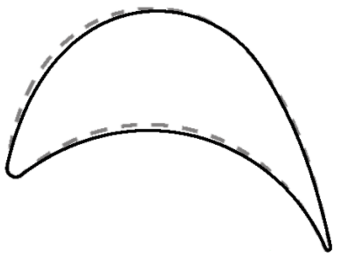

Figure 4: **Mean cut of the initial (——) and obtained (– – –) blade profile with incidence angle -3º**

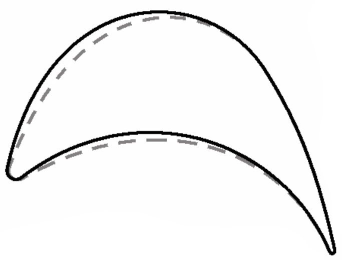

Figure 5: **Mean cut of the initial (——) and obtained (– – –) blade profile with incidence angle +3º**

In this way it was shown that using CFD it is possible to make a qualitative and quantitative assessment of the incidence angle influence on the blade losses while working on the turbine stage. Moreover, the method of blade building which contains the inlet metal angle change and its adaptation to the real inlet flow angle was tested.

Verification of the 3D calculation results of the turbine stage while working with the leakages at the radial seals.

This verification phase is the final one and represents a comparison between the turbine stage flow field numerical calculations and the experimental data (see table 3.) The turbine stage whose geometric characteristics are presented in table 1 with an axial gap value of 9.7 mm was taken as the object of interest for this verification phase. The enlarged axial gap is explained by the necessity of measurements of the flow parameters using Pitot tube.

Table 3: **Averaged stage working characteristics**

Parameter	Experiment	Calculation
Efficiency	83.80 %	82.71 %
Nozzle losses	3.15 %	4.93 %
Blade losses	8.15 %	7.53 %
Exit velocity losses	4.90 %	5.33 %
Mass flow	1.63 kg/sec	1.60 kg/sec

In addition to the averaged characteristics the local height distributions of the flow parameters in the axial gap and behind the stage were obtained.

The comparison of the averaged characteristics and the local height distributions of the flow parameters obtained in the CFD simulations with the experimental data have shown an acceptable qualitative and quantitative assessment of the turbine stage flow field in the real turbine stage and its numerical calculations while working with the peripheral leakage.

The verification of the 3D assigned numerical task made it possible to start the next phase of this investigation – creating the optimal turbine stage taking into account not only the losses in the nozzle and the blade cascades but also the leakages at the radial seals.

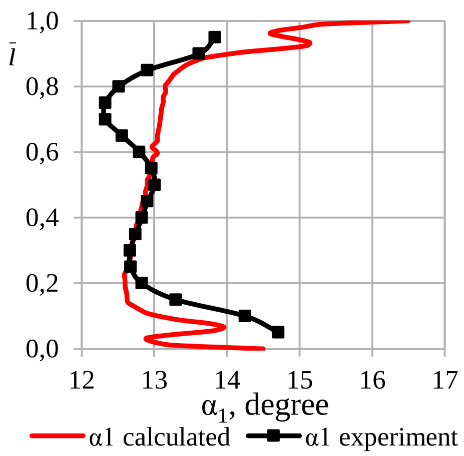

Figure 6: **Outlet flow angle distribution from the nozzles in absolute motion**

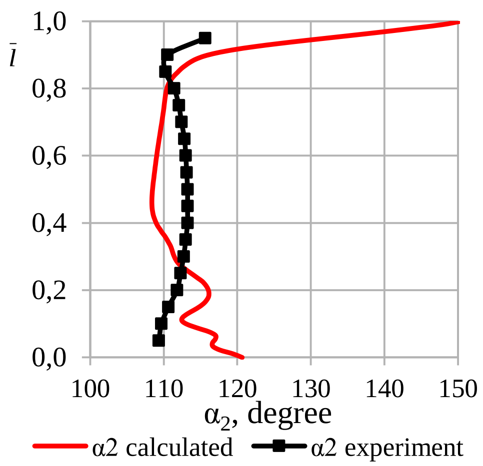

Figure 7: **Outlet flow angle distribution from the blades in absolute motion**

OPTIMIZATION TASK

The goal of the further investigation is to develop the turbine stage optimization method and to make qualitative and quantitative analyses of the optimization results.

As the initial one the turbine stage with the aforementioned geometric characteristics (see table 1), with an axial gap value of 4 mm and TC-1A profile without extension in a nozzle cascade was chosen.

For the optimization research a 3D model of the model turbine stage was built. The model was divided between the nozzle and blade

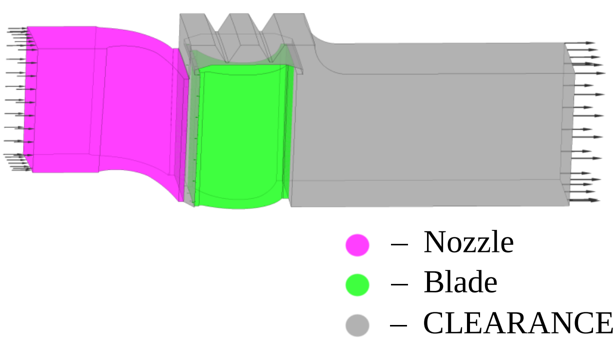

Figure 8: **3D model of the turbine stage**

domains. The axial gap, the radial gap and the exhaust pipe were formed as a separate domain – CLEARANCE (figure 8.)

The nozzle mesh has 180x98x90 elements while the blade mesh has 120x98x90 elements. The CLEARANCE domain consists of 2.2 million elements. On each solid surface the computed mesh has the thickening with the first element height of 0.01 mm (which is equal to $y^+ < 1$ on blades and $y^+ < 10$ on the other solid surfaces). The k-ω SST turbulence model are used for all simulations.

The variable parameters are the nozzle and blade twist laws.

At the stage of the optimal solution search, the developed optimization method suggests replacing the mathematical model which describes physical phenomena and processes in the axial turbine flow path to the approximation dependencies of the vector components of the Y function as the full quadratic second order polynomial (formal metamodel):

$$Y(q) = A_0 + \sum_{i=1}^{n} A_i q_i + \sum_{i=1}^{n} A_{ii} q_i^2 + \sum_{i=1}^{n-1} \sum_{j=i+1}^{n} A_{ij} q_i q_j, \quad (1)$$

The dependency (1) reflects only the formal relationship between the income and the outcome parameters. The metamodel building of the quality criteria and the functional limitations is performed by experiment planning theory methods (it uses three-staged Box-Behnken plans (Box and Behnken, 1960) and saturated Rechtshaffner plans (Rechtschaffner, 1967).) The formal metamodel (1) was specified by using cubic interpolation splines which allows to describe different functions of varying complexity with a high degree of accuracy (Boiko et al, 2013).

In the developed optimization algorithm for the optimal solution search a LPτ pseudo-random sequence of numbers is used (Sobol and Statnikov, 1981). The use of this LPτ is explained by its good adaptation for solving the turbine optimal projection tasks in modern formulations: it makes minimal demands for the response function smoothness; it is suitable for solving multicriteria tasks within the functional and parametric constraints; it allows to solve optimization tasks with many optimized parameters.

For the twist law variations it is proposed to use the following dependencies:

$$r_1^{m1} ctg\,\alpha_{st} = const,\ r_2^{m2} ctg\,\beta_{st} = const, \quad (2)$$

where m_1 and m_2 are the independent variables which characterize the twist law gradients for, respectively, the stator and the rotor. When $m > 0$ increasing angles from the hub to the shroud are obtained (straight twist), when $m < 0$ decreasing angles from the hub to the shroud are obtained (reverse twist) (Boiko and Garkusha, 1999).

When the profile stagger angles are known it is easy to determine the profile outlet flow angles.

After the calculations the optimal stage was obtained with parameters $m_1 = 0.53772$ and $m_2 = 2.1698$. The accuracy of the formal metamodel was 0.005 % (in this case the accuracy of the metamodel is defined as the difference between the metamodel prediction and verification by CFD.) The averaged stage working characteristics in the initial and the optimal stage are presented in table 4.

Table 4: **Averaged working characteristics of the initial and optimal stage**

Parameter	**Initial**	**Optimal**
Efficiency	85.05 %	85.34 %
Nozzle losses	3.30 %	3.23 %
Blade losses	7.46 %	7.51 %
Exit velocity losses	5.17 %	4.98 %
Mass flow	1.68021 kg/sec	1.69344 kg/sec
Degree of reaction	0.166299	0.169824

Thus, as the result of the optimization a new stage with an efficiency increase of 0.29 % compared to the original design was obtained. However, while calculating it was considered that in all numerical versions of the experiment plan (including the optimal one) the non-zero incidence angle takes place on the blades (see figure 9.) To avoid an incidence angle with increasing the quality of the flow around the blade is possible by the adjustment of the blade stagger angle. However, in this case modifications of the blade geometrical parameters will lead to changes in the stage mass flow and the stage outlet angle. Therefore, it was

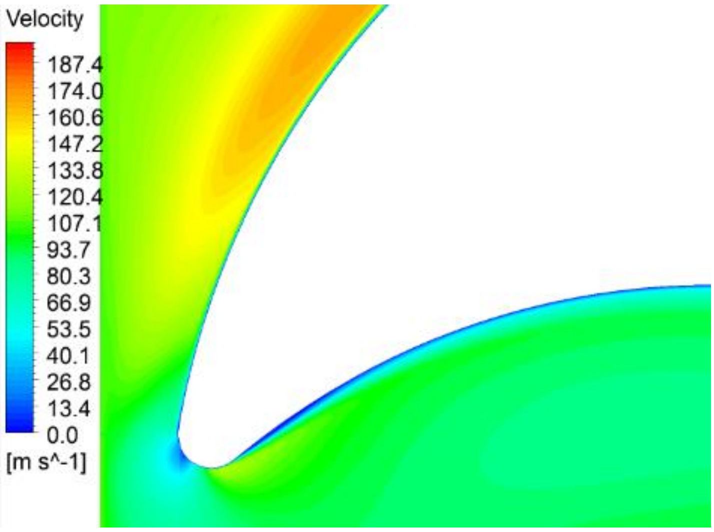

Figure 9: **Flowing at the leading edge of the blade after the first step of optimization**

decided to change only the blade inlet metal angle with following after that the profile conversion (see paragraph NUMERICAL DATA VERIFICATION.) Such an approach provides a zero incidence angle on the blades and does not cause significant changes in the inflow (mass flow) and outflow stage characteristics.

The stage with a zero incidence angle on the blades is not always the best to obtain the lowest losses. Therefore, in the optimization task the value of the blade inlet metal angle adjustment ($\Delta\beta_g$) was added as a variable. This allowed to obtain blade inlet flow angles which give the lowest blade profile losses.

The full list of the varied parameters which consists of the three components is presented hereafter: twist laws for the nozzle and blade (2) and blade inlet metal angle adjustment ($\Delta\beta_g$) which is uniformly applied to the entire blade. The ranges of variation were chosen based on previous investigations and can be seen in table 5.

Table 5: **Ranges of variation**

Parameter	Nozzle	Blade
m_1	-1 to 2	X
m_2	X	-1 to 4
$\Delta\beta_g$	X	0 to 12

The order of the numerical experiment being conducted is:

1. Experimental plan making;
2. Ranges of variation specification;
3. Calculation of the constants in the twist laws (2) on the mean diameter of the initial stage;
4. The nozzle and the blade stagger angles from the hub to the shroud are calculating by the dependencies obtained from the formulas (2):

$$\alpha_{st} = arctg\left(\frac{r_1^{m1}}{const}\right),\ \beta_{st} = arctg\left(\frac{r_2^{m2}}{const}\right); \qquad (3)$$

5. The blade inlet metal angle are changing to the value of the $\Delta\beta_g$;
6. The new profiles are created;
7. The turbine stage is calculated.

After the second step of the optimization the optimal turbine stage was obtained with the parameters m_1 = 0.94014, m_2 = 1.550125 and $\Delta\beta_g$ = 7.653259º.

Table 6: **Averaged working characteristics of the initial and the two optimal stages**

Parameter	Initial	Optimal #1	Optimal #2
Efficiency	85.05 %	85.34 %	85.53 %
Nozzle losses	3.30 %	3.23 %	3.38 %
Blade losses	7.46 %	7.51 %	6.92 %
Exit velocity losses	5.17 %	4.98 %	5.05 %
Mass flow	1.68021 kg/sec	1.69344 kg/sec	1.69214 kg/sec
Degree of reaction	0.166299	0.169824	0.160318

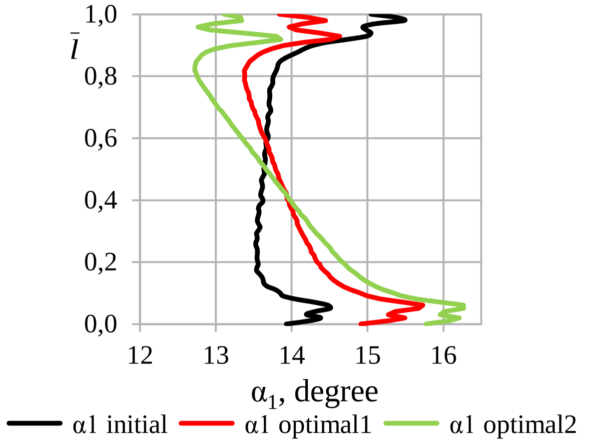

Figure 10: **Outlet flow angle distribution from the nozzles in absolute motion**

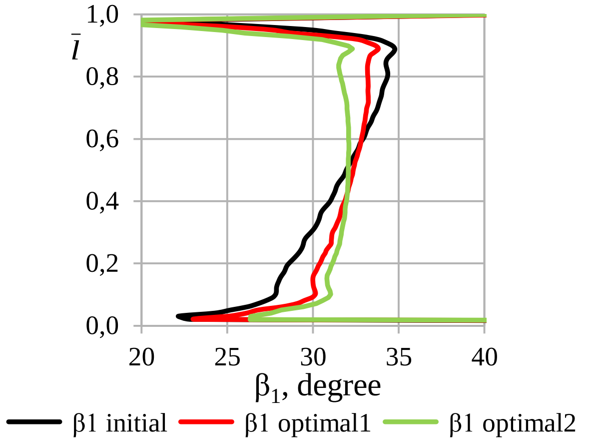

Figure 11: **Inlet flow angle distribution on the blades in relative motion**

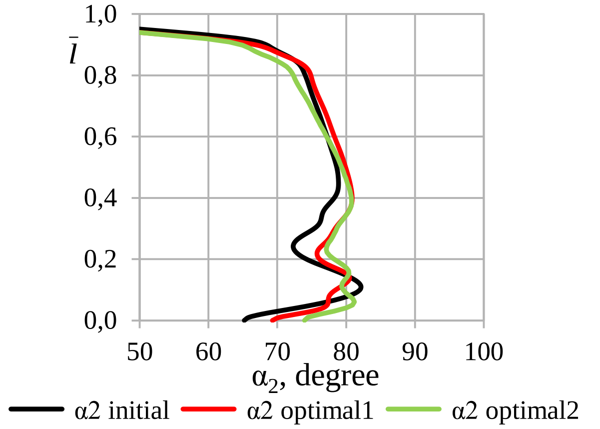

Figure 12: **Outlet flow angle distribution from the blades in absolute motion**

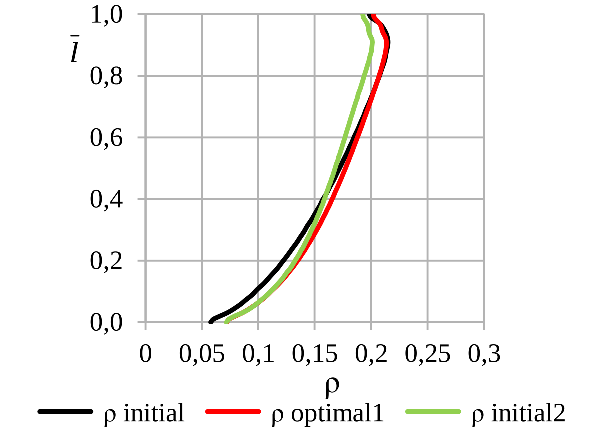

Figure 13: **Distribution of the degree of reaction in the stage**

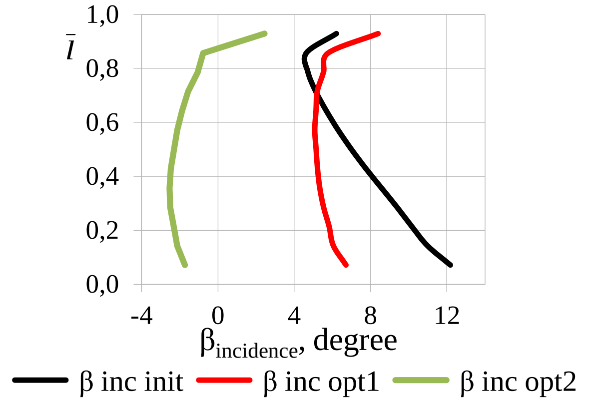

Figure 14: **Incidence angle distribution on the blades**

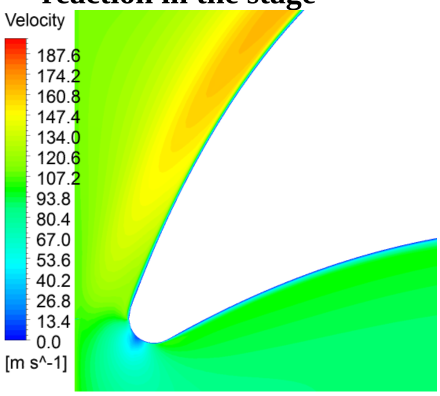

Figure 15: **Flowing to the leading edge of the blades after the second step of optimization**

The accuracy of the formal metamodel was 0.04 %. The averaged stage flow characteristics are presented in table 6.

One can see that the specified optimization method is better. The efficiency of the new turbine stage increases by 0.48 % compared to the original design. The blade profile losses are significantly decreased while the exit velocity losses are not so significantly decreased but are also less than in the original design.

For the more succinct analysis of the reasons leading to this stage performance boost the local stage characteristics from the hub to the shroud were obtained (see figures 10 – 15.) From figure 10 it can be seen that the nozzle outlet flow angle has considerably changed. Thus, such an alteration of the α_1 angle in combination with the changes of the β_{st} angle leads to a significant redistribution of the blade inlet flow angle. From figure 11 it can be seen that the β_1 angle became more uniform from the hub to the shroud of the blade cascade. A consequence of this is the normalization of the blade incidence angle to the entire blade cascade (see figure 14.) Moreover, by varying the β_{1g} value the optimal blade inlet flow angle was found. The obtained velocity contours around the blade confirmed this conclusion (see figure 9, figure 15.)

The degrees of reaction in the initial stage and in the optimal stage stayed virtually unchanged at the mean diameter (see figure 13.) However, in the optimal stage the gradient of the degree of reaction is aligned from the hub to the shroud. This leads to decreasing the leakages through the radial gap (- 0.17 %.)

It should also be noted that the conclusion about the variation of the blade inlet metal angle instead of changing its stagger angle has been confirmed. As can be seen from the plots, the stage outlet angle has not been changed significantly (figure 12.)

The important feature of this investigation is the concordance between the averaged and the local stage characteristics from the hub to the shroud. From table 6 it can be concluded that the efficiency of the optimal stage is increased due to the blade losses decreasing and it is also slightly increased due to the exit velocity loss reduction. The obtained local stage flow characteristics confirmed that the blade losses were decreased due to the reduction of the blade incidence angle and as a result the boundary layer became thinner which then promoted a more favorable flow around the blades.

CONCLUSIONS

1. The use CFD simulations for the qualitative and quantitative assessment of the incidence angle influence on blade losses while working on the turbine stage was demonstrated.

2. A turbine stage shape optimization method, which takes into account the nature of the flow around the turbine profiles and the leakages through the radial seals was developed.

3. The application of the developed method allowed to obtain a new turbine stage with an efficiency increase of 0.48 % compared to the original design.

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