Development of Recommendations on Building of the Lightweight Calculation Mathematical Models of the Axial Turbines of Gas Turbine Engines

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Abstract: The article contains the results of validation of the numerical computational models of the uncooled turbines of gas turbine plants created in the software suite NUMECA. The findings of the study of impact of the computational mesh dimension, account of the pre-path cavities on the design characteristics of turbines by modeling are represented in this article. As a result, there have been developed recommendations on building of the reliable computational models of the axial gas-turbine engines and the possibility of use in the optimization calculations of the turbine models without pre-path cavities without significant impact on its result has been substantiated.

Keywords: axial turbine, gas turbine power plant, identification, numerical methods of the gas dynamics, modeling, computational research.

1. INTRODUCTION

A turbine is the key unit of a gas-turbine engine determining the fuel efficiency of the engine in whole. For example, for TFE (TFE, bypass turbofan engine) with moderate cycle parameters the shortage of 1% efficiency of a HP turbine results in the increase in the specific fuel consumption by *0,7%* [1,2].For this reason and because of the continuously enhancing due to the growth of the expansion ratio, temperature and pressure conditions of operation the task of improvement of the turbine efficiency remains always relevant.

One of the most important tools that is used today by designers by the turbine design and development are the numerical methods of the computational gas dynamics. As of today this is the most exact method of computational study of the gas-dynamic processes in turbines since the basis for it are the equations describing the gas flow with minimal allowances [3]. The application of the numerical methods allows obtaining a large volume of information on the flow structure and parameters significantly exceeding the experimental data. At the same time the cost and term of obtainment of the computational results is lesser by times than that of the experimental ones.Forthisreasonthenumericalmethodsofgasdynamicshavebeenfirmlyfixedas the method of confirmatory analysis - «virtual experiment" that allows considering a number of options and selecting the best one [4].

Despite the high potential of the numeric methods of gasdynamics by solving the tasks of the turbo machine design and development many researchers point out that upon the good qualitative results the quantitative values of the parameters may often differ from the actual ones [5, 6]. In this regard those studies seem to be topical that are aimed at searching the recommendations on construction of the computational models allowing obtaining the most realistic results. In particular, it is needed to substantiate the sufficient number of elements of the computational mesh in the blade channels for numerical simulation of the process in turbines.

The turbine flow path is described by a few dozens of several variables. By searching its rational form a designer shall find such a combination thereof that will ensure obtainment of the best result. It is obvious that by sorting out the options manually it makes not much sense to expect success since even the processing and analysis of such a huge volume of information presents difficulties for an average man not to mention that such approach requires a huge amount of the computational time. This problem may be solved through the search for the rational combination of parameters by means of the optimization algorithms. By such approach the optimizing program will on its own, with the use of the parametric model [7, 8] automatically form some flow path, and then its basic parameters will be determined by means of numerical calculations. On the basis of a series of such calculations the optimizing program will by some means or other find the optimal value of the required parameter. Such approach allows reducing the time of searching the final variant due to automation of the data processing and model building as well as replacement of the random search by a certain algorithm [9, 10]. However, even by using this approach the searching time will run into days and months. In this regard in order to use it within the optimization process it is important to find a method of construction of such computational models that will allow obtaining the solid and reliable results upon the moderate computational cost.

Thus, within this study we intend to solve the two tasks:

1. to substantiate the selection of the sufficient number of elements of the computational mesh in the blade channels for the numerical simulation of the process in the axial uncooled turbines.
2. to provide recommendations on the founded simplification of the computational models of the axial turbines for the use thereof in the optimization process.

2. COMPUTATIONAL MODELS BUILDING

The mentioned study was performed using as an example the axial uncooled low pressure turbines (LP) and the free turbine (FT) of a three-shaft gas-turbine engine for the gas-compressor unit drive (Fig.(**1**)). The numerical models of the process in the specified turbines were generated with the use of the software suite *Numeca AutoGrid 5*[11].

The four groups of the computational models have been built in total. They distinguished by the number of elements of the finite element meshes or absence or presence of the pre-path cavities. Thus, the models of the first group (*LPT_model 1* and *PT_model 1*) contained at average *450 000*elements per one blade row (BR). The models of the second group (*LPT_model 2* and *PT_model 2*) - *1000 000*elements per one BR. The models of the third group (*LPT_model 3* and *PT_model 3*) - *2000 000* elements per one BR. In all the models the structured grid was used. The construction of grids was also performed in the software suite *Numeca AutoGrid5* [11].

All the above-mentioned models did not account for the presence of the pre-path cavities close to the hub and in the banding area. (Fig. (**2**).) The models of the fourth group (*LPT_model 4*and*PT_model 4*) are the models of the second group that differ through account of the pre-path cavities.

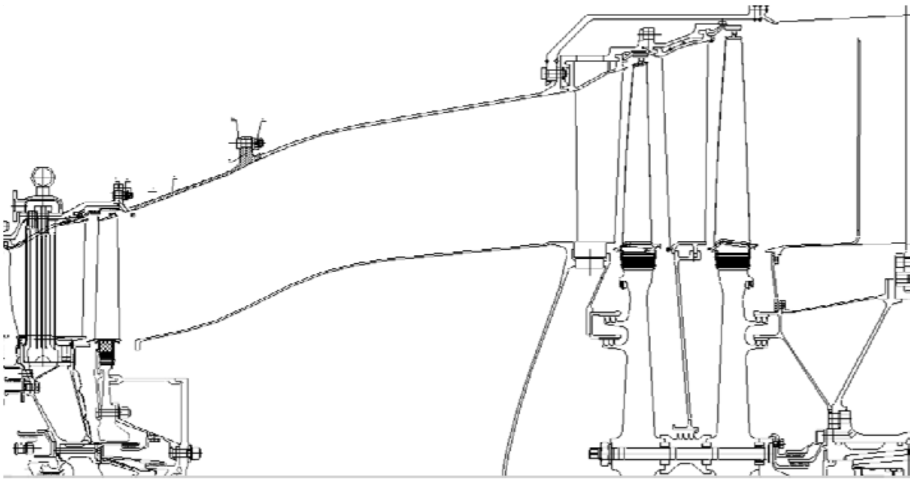

Fig. (1).Turbine under consideration.

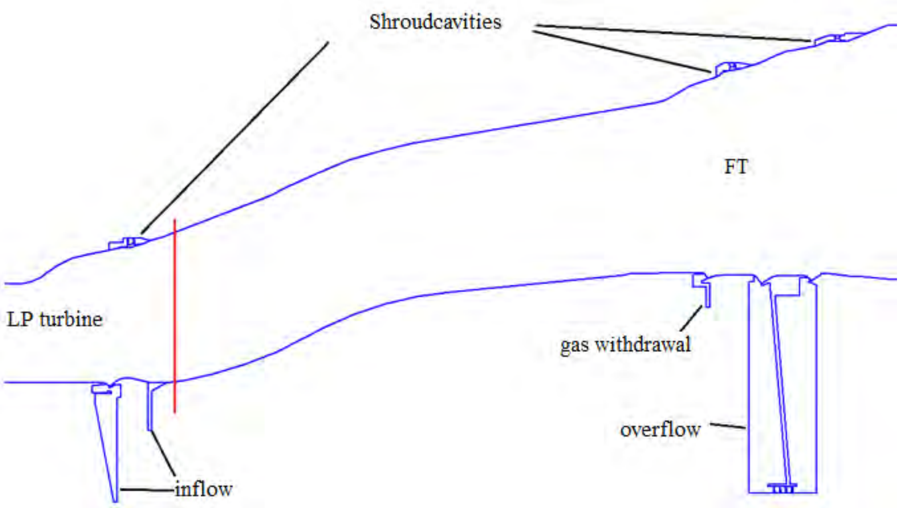

Fig. (2).Design of the flow section of the investigated turbine with indication of the pre-path cavities taken into account

The following assumptions were used by construction of the numeric models of the process in turbines.

The flow in each BR of the nozzle diaphragm (ND) and the blade wheel (BW) as well as in the pre-path cavities features cyclic symmetry characteristics. I. e., flows in all the blade channels within one BR are the same. That's why all the models contained only one blade channel with periodic boundary conditions at the lateral surfaces.

1. The calculation was performed on the basis of a stationary setup.

2. The strains of the turbine blades under the gas load were not taken into account by calculation. The strains of the blades and disks under the centrifugal and thermal load were taken into account by describing the form of the flow path meridian section. The deformation values were provided by the manufacturer of the investigated turbine.

3. The created model made allowances for assembly deformations of the rotor blades (swing of the peripheral sections) occurring by assembly of the blade rows combined with a platform.

4. The ideal gas with the properties of the kerosene combustion products was used as a fluid. The gas constant made: $R = 287{,}335\, J/kg \cdot К$.

5. During the calculation it was taken into account that the fluid properties depend on its temperature. In particular, the heat capacity at constant pressure was calculated with the use of the following polynomial:

$$c_p = 829{,}2 + 0{,}5068 \cdot T - 0.00019254 \cdot T^2 + 0{,}000000027364 \cdot T^3, \frac{J}{kgК},$$

where,

T – fluid temperature.

The fluid viscosity was described by the Sutherland's equation:

$$\mu = 1{,}49 \cdot 10^{-5} \frac{273 + 200}{200 + T} \left(\frac{T}{273}\right)^{1.5}, \frac{kg}{m \cdot s}.$$

6. The turbulence was taken as isotropic in all directions. The *Spalart-Allmaras* model was used for simulation thereof.

7. The heat exchange between the walls of the flow path and the flow was not taken into account because of the fast progress of the process under consideration.

8. The models of the first, second and third groups did not account for the presence of radial clearance between the blade tip and the stator.

In the Fig. (**3**) the appearance of the LP turbine computational domain is presented (*LPT_model*) and in the Fig. (**4**) – the geometrical model of the FT computational domain with the pre-path cavities (*PT_model 4*).

Within the computational domain there have been distinguished the spatial domains around the rotating and nozzle blades. The ND area was calculated within the fixed system of reference. TheRBareawascalculatedwithintherotatingcoordinatesystemtherotation rate of which matches the speed of the corresponding rotors (5005 rpm for LP turbines and 5000 rpm for FT).

The interface *FullNonMatchingMixingPlane* integrated in the software suite was used for the data transfer between the ND and BR areas. Itaveragestheflowparametersinthecircumferentialdirectionintheupstreamareaand transfers as the boundary condition to the downstream area [11].

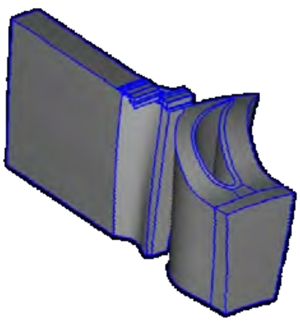

Fig. (3).Geometrical model of the LP turbine computational domain.

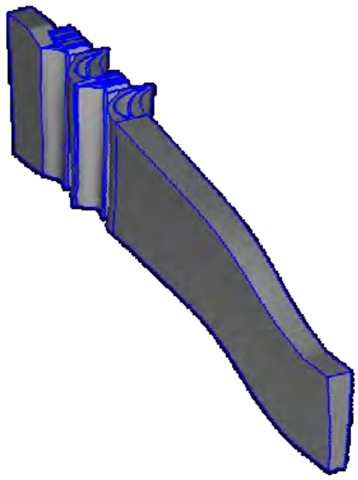

Fig. (4).Geometrical model of the FT computational domain with pre-path cavities.

As the boundary conditions at the turbine in let the radial diagrams of distribution of the total pressure $p*$, total temperature $T*$, flow angle α and eddy viscosity were applied to the relevant section of the flow path of the gas-turbine plant. For a LP turbine such diagrams were obtained as the result of numeric calculation of an intermediate pressure turbine (IP turbine) (diagrams of parameters at the output of the IP turbine) [13], for a FT – as the result of calculations for a LP turbine.

At the output of the turbines the static pressure was set for the hub radius. The pressure at the other radii was calculated by the program automatically according to the radial force balance equation.

3. ANALYSIS OF THE TURBINE PERFORMANCE

For each turbine the values of the flow, sector, streamline and integral parameters were calculated at all points of the computational domain (Fig. (**5**) and (**6**)).

The relevant characteristics were calculated for all groups of models. The three dependences were obtained per a model (Fig. (**7**)):

- Turbine throughput capacity $A = \frac{G\sqrt{T^{*}}R}{p^{*}}$ of the expansion ratio π_T^{*}. For calculation of the throughput capacity where G – mass fluid flow, p^{*} and T^{*} - total pressure and temperature of the fluid at the turbine input, R - gas constant for the combustion products;

- Turbine efficiency η^{*} depending on the expansion ratio π_T^{*};

- angleoftheflow exit from the turbine α depending on the expansion ratio π_T^{*}.

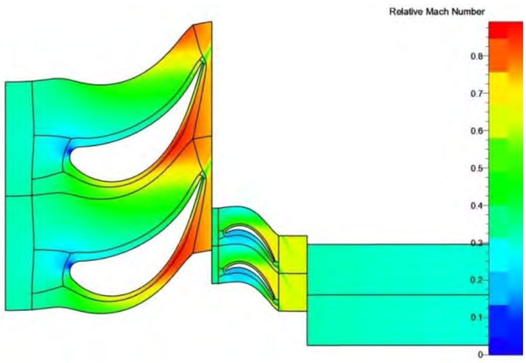

Fig. (5).Mach number field at the LP turbine mean diameter.

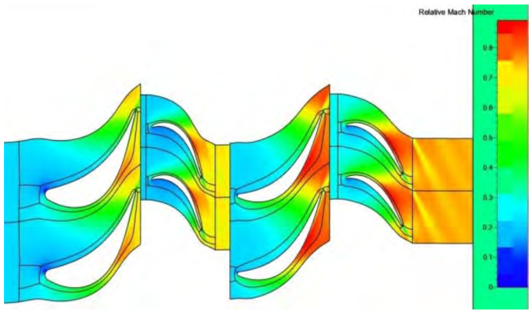

Fig. (6).Mach number field at the FT mean diameter.

The dependences derived for different models were compared with each other as well as with the similar characteristics calculated for these turbines with the use of a one-dimensional computational model generally accepted in the Russian Federation and approved as the Technical Guides[12].

By analyzing the dependences indicated in the Fig. (**7**)one may draw the following conclusions:

1. The models of all the three levels distinguishing by the grid density are in good agreement with the accepted in the Russian Federation and approved as the Technical Guides procedure of the characteristics calculation [12].

2. The results of the numerical simulation differ from the results of calculations according to the one-dimensional computational model [12] by possibility to take into account the impact of the complete spatial structure of the blade on its gas-dynamic behavior.

3. Despite the good qualitative coincidence of the results of calculations with the use of different models the significant quantitative non-coincidence takes place.

4. The most significant differences in the quantitative estimates made with the use of different models are observed by forecasting the efficiency value. The difference between the values calculated with the use of different models may exceed *1%*in absolute magnitude. The results derived from the model 2 and 3 (the medium and dense grid) nearly match each other while the results according to the coarse grid (model 1) significantly differ from them in magnitude.

5. To perform the qualitative study of the process in an axial turbine it suffices the finite element meshes containing about *1*million cells since further mesh refinement hardly influences the final result.

6. The fact that the results of calculation with the use of the finite element meshes of different density coincide means that if few same variations of the turbine modernization will be calculated with the use of different models with variable mesh density there is a high probability that from the perspective of the gas-dynamic efficiency the same alternate design will be selected as the best one. In this regard the approach to the search for the optimal flow path form is seen as follows. By using the computational model with a coarse grid it is possible to consider within a relatively short period of time a great number of design variants and choose the one providing for the maximum improvement of efficiency. Then with the use of a model with a qualitative grid to specify the work-process related parameters of this variant and its certain characteristics.

As noted above, searching for the rational form of the turbine flow path may be performed not only by sorting out the variants manually but also with the use of the optimizing programs like *IOSO* [9, 10]. The use during the optimization process of the coarse-grid model will allow finding the optimal variant within a shorter period of time. Further on the position of optimum may be specified during the second optimization cycle with the use of the models with qualitative grids. However, in this case the range of the variables variation may be reduced by times based on the results of the initial optimization search which also significantly saves time for achievement of the final result.

4. ASSESSMENT OF THE IMPACT OF PRE-PATH CAVITIES

In order to assess the impact of the pre-path cavities on the turbine design characteristics there has been created the fourth group of the computational models built on the basis of the second group models and distinguishing from it by the presence of pre-path cavities (Fig.(**4**)). Compared to the models of the other groups the model with pre-path cavities accounts for the presence of radial clearance above the platform tips.

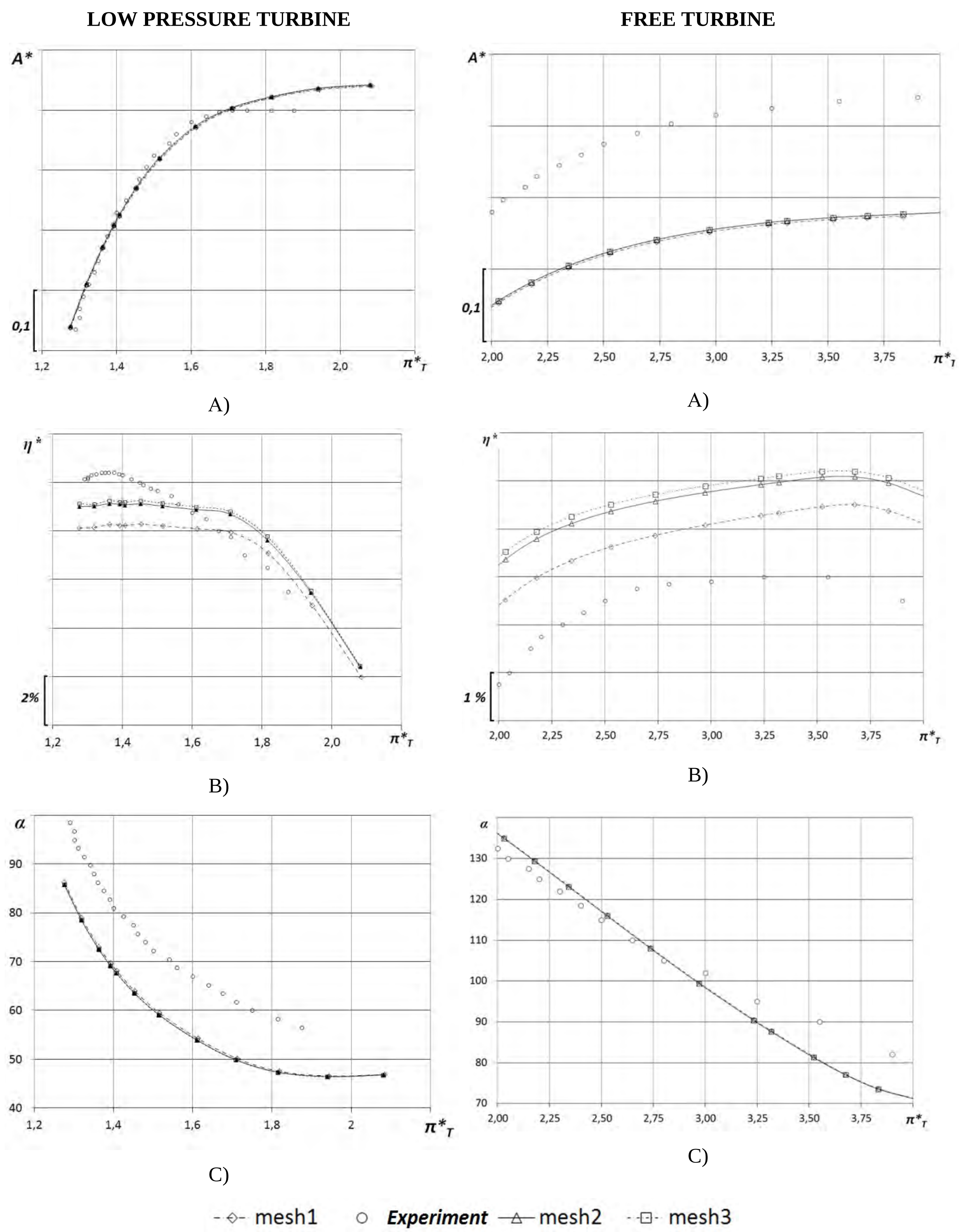

Fig. (7). Comparison of the dependences of through put capacity(A), efficiency(B) and angle of the flow exit (C) on the expansion ratio derived with the use of the different computational models.

With the use of these models the turbine characteristics were derived in the form of dependences (Fig. (**8**)):

- the turbine throughput capacity Adepending on the expansion ratio π_T^* ;

- the turbine efficiencyη^*depending on the expansion ratio π_T^* .

The characteristics derived were compared with the results of calculation with the use of the second group models and findings of calculation with the use of the one-dimensional computational model [12].

LOW PRESSURE TURBINE **FREE TURBINE**

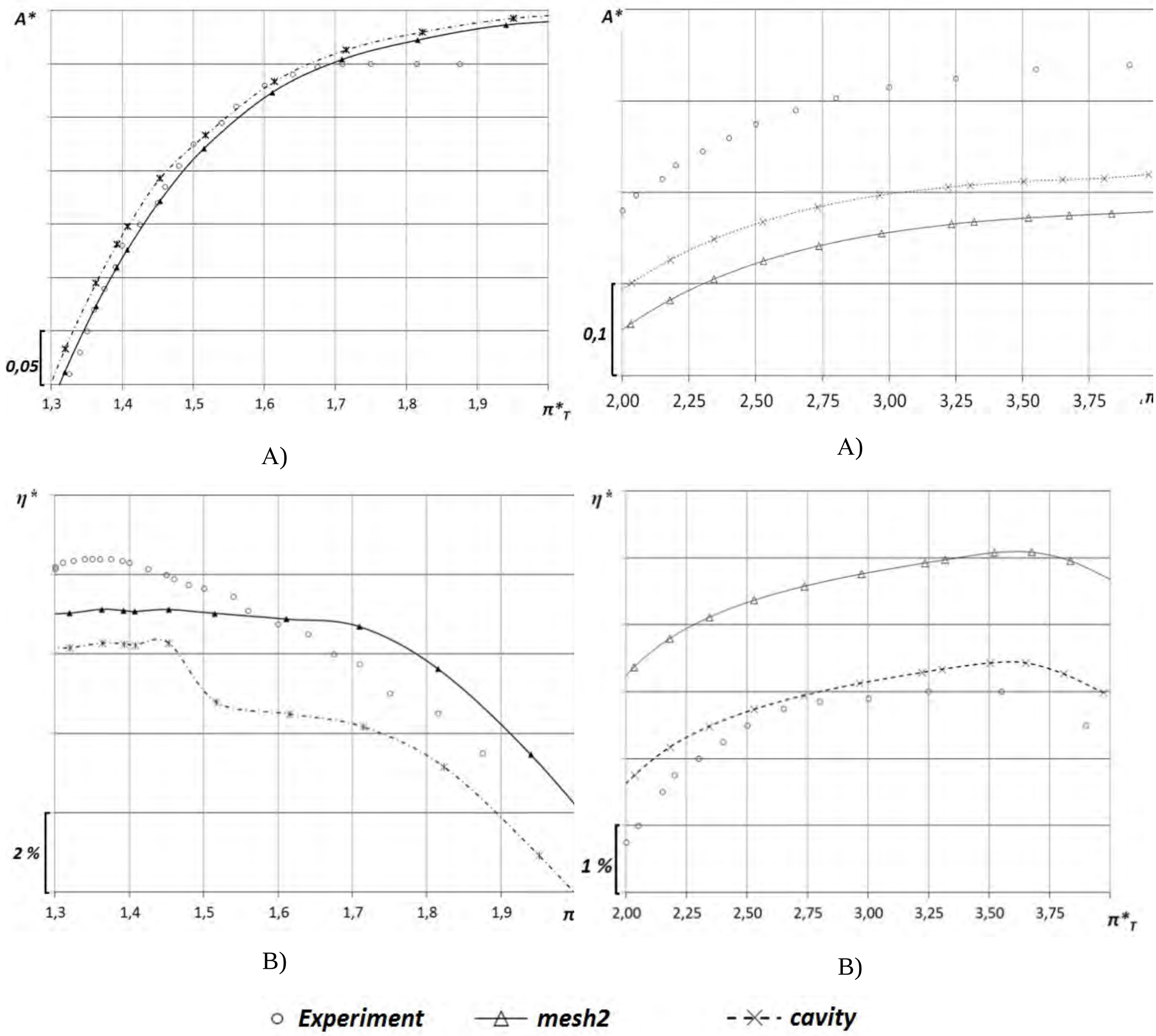

Fig. (8).Comparison of the dependences through put capacity(A) and efficiency (B) on the expansion ratio with and without account for the pre-path cavities.

By comparing the results of calculation it can be seen that account of the pre-path cavities does not significantly influence the turbine characteristics behavior. The account thereof results in the reduction of the estimated efficiency value by *1%* (abs.), and increase of the throughput capacity by up to *5%* (relative). This fact seems to be related to the account of the gas overflow through the radial clearance which increases the losses due to appearance of the eddy (vortex) structure in the shroud cavities (Fig. (**9**)) as well as actual increase in the clear area since the clearance area is added to that of the blade channels.

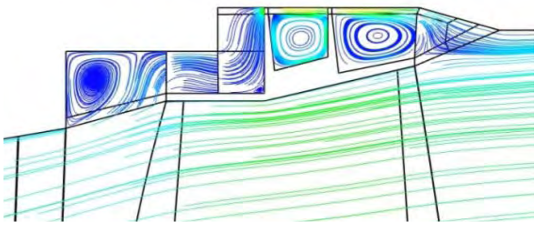

Fig. (9).The structure of the flow in the cavity above the platform.

The fact that account of the pre-path cavities does not have a qualitative impact on the characteristics behavior means that there is no need to take into account the presence of such cavities by searching the optimal flow path form (except for the cases of the cavity form adjustment). It makes sense to use the simulation model with pre-path cavities for calculation of the certain characteristics of the final turbine design.

6. CONCLUSIONS

During the study it has been shown that despite the quantitaive difference between the results derived with the use of the different simulation models with variable grid density and account of the pre-path cavities all of them provide the same results in qualitative terms. In this regard in order to reduce the computation time and required computer resources by searching the rational form of the turbine flow path, especially with the use of the optimization algorithms it makes sense to use the models without account of the pre-path cavities and with a coarse grid. The models with a qualitative grid and enhanced detalizations hould be used for calculation of the certain characteristics of the final variant and clarifying optimization in the area close to the extremum derived.

The study is performed with financial support from the Ministry of Education and Science of Russia within the frameworks of the basic part of the government task.

CONFLICT OF INTEREST

The author confirms that this article content has no conflict of interest.

ACKNOWLEDGEMENTS

This work was financially supported by the Russian Ministry of Education as part of the basic tasks of the state.

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