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DEVELOPMENT EXPERIENCE OF THE CENTRIPETAL TURBINE FLOW PART FOR AN AVIATION ENGINE AIR STARTER

*The **subject of this** paper is to study the approaches for the flow part design of a centrifugal turbine stage, which are considered on the example of an air starter with a capacity of 150 kW. The **goal** is to develop the spatial shape of the centripetal turbine flow part of the air starter of an aircraft engine. The **tasks** were as follows: determine the full 3D shape of the centripetal turbine of the air starter of an aircraft engine, sufficient for the development of design documentation, and calculate the gas-dynamic characteristics of the turbine, including the self-braking frequency. The research was carried out using modern numerical calculation **methods** and the design of radial-axial centrifugal turbomachine flow parts, implemented as a software complex IPMFlow. The following **results** were obtained. Two versions of the flow part have been developed: with profiled and thin rotor blades. The first version with profiled blades has better gas-dynamic characteristics, as well as reserves for further improvement, but it does not meet the requirements for strength. To eliminate the mentioned shortcomings of the first version, the method of blade profiling of radial-axial flow parts was improved, and with its help, a second version of the flow part with a rotor with thin blades, which has better characteristics in terms of strength and acceptable weight, was developed. This version was adopted as the basis for further design and production. **Conclusions.** The scientific novelty of the obtained results lies in the improvement of the method of radial-axial flow-part blade profiling. The proposed method provides such a shape of the blades that in a plane perpendicular to the rotor rotation axis, the blade centerline coincides with the radial line. In this case, the possibilities of so-called "spatial profiling" are reduced, in particular, the compound offset (saber-like) of the blades. This approach makes it possible to significantly reduce the load and stress under the action of centripetal forces. With the help of improved methods, a version of the flow part with a rotor and thin blades was developed. Although the efficiency of the turbine was slightly lower than that of the first version, it is quite high, and the power is more than the design value.*

***Keywords:** gas-turbine engine; flow part; meridional contours; spatial flow; gas-dynamic characteristics; numerical analysis.*

Introduction

Different types of turbines are widely used in technical devices, such like air compressors [1], gas turbine engines [2], high-pressure centrifugal turbines [3], ORC turbines [4], steam turbines [5], and others. The process of turbine development is a complex, multifactorial task. For the turbine development researches and manufacturers use a lot of different approaches, like inverse design and optimization techniques [6], the inverse design [7], integrated optimization algorithm [8], multi-objective optimization algorithm [9, 10], or multidisciplinary optimization algorithm [11, 12], multiple surrogate model algorithm [13]. Solution of this task takes into account many conditions and requirements [14], such as efficiency, reliability, resource, compliance with the technical task in terms of gas-dynamic characteristics [15] (mass flow, thermal drop, dimensions, mass, cost, etc.).

Turbines (turbine stages) are classified according to many parameters, but the most common classification is according to the flow direction of the working media: axial [15], centripetal [2] and centrifugal [3], in addition, they can be mixed. Each type of turbine has its own field of application and corresponding advantages and disadvantages.

The most common are axial-type turbines [15, 16], they are used in various types of energy and technological machines. There is the greatest experience in working with this type of turbines and with their help it is possible to ensure a high technical level of the product, for example, gas turbines [16, 17], or ORC turbines [18, 19].

Centripetal turbines are the least common due to their low efficiency and quite exotic. This is explained primarily by their low efficiency. As a rule, they are used where the relevant design features require it [20, 21].

Centripetal turbines can have stages of radial or radial-axial types [9, 22]. Radial stages, as a rule, have a simple form, but at the same time they are less efficient. They will not be considered in this paper.

Radial-axial stages of centripetal turbines [2, 9], in comparison with other types, have the highest gas-dynamic efficiency under the same conditions. At the same time, with the same maximum diameters of the rotors and rotation speeds, they can ensure the operation of larger thermal drops [9, 23]. In such operation conditions of rotors, there are numerous problems associated with the appearance of a dense interaction between the flow and blades [24], and with the selection of materials for structural elements [24, 25]. Turbines of this type, as a rule, are used when the required characteristics can be provided with the help of only one stage. Another limiting factor for the use of such turbines is the complex spatial shape, which complicates their manufacture, but with the technological progress, this problem has significantly decreased.

The authors have extensive experience in the development of radial-axial turbines stages for turboexpander units used in technological processes of natural gas drying, as well as axial and radial-axial turbines stages of other types [23], like ORC turbines [26], high-pressure steam turbines [27, 28], low-pressure steam turbines [29], hydrogen turbines [30] and hydro-turbines [31]. The results of the generalization of this experience for the design of a new object – a centripetal turbine of an aircraft engine air starter – are given in the paper. Three-dimensional calculations and design were performed with the help of the IPMFlow software complex [23] developed at the IPMach of the NAS of Ukraine, in which, specifically to ensure the increased strength requirements of the air starter rotor, appropriate refinements were made.

The purpose and objectives of the study should be interrelated and should cover the theme stated in the paper title. Research objectives should be numbered.

1. Method of gas-dynamic calculation of three-dimensional viscous turbulent flow in the turbomachines flow parts

At the current stage of the development of hydrogas dynamics, many researchers believe that the Navier-Stokes equations fully describe the movement of a continuous medium in which there are no physical and chemical transformations [32]. However, their application to real problems is complicated due to the limited capabilities of computer technology.

Today, the main direction of development of computational fluid dynamics is the use of Reynolds-averaged Navier-Stokes (RANS) equations [33, 34]. The most common methods of closing the (RANS) equations are based on the Boussinesq hypothesis [35] is used.

In the paper, the numerical study of the three-dimensional flow and the design of the turbine flow part was performed using the IPMFlow software complex, which is a development of the FlowER and FlowER-U [36]. The complex's mathematical model is based on the numerical integration of the Reynolds-averaged unsteady Navier-Stokes equations using the implicit quasi-monotonic ENO scheme of increased accuracy and SST Menter's two-parameter differential turbulence model [37]. To take into account the thermodynamic properties of working medias, various equations of state are used. When steam is used as a working media, the interpolation-analytical method of approximation of the equations of the IAPWS-95 formulation [38, 39] is used to describe its thermodynamic properties. Calculation results obtained with the help of the IPMFlow software complex have the necessary reliability both in terms of physical conceptions of the flow structure and quantitative values of integral gas-dynamic quantities [23, 40].

To speed up the calculation time, an original, highly efficient parallel computing technology is implemented in the IPMFlow software complex. For example, the parallelization of the computational process of the flow part, consisting of 18 stages for 9 processes when using a computer with 8 cores (threads), gave an astronomical acceleration of the calculation time by 7.1 times with the maximum theoretically possible acceleration of 8.

2. Initial data. General principles of gas-dynamic design of centripetal turbine flow parts stages

According to the technical task, the flow part of the turbine should consist of one stage with a radial stator and a radial-axial rotor (Fig. 1).

The initial data for the turbine designing are:

- Working media – air;
- Mass flow – 1.24 kg/s;
- The turbine power is not less than 150 kW;
- Total pressure at the inlet – 353.04 kPa;
- Total temperature at the inlet – 481 K;
- Static pressure at the outlet – 101.32 kPa;
- Estimated rotation speed of the turbine rotor – 40000 rpm.

Geometric dimensions and overall limitations (see Fig. 1):

- $D0 \leq 200$ mm;

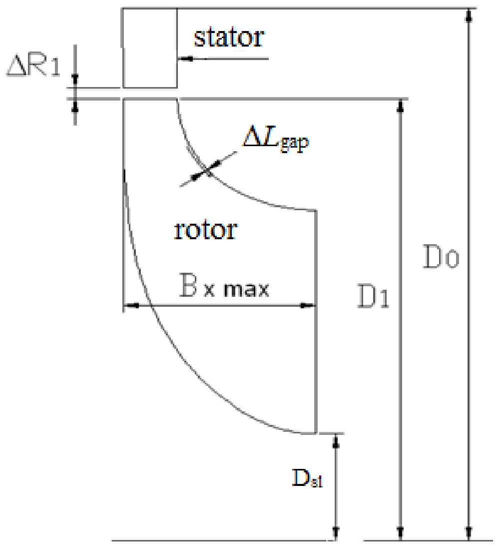


Fig. 1. Scheme of the flow part meridional section

– D1 ≤ 166 mm;
– Dsl ≥ 40 mm;
– Bx мах ≤ 35 mm;
– ΔR1 ≥3 mm;
– ΔLgap = 0,3 mm.

It is necessary to determine the full 3D shape of the flow part, sufficient for the development of design documentation, as well as to calculate the gas-dynamic characteristics of the turbine, including the frequency of self-braking. After checking, if necessary, it is needed to make changes to the shape of the flow part to meet the strength requirements.

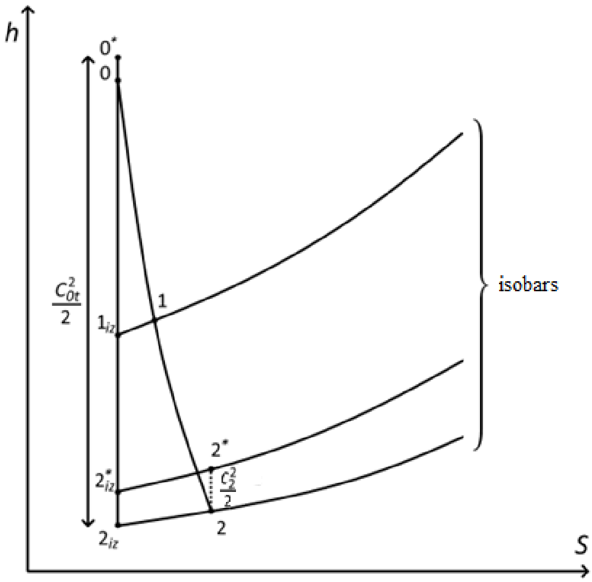


Fig. 2. h-s diagram of the thermodynamic process in the turbine stage

One of the main tasks in the design of the flow part is to achieve minimum values of kinetic energy losses and maximum efficiency. The thermodynamic process in the turbine stage occurs according to the h-s diagram shown in Fig. 2. The efficiency of the process is defined as:

$$\eta = \frac{h_0^* - h_2^*}{h_0^* - h_{2iz}} = \frac{h_0^* - h_2 - {c_2^2}/{2}}{{c_{0t}^2}/{2}}. \qquad (1)$$

Since the kinetic energy that the flow has at the flow part outlet is not further used to obtain work, this energy in equation (1) is considered as a loss with the outlet velocity ($c_2^2/2$). Equation (1) can be written in the following form:

$$\eta = \frac{c_1^2 - c_2^2 - w_1^2 + w_2^2 + u_1^2 - u_2^2}{c_{0t}^2}. \qquad (2)$$

From the analysis of equations (1) and (2), it is possible to give some explanations to the well-known design principles of centripetal turbine stages, as well as why their efficiency is, as a rule, higher compared to axial ones. The thermodynamic process in the turbine stage takes place between two isobars marked by lines 0* and 2 in Fig. 1. An ideal process is isentropic, and a real one takes place with an increase in entropy. In the case when the real process coincides with the ideal one, according to (1) and (2), the efficiency equal to 1 will not be achieved. This is due to the fact that the value of the speed c_2 cannot be equal to 0, since in this case there will be no flow of the working media and the turbine stage will not perform its functions. At the same time, in order to increase the efficiency, it is necessary to achieve the maximum possible reduction in speed c_2. Also, the increase in efficiency will occur when the real process approaches the ideal one (isentropic). It is interesting to note that the value of u_1^2–u_2^2 is not affected by how close the thermodynamic process is to the real one, but only by the rotor rotation speed and the geometric characteristics of the rotor. For axial flow parts, in which the average diameters of the rotor at the inlet and outlet edges are equal or barely different, the value of u_1^2–u_2^2 is equal to or close to 0. For centripetal turbines, the value of u_1^2–u_2^2 can be quite significant and be equal to 0.3 or more of the total value of the numerator in equation (2). This fact may explain to some extent why centripetal turbines, under the same conditions, usually have a higher efficiency compared to axial ones. This happens because a significant part of the flow energy is transformed into mechanical energy due to a change in the tangential rotor rotation velocity, which is not affected by the loss of kinetic energy due to the increase in entropy. Then, to increase the efficiency of the centripetal turbine, it is necessary to increase the value of u_1^2–u_2^2.

Thus, it can be argued that in order to achieve the maximum efficiency of the centripetal stage of the turbine, it is necessary to try to achieve:

a) the minimum value of c_2;

b) the maximum approximation of a real thermodynamic process to an ideal (isentropic);

c) the maximum value of $u_1^2 – u_2^2$.

To fulfill condition (a), it is necessary to approach the angle of the flow in absolute motion at the rotor outlet to the axial direction, increase the area of the outlet cross-section, and bring the thermodynamic process closer to isentropic (condition (b)).

Fulfillment of condition (b) is achieved by using many techniques, among which are: selection of rational general mode and geometric characteristics of the stages, creation of smooth surfaces of the flow part (without breaks of the second derivative), confuser channels, shock-free inflow of the flow onto the blades, prevention of the formation of large vortices and flow separations, etc.

Condition (c) is ensured primarily by an increase in the difference between the average diameters of the rotor leading and trailing edges.

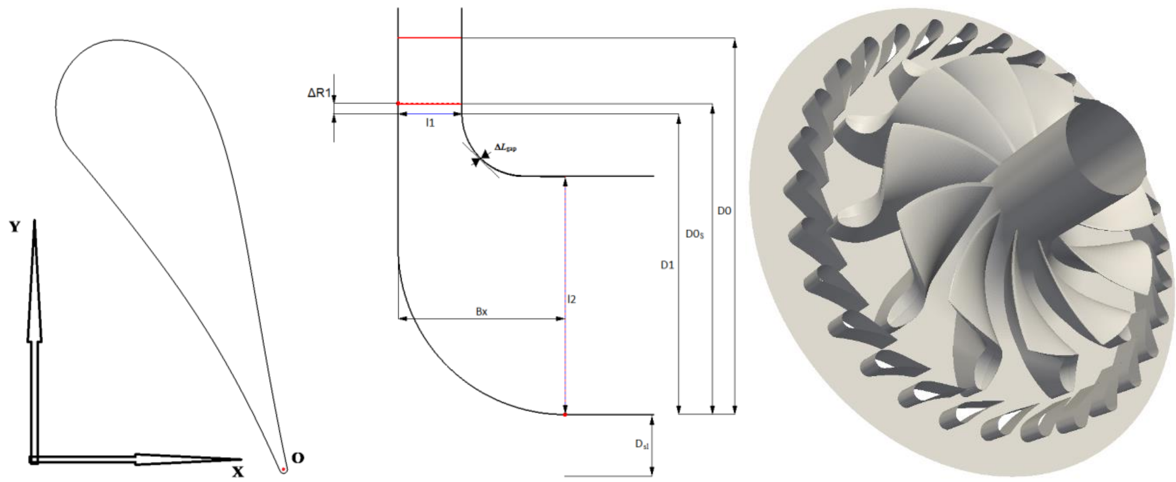
The measures necessary to fulfill the above conditions may contradict each other. For example, to increase the outlet cross-sectional area (condition (a)) it is necessary to either increase the average diameter or the height of the channel at the outlet. These measures lead to a decrease in $u_1^2 – u_2^2$ (a contradiction to condition (c)), and/or to an increase in the unevenness of the flow along the height and, as a result, to an increase in entropy (a contradiction to condition (b)). Thus, the fulfillment of conditions (a)-(c) is the solution of a complex (optimization) problem.

Optimizing the flow part of even one stage in a classical setting based on the variation of a large number of parameters and taking into account all restrictions is a rather difficult task. As a rule, various techniques based on previous experience, which make it possible to significantly reduce the range of searching for solutions, are used. In this case, it is rather possible to talk not about optimization, but about finding a rational solution [6]. Thus, on the basis of various studies, it is known that for turbine stages, including centripetal ones, the highest efficiency values are achieved at $u_1/c_{0t} \approx 0.7$ and reactivity degree of ≈ 0.5. For the given initial data, with the maximum possible diameter D1=166 mm, the condition $u_1/c_{0t} \approx 0.7$ is fulfilled, so this value of the average diameter at the rotor inlet was adopted for further development of the flow part.

3. Centripetal stage of the turbine with profiled rotor blades

A version of the flow part of the air centripetal turbine with profiled rotor blades was developed first using the IPMFlow software complex according to the original methodology that has proven itself well in the design of various types of turbines, like ORC [26], steam [27, 36] with complex blade shape [40]. A modified method of axial turbine blades profiling [27, 28] was used to create the stator blades. Calculations were performed on a difference structured H-type mesh with a total number of cells close to 1 million. Mesh thickening near solid surfaces is made in such a way as to ensure y+≤5, which is sufficient for engineering calculations [26]. The description of the thermodynamic properties of the working media is made using the equation of state of a perfect gas, which is well suited for the considered operating conditions of the turbine.

The view of the flow part of the turbine is shown in Fig. 3.



a) b) c)

Fig. 3. The flow part of the turbine with profiled blades:
a) – stator blade profile; b) – meridional cross-section; c) – isometry

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The process of the flow part developing took place in two stages. At the first stage, only rotor was developed, and the presence of stator was simulated by setting the angle of the flow on the rotor blades – α_1. At the second stage, after determining the shape of the rotor, the flow part of the stator was developed in such a way as to ensure the previously determined angle.

There was no need to consider many versions to create the flow part shown in Fig. 3. Most of the geometric characteristics were set immediately based on experience. Among the main characteristics that had to be varied are the heights of channels l1 and l2 (see Fig. 3), as well as the angle of the flow on the rotor blades – α1 and the geometric angle of the blades at the rotor outlet – β_{g2}. In order to ensure the efficient operation of turbines in a wide range, the rotor is developed with profiled (body) blades, which are less sensitive to off-designed flow angles. To reduce losses at outlet velocity, the rotor has a relatively large outlet area, i.e. a relatively high blade height at the outlet. A large relative height of the blade usually creates a flow separation at the periphery, which has a bad effect on the turbine efficiency. To reduce this negative impact, the rotor is made with saber-shaped blades at the outlet. The main geometric characteristics of the flow part are:

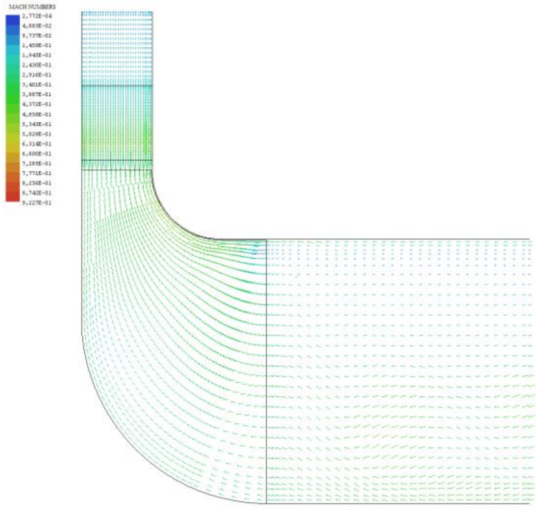
- D0 = 198 mm;
- D0s = 174 mm;
- D1 = 166 mm;
- Dsl = 40 mm;
- Bx мах = 35 mm.

Figs. 4-5 show the visualization of the flow in the flow part, and Fig. 6 shows the distribution of static pressure on the surfaces of the stator and rotor blades.

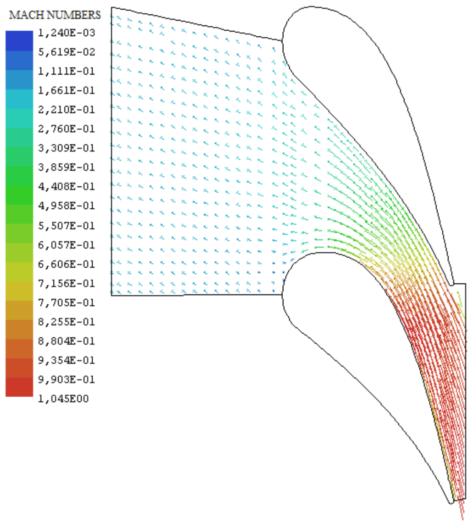
It can be seen that the obtained flow pattern is favorable. There are no flow separations, the pressure distributions are quite monotonous. The integral gas dynamic characteristics are:

- kinetic energy losses – 6.1%;
- kinetic energy losses with outlet velocity – 3.8%;
- stage efficiency taking into account losses with outlet velocity – 89.9%;
- power – 162.0 kW.

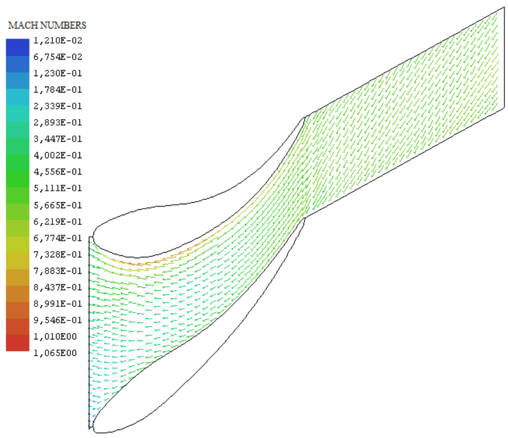
Despite the fact that the obtained indicators are quite high, and the power significantly exceeds the design one, the flow part has significant reserves for further improvement. The inspection of the rotor of the centripetal turbine showed that it does not meet the requirements of strength and has a large weight. Excess weight is caused by the use of profiled (body) blades. The main stresses in the rotor are associated with the bending deformations of the blades arising under the action of centripetal forces associated with the rotation of the rotor. Moreover, the obtained values of the maximum stresses were several times higher than the maximum allowable ones. It was concluded that within the framework of the existing methodology for building the spatial form of the rotor, it is not possible to satisfy the conditions for ensuring strength, therefore, further improvement of the given version was stopped.



Longitudinal section

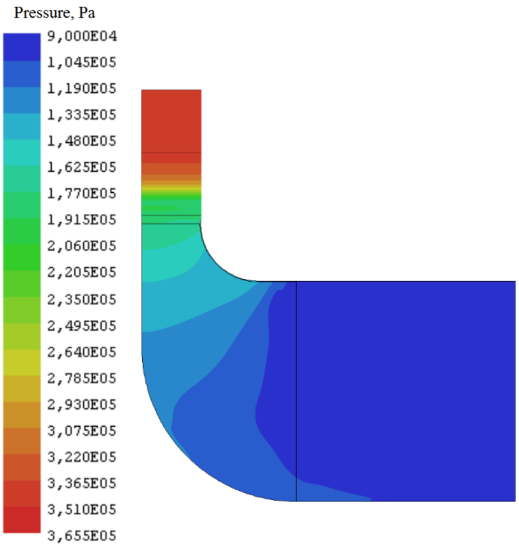


stator, mid-tangential section

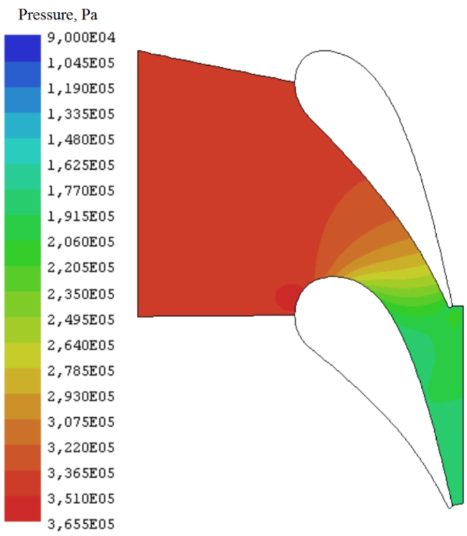


rotor, mid-tangential section

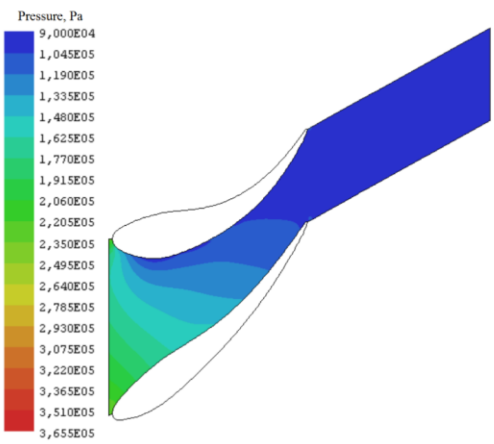
Fig. 4. Velocity vectors, nominal mode



Longitudinal section

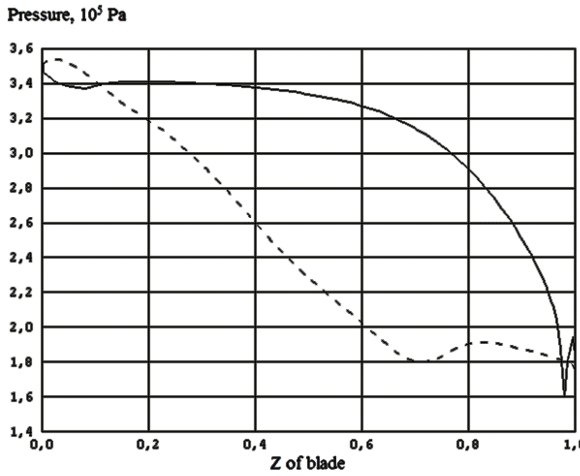


stator, mid-tangential section

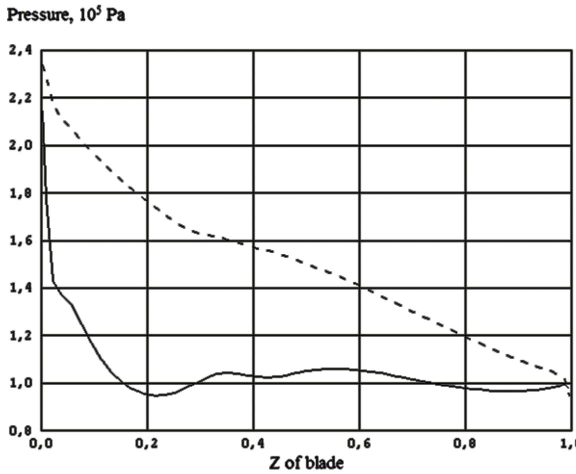


rotor, mid-tangential section

Fig. 5. Isolines of static pressure



a



b

Figure 6. Distribution of static pressure on the stator (a) and rotor (b) blade surfaces, where: x-axis is axial width of the blade in mid-tangential section, y-axis is static pressure

4. Centripetal turbine stage with thin rotor blades

Calculations of the second version were performed on difference mesh and the equation of state as for the first version.

To eliminate the mentioned shortcomings of the first version, the method of blades profiling of radial-axial flow parts was improved. The methodology provides such a shape of the blades, in which in the plane perpendicular to the axis of rotation of the rotor, the middle line of the blade coincides with the radial line. In this case, the possibilities of so-called "spatial profiling" [21, 23] are reduced, in particular, the compound lean (saber-like) of the blades, but this approach makes it possible to significantly reduce the load and stresses arising under the action of centripetal forces.

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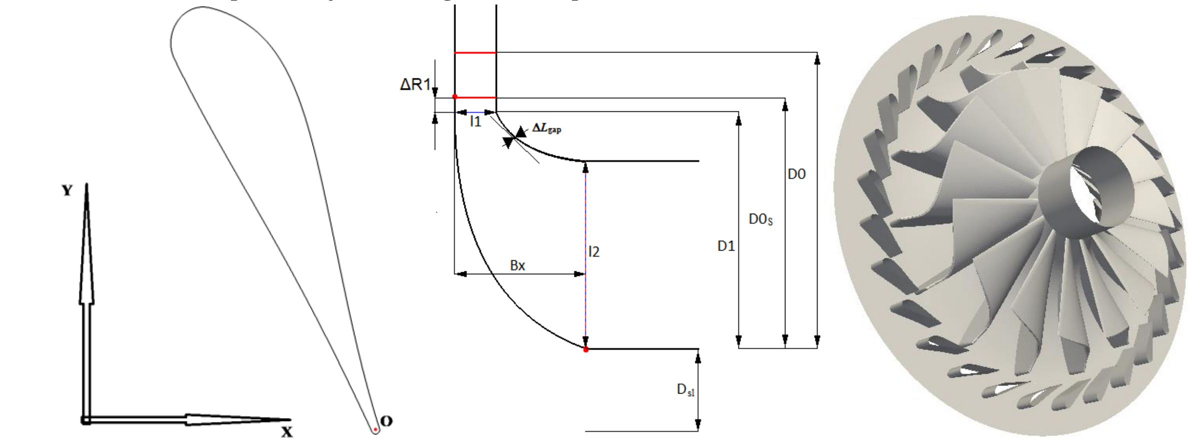
With the help of improved methods, the second version of the flow part with a rotor with thin (easier) blades was developed, the view of which is shown in Fig. 7.

The main dimensions of the flow part remained the same as in the first version.

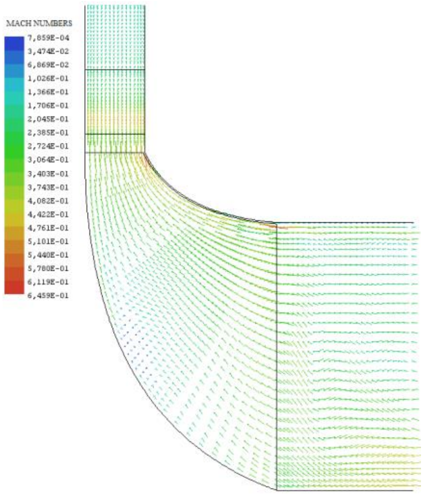
In addition to fundamentally different blades of the rotor, the shape of the meridional contours is significantly different in this flow part. This is due to the fact that due to the impossibility of using saber-shaped blades, a powerful separation of the flow was formed at the rotor outlet in the peripheral part. The proposed shape of the contours provides "pressing" of the flow to the periphery, which prevents its separation. Stator blades, which provide the required flow angle and lower kinetic energy losses, have also been improved.

Figs. 8-9 show visualization of the flow in the flow part, and Fig. 10 shows the distribution of static pressure on the surfaces of the stator and rotor blades.

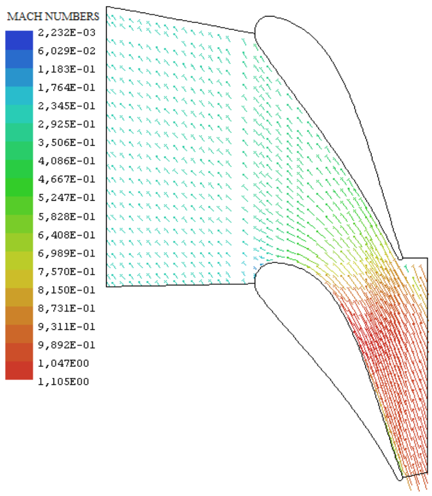


a) b) c)

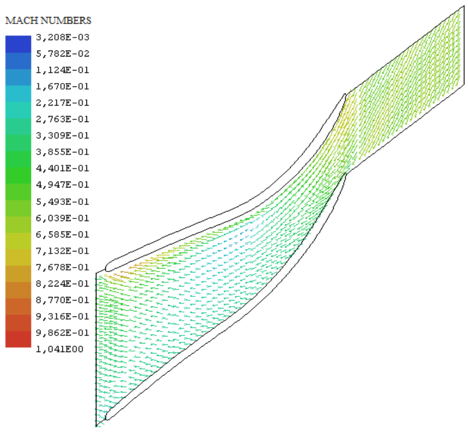
Fig. 7. Turbine flow part: a) – stator blade profile; b) – meridional cross-section; c) – isometry



Longitudinal section

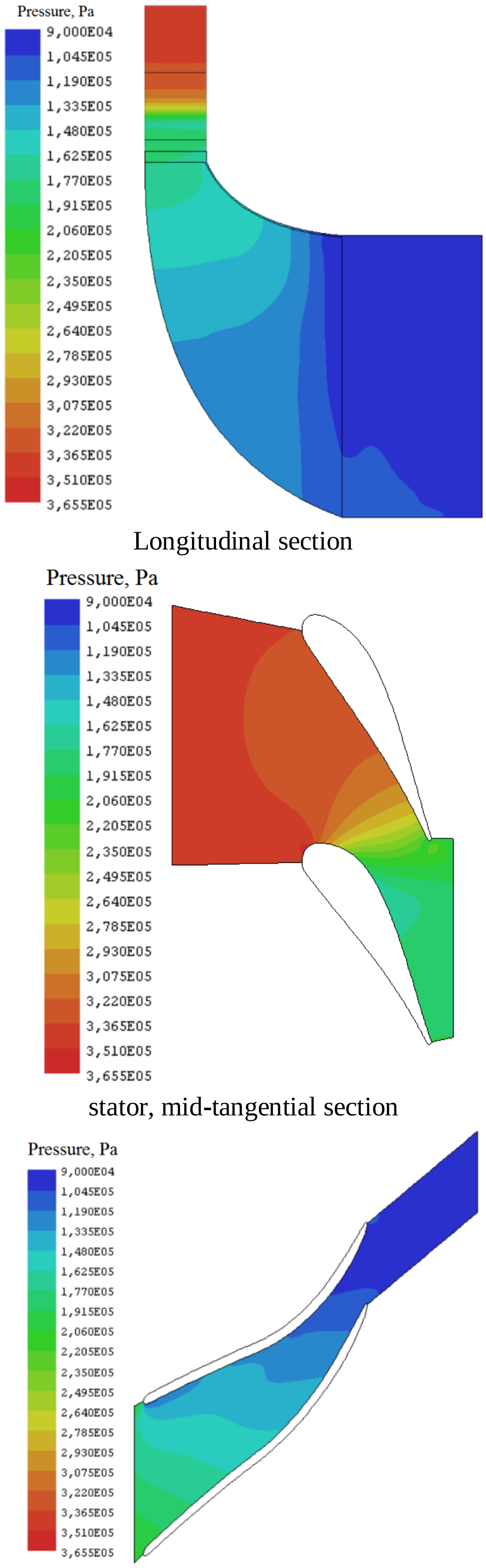


stator, mid-tangential section



rotor, mid-tangential section

Fig. 8. Velocity vectors



Longitudinal section

stator, mid-tangential section

rotor, mid-tangential section

Fig. 9. Isolines of static pressure

It can be seen that in the second version of the turbine stage flow part, the flow pattern is also favorable. There are no flow separations, but the pressure distributions are less monotonous. The integral gas-dynamic characteristics are:

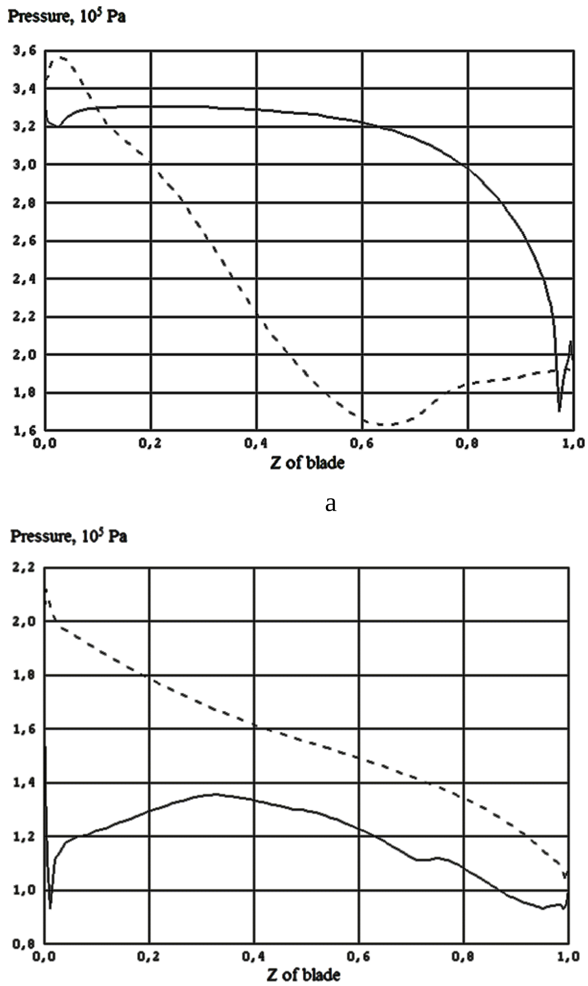
– kinetic energy losses – 6.4%;

– kinetic energy losses with outlet velocity – 3.7%;

– stage efficiency taking into account losses with outlet velocity – 89.7%;

– power – 161.6 kW.

The obtained results fully satisfy the requirements. Despite the fact that the efficiency of the turbine is slightly lower than in the first version, it is quite high, and the power is more than the design value (150 kW). In addition, testing of the centripetal turbine rotor showed that it meets the strength requirements and has an acceptable weight. This version was adopted as a basis for further design and production.



a

b

Fig. 10. Distribution of static pressure on the stator (a) and rotor (b) blade surfaces, where: x-axis is axial width of the blade in mid-tangential section, y-axis is static pressure

Conclusions

Two versions of the flow part, with profiled and thin rotor blades, are presented. From the given results, it can be seen that both versions of the developed flow parts have a high level of aerodynamic perfection and provide a satisfactory flow pattern.

The first version with profiled blades has better gas-dynamic indicators, as well as reserves for their further improvement, but it does not meet the requirements for strength.

To eliminate the mentioned shortcomings of the first version, the method of blades profiling of radial-axial flow parts was improved.

With the help of improved methods, a second version of the flow part with a rotor with thin blades, which has better characteristics in terms of strength and acceptable weight, was developed. This version was adopted as a basis for further design and production.

List of symbols

ΔR_1 radial clearance between the stator and rotor edges [mm]
ΔL_{gap} the gap between the rotor blades and the body [mm]
D_0 the maximum diameter of stator blades [mm]
D_1 the maximum diameter of rotor blades [mm]
D_{sl} rotor sleeve diameter [mm]
$B_{x\ max}$ axial size of rotor blades [mm]
η efficiency [%]
h enthalpy [J]
c_{0t} conditional rate of thermal drop in the stage [J];
c flow rate in absolute motion [m/s]
u circumferential speed [m/s]
w flow rate in a moving coordinate system [m/s]
y+ dimensionless distance to the wall [-]
α flow angle in absolute motion [^{0}]
β flow angle in the rotating coordinate system [^{0}]

Indexes

0 parameters at the inlet to the stage [-]
1 parameters between stator and rotor blades [-]
2 parameters at the outlet of the stage [-]
* parameters of retarded flow [-]
iz isentropic value [-]

Contributions of authors: conceptualization, methodology – **Andrii Rusanov, Igor Kravchenko, Sergiy Riznyk, Yurii Kukhtin;** formulation of tasks, analysis **– Igor Kravchenko, Sergiy Riznyk, Yurii Kukhtin;** development of model, software, verification **– Andrii Rusanov, Roman Rusanov, Sergiy Riznyk;** analysis of results, visualization **– Roman Rusanov, Andrii Rusanov, Marina Chugay;** writing – original draft preparation, writing – review and editing **– Marina Chugay, Mykhailo Sukhanov.**

Conflict of Interest

The authors declare that they have no conflict of interest in relation to this research, whether financial, personal, author ship or otherwise, that could affect the research and its results presented in this paper.

Financing

This study was conducted without financial support.

Data Availability

The manuscript has no associated data.

Use of Artificial Intelligence

The authors confirm that they did not use artificial intelligence methods while creating the presented work.

All the authors have read and agreed to the published version of this manuscript.

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ДОСВІД РОЗРОБКИ ПРОТОЧНОЇ ЧАСТИНИ ДОЦЕНТРОВОЇ ТУРБІНИ ПОВІТРЯНОГО СТАРТЕРА АВІАЦІЙНОГО ДВИГУНА

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Предметом розгляду у презентованій статті є вивчення підходів до проєктування проточної частини ступеня доцентрової турбіни, які розглядаються на прикладі повітряного стартера потужністю 150 кВт. **Мета** – розробка просторової форми проточної частини доцентрової турбіни повітряного стартера авіаційного двигуна. **Завдання:** визначити повну 3D форму доцентрової турбіни повітряного стартера авіаційного двигуна, достатню для розробки конструкторської документації, а також розрахувати газодинамічні характеристики турбіни, у тому числі частоту самогальмування. Дослідження виконані з використанням сучасних чисельних **методів** розрахунку й проєктування проточних частин радіально-осьових доцентрових турбомашин, реалізованих у вигляді програмного комплексу IPMFlow. Отримані такі **результати.** Розроблено два варіанти проточної частини з профільованими й тонкими лопатками робочого колеса. Перший варіант з профільованими лопатками має кращі газодинамічні показники, а також резерви для їх подальшої модернізації, але він не задовольняє вимогам міцності. Для усунення вказаних недоліків першого варіанта удосконалено методику профілювання лопаток радіально-осьових проточних частин, а вже за її допомогою розроблено другий варіант проточної частини з робочим колесом із тонкими лопатками, який має кращі характеристики за критеріями міцності та прийнятну вагу. Цей варіант було прийнято за основу для подальшого проєктування й виробництва. **Висновки.** Наукова новизна отриманих результатів полягає у вдосконаленні методики профілювання лопаток радіально-осьових проточних частин. Методика забезпечує таку форму лопаток, при якій у площині, перпендикулярній осі обертання ротора, середня лінія лопатки співпадає з радіальною лінією. У цьому випадку зменшуються можливості так званого «просторового профілювання», зокрема, складного навалу (шаблеподібності) лопаток. Крім того, такий підхід дав змогу суттєво зменшити навантаження і напруження, що виникають під дією відцентрових сил. За допомогою удосконаленої методики розроблено варіант проточної частини з робочим колесом із тонкими лопатками. Незважаючи на те, що ККД нової турбіни трохи нижчий, ніж у першому варіанті, він є досить високий, а потужність більша за проєктне значення.

Ключові слова: газотурбінний двигун; проточна частина; меридіональні обводи; просторова течія; газодинамічні характеристики; чисельні дослідження.

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