[1] **R. A. Rusanov**
[2] **A. V. Rusanov**, D. Sc.
[1] **P. Lampart**, D. Sc.
[2] **M. A. Chugay**, Ph.D.

[1] The Szewalski Institute of Fluid-Flow Machinery Polish Academy of Sciences, Gdansk, Poland
[2] The A. N. Podgorny Institute for Mechanical engineering problems NAS of Ukraine, Kharkov, Ukraine,
e-mail: rusanov@ipmach.kharkov.ua

Ключові слова: *радіально-осьова турбіна, проточна частина, аналітичний метод профілювання, просторова течія, чисельне моделювання, складний навал.*

УДК 621.165:532.6

IMPROVING THE EFFICIENCY OF RADIAL-AXIAL ROTORS OF TURBINE STAGES THROUGH THE USE OF COMPLEX LEAN OF TRAILING EDGES

Запропоновано модифікований аналітичний метод задання просторових радіально-осьових робочих коліс зі складними навалами вхідних і вихідних кромок. Досліджено вплив форми навалу вихідних кромок лопаток радіально-осьових робочих коліс на їх ефективність. Показано, що використання складного окружного навалу вихідних кромок дозволяє забезпечити істотне підвищення коефіцієнта корисної дії низьконавантажених радіально-осьових робочих коліс.

Introduction

Radial-axial turbines are widely used in various technical enterprises like: cogeneration plants working with low-boiling fluids, turboexpander units of various purpose, pump driving systems, etc. These turbines have a high efficiency applied to machines with a relatively small volumetric flow rate of the working fluid.

Until recently, blades the with «plate» profile have mainly been used in rotors of radial-axial turbines. Despite of that they provided a fairly high level of internal efficiency at design conditions. In off-design conditions or at varying operating modes, significant separated flows occurred due to leakages and off-design flow angles in the flow channels, leading to a significant reduction of the turbine efficiency. In order to improve the radial-axial turbine aerodynamic characteristics both for a design point and especially for off-design operating modes, blades of complex spatial shape with thick profiles were used [1-4].

The problem of blade profiling of stator blades for radial and radial-axial turbines has recently received an attention in paper [5]. The investigations were strongly motivated by the consequences of introduction of new technologies in the manufacture of turbomachinery. Such innovations, including the use of magnetic bearings, made it possible to create high-speed turbines, in which there are large thermal drops in one stage, resulting in an increased level of kinetic energy losses in the stator, becoming similar, or even higher, than the kinetic energy losses in rotor [6, 7]. The approach to stator blade profiling described in [5] have achieved a significant progress in creation of high-efficiency flow parts of radial-axial turbines with high and medium thermal drops.

In radial-axial turbines working with relatively low thermal drops, it is necessary to ensure a low level of losses due to the outlet speed. The radial-axial stages, as a rule, are designed in a way to obtain the absolute velocity at the outlet in a range between 50–60 m/s. In the case of significant thermal gradients, the kinetic energy losses connected with the output speed are relatively small (acceptable) in the overall loss balance and are equal to 2–4%. With a decrease in stage load, the losses due to the outlet speed will increase and can reach about 10%. To reduce them it is necessary to decrease the outlet speed to 25-30 m/s. It is possible by increasing the cross-sectional outlet area of the rotor, which leads to an increase in the relative height of the blade and reducing the value D/L at the outlet.

Because of technological and strength limitations trailing edges of radial-axial rotors usually coincide with radial lines. But for low-load stages with this form of trailing edges it is difficult to simultaneously obtain a low flow velocity at the outlet and an unseparated flow. The paper shows results of investigations of influence of the trailing edge shape on the flow structure and integral characteristics of radial-axial turbine stages, and describes a modified analytical method of profiling of spatial radial-axial rotors with complex lean of leading and trailing edges.

The modified analytical method of profiling of spatial radial-axial rotors

A view of a radial-axial rotor in a meridional plane is shown in fig. 1. The leading edge of the rotor blade is parallel with the axis of rotation x, and the trailing edge is perpendicular to it.

Tip and hub endwalls are described by curves consisting of an arc and a straight connected with the arc. Initial data for the construction of meridional endwalls are: r_{max}, r_{min} – maximum and minimum radiuses of the rotor channel; x_{max} rotor width; l_{in}, l_{ex} – inlet and exit heights; α_{in}^{hub}, α_{ex}^{hub}, α_{in}^{tip}, α_{ex}^{tip} – angles of the tip and hub endwalls at the rotor inlet and exit (fig. 1).

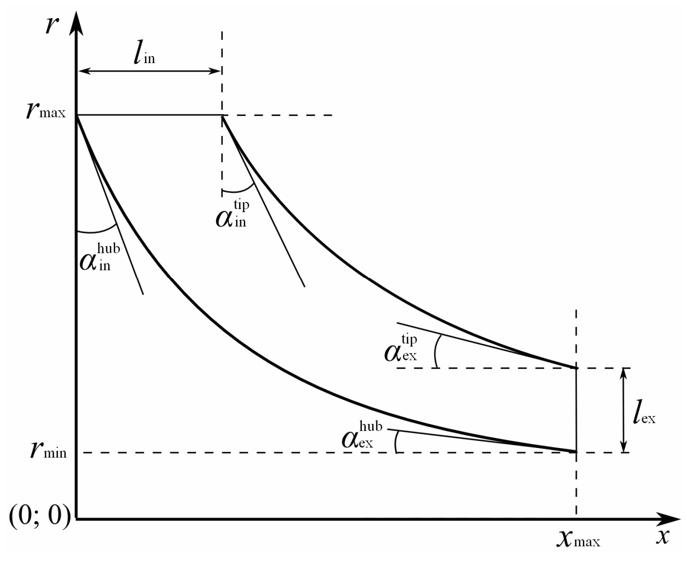


Fig. 1. A view of rotor in meridional plane

The blade is defined by two sections of rotational surfaces of the tip and hub endwalls. The sections are described by coordinates: $r\varphi$ – along the circumference; s – distance from the leading edge along the respective section at the hub endwall in projection onto the meridional plane (fig. 1). Profile coordinates on rotational surfaces of the tip and hub endwalls are found as a sum of coordinates for the mean line $r\varphi_{ml}$ (fig. 2, a) and profile thickness $\Delta r\varphi$ (fig. 2, b):

$$r\varphi(s) = r\varphi_{ml}(s) + \Delta r\varphi(s).$$

The mean line is a 3rd order polynomial (fig. 2, a):

$$r\varphi_{ml} = \sum_{i=0}^{3} a_i s^i ,$$

where a_i – coefficients.

The profile is assumed symmetrical with respect to the mean line (fig. 2, b). Each of profile curves (pressure or suction curve) consists of four parts conjugated by continuity of the first derivatives: 1–2 – leading edge; 2–3 – circle arc; 3–4 – straight interval; 4–5 – trailing edge.

The initial data for building the profile are: b_s – profile width; β_{in}, β_{ex} – mean line angles at the leading and trailing edge; r_{in}, r_{ex} – leading/trailing edge radiuses; c_{max} – maximum profile thickness; $\Delta\alpha$ – leading edge angle; d – distance from the trailing edge, where the second derivative of the mean line equals 0 (fig. 2).

In the case of a blade without lean, the position of the profile mean line on the surface of rotation located at the tip endwall is found first, then at the hub endwall. Points at opposite endwalls are similar when the dimensionless distance from the leading edges at relevant endwalls is equal:

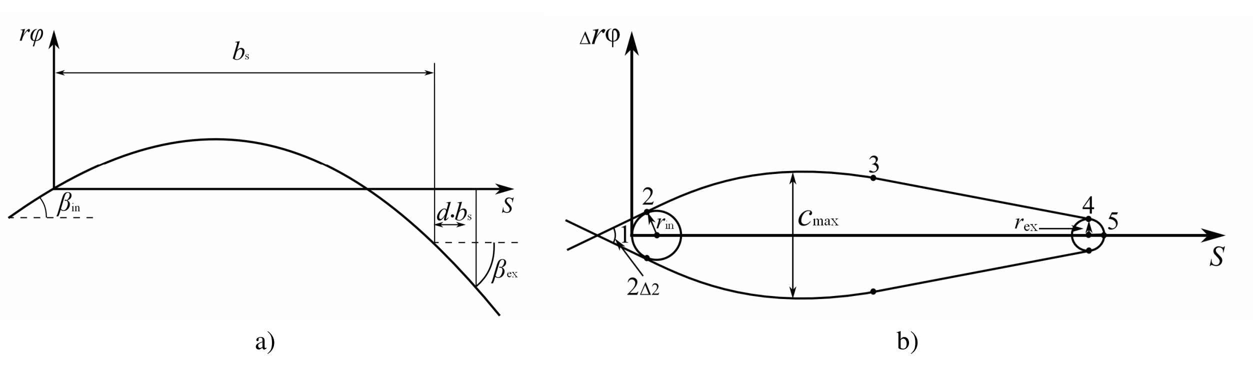


Fig. 2. Blade section:
a) – blade mean line; b) – blade thickness with respect to the mean line

$$\bar{s} = s/s_b \ ,$$

where s_b – distance from the leading edges at relevant endwalls.

In a point at the tip endwall the angle coordinate of the profile mean line is equal to that of the similar point at the hub endwall. In this way coordinates of hub endwall $\Delta r\varphi$ can be determined relative to the mean line. Similar points of profile mean lines at the tip and hub endwall are linked by straight lines to build a mean profile surface.

If it is necessary to create a blade with leans, the main difference from the above approach is in the description of angle coordinates of the profile mean line in similar points as a function of distance from the hub endwall:

$$\Delta\varphi_{\mathrm{ml}} = \Delta\varphi_{\mathrm{ml}}\left(l/l_{\mathrm{r}}, \bar{s}\right), \quad (1)$$

where l – distance from the hub endwall on the line of similar points, l_r – distance between the similar points at the hub and tip endwalls.

The angle coordinate function (1) is given in a way to enable the line of similar points of the profile mean lines have a shape of a straight line or an arc. The dependence of the function (1) from $\bar{s}$ is usually chosen as linear.

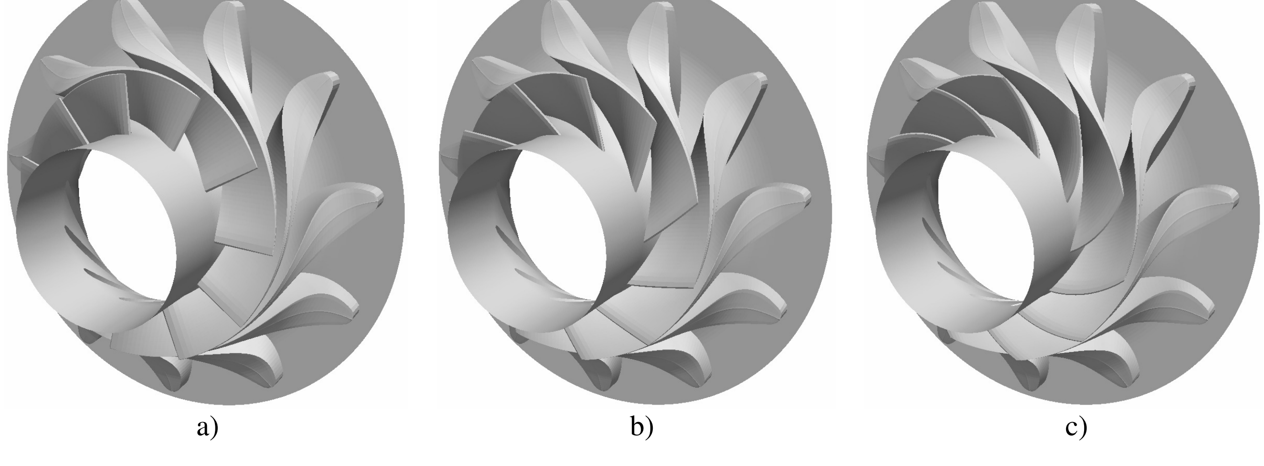
The study of influence of the form of blade trailing edges on the gas dynamic characteristics of radial-axial rotors of turbine stages

A radial-axial rotor designed for a turboexpander unit installed in a complex gas treatment plant working on a gas condensate field was chosen as an object of study. The turbine is designed for a pressure drop 1.225 with a rotational speed 8260 rpm. The total pressure and temperature at the inlet are equal to 5.52 MPa and –23 °C, respectively. A working medium is natural gas of the mass flow rate 24.5 kg/s. The rotors were developed using the method described before and were investigated without a stator. The inlet angle to the rotor was set based on the given mass flow rate, but with a restriction of the minimum value of 12° from the grid front. In the future, it is expected to develop the stator using the method described in [5].

Three types of rotors were created. They have the same geometry of meridional contours, but differ in the form of trailing edges (fig. 3). In the first variant, the trailing edges are radial, in the second variant they are straight with a circumferential lean, and in the third variant the trailing edges are assumed in the form of arcs leaned in circumferential direction. For each variant of the rotor its gas dynamic characteristics are close to an optimum for a given trailing edge shape.

For numerical investigations of flow, the software complex *IPMFlow*, which is the development of the software systems *FlowER* and *FlowER-U* is used [8, 9]. Computational grids consisted of more than 600 thousand cells.

Fig. 4 shows flow visualization in a meridional view at the mid blade-to-blade section for all variants of the rotor. Velocity vectors and isolines of static pressure are shown.



a) b) c)

Fig. 3. View of rotor:
a) – variant 1; b) – variant 2; c) – variant 3

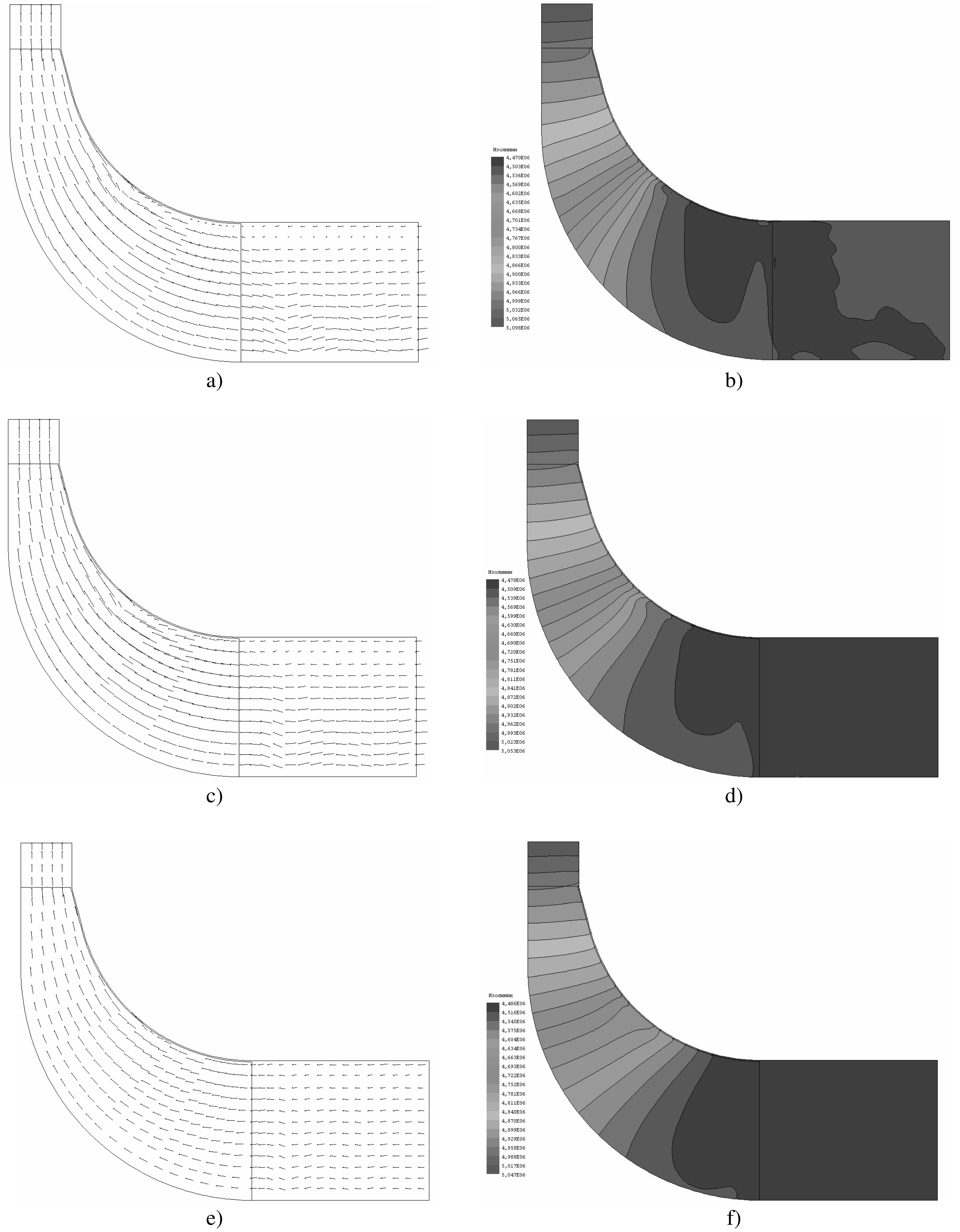


Fig. 4. Velocity vectors and static pressure contours in a meridional view at the mid blade-to-blade section:
a), b) –variant 1; c), d) – variant 2; e), f) – variant 3; a), c), e) – velocity vectors; b), d), f) – static pressure

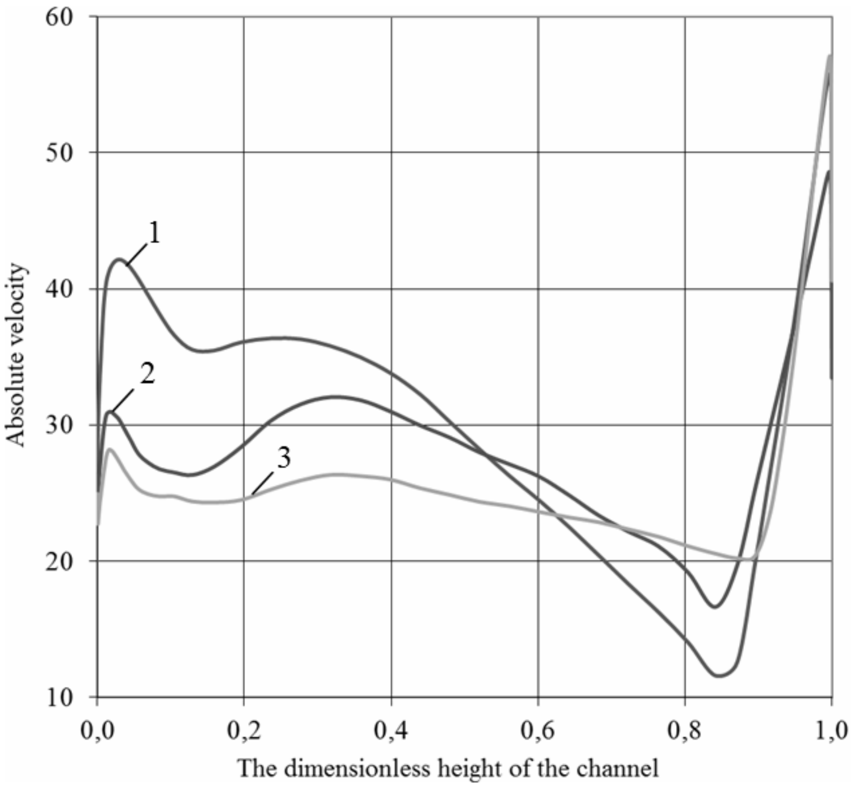


Fig. 5. The distribution of absolute velocity along the channel height at the outlet from the computational area:
1 – variant 1 of the rotor; 2 – variant 2; 3 – variant 3

The enclosed pictures exhibit the presence of flow separation at the tip endwall in the first variant of the rotor (fig. 4, a). The separation inception is located somewhere half distance from the leading to the trailing edge. The separation region is extended downstream almost to the exit from the calculation area. It is also seen that the minimum value of static pressure at the tip endwall is reached just after the start of the separation zone, far upstream of the trailing edges (fig. 4, b). This pressure distribution indicates that the tip part of the blade does not «work» through the large part of the blade-to-blade channel starting from the zone of pressure minimum. The described picture can be explained by a high curvature of the tip endwall and a fact the tip section is not enough «open» for flow.

In the second variant of the rotor, by leaning of the rotor blade trailing edges (fig. 3, b) the flow channel at the tip becomes more «open». This leads to an improvement in the overall flow pattern: flow separation at the tip is reduced and delayed downstream of the point of pressure minimum (fig. 4, c, d). Besides, the velocity distribution along the channel height at the outlet from the computational area becomes more uniform (fig. 5).

Consequently, an increased rotor gas-dynamic efficiency is observed in the form of reduced overall kinetic energy losses and outlet speed losses. As a result the overall stage efficiency (taking into account outlet speed losses) increases by 2,3% (table).

A further increase of straight lean of the trailing edge does not lead to an improvement because in this case blade-to-blade channels become more «closed» at the hub, which leads to deterioration of flow conditions there.

Table. Integral characteristics

Variant of the rotor	Outlet velocity, m/s	Kinetic energy losses, %	Outlet velocity losses, %	Rotor efficiency without outlet velocity losses, %	Rotor efficiency with outlet velocity losses, %
1	31,7	6,4	3,2	93,5	90,6
2	28,4	5,2	2,5	95,2	92,9
3	26,3	4,5	2,2	95,8	93,7

The third variant of the rotor with a complex lean allows for an even stronger «opening» of blade-to-blade channels at the tip. This only insignificantly «closes» the channel in the hub section. In this variant, the flow pattern becomes even more favorable than in the second variant and particularly better as compared with the first variant. The separation at the tip in the third variant is found to disappear almost completely (fig. 4, e). The minimum of the static pressure is shifted downstream and close to the trailing edge, while the picture of static pressure contours in the flow channel becomes more monotonous (fig. 4, f). A more uniform is also the distribution of absolute velocity at the outlet from the computational area (fig. 5). Besides, there is a further increase in the rotor gas-dynamic efficiency. The flow efficiency with outlet velocity losses is increased by 0,8% compared with the second variant and by 3,1% compared with the first variant (table).

Conclusions

The modified analytical method of profiling of spatial radial-axial rotors with lean of rotor blade leading and trailing edges is described in the paper. Three types of rotors, having the same geometry of meridional contours, but differing in the form of trailing edges were considered. In the first variant the trailing edges were radial, in the second variant they were straight with circumferential lean, and in the third variant the trailing edges were designed in the form of arcs leaned in the circumferential direction. The application of complex lean to trailing edges of the low-loaded radial-axial rotor blades allowed us to increase the flow efficiency in the rotor by 3,1% compared with the design with radial trailing edges.

References

1. *Pasquale, D.* Shape Optimization of an Organic Rankine Cycle Radial Turbine Nozzle / D. Pasquale, A. Ghidoni, S. Rebay // J. Eng. Gas Turbines Power. – 2013. – № 135 (4). – P. 042308–1–042308–13.
2. *Preliminary* design and performance estimation of radial inflow turbines: an automated approach / P. A. Jacob, Carlos Ventura, Andrew S. Rowlands, Emilie Sauret // Trans. ASME. J. Fluids Eng. – 2012. – № 134. – P. 1–13.
3. *Метод* проектирования высокоэффективных проточных частей турбодетандерных агрегатов / А. В. Русанов, С. В. Моисеев, П. Н. Сухоребрый [и др.] // Авиац.-косм. техника и технология. – 2012. – № 8 (95). – С. 67–72.
4. *Rusanov, A.* Designing and updating the flow part of axial and radial-axial turbines through mathematical modelling / A. Rusanov, R. Rusanov, P. Lampart // Open Eng. (formerly Central European J. Eng.). – 2015. – №. 5. – P. 399–410.
5. *Rusanov, R. A.* Analytical method for profiling of radial stator blades of turbine stages / R. A. Rusanov, A. V. Rusanov, P. Lampart, M. A. Chugay // Пробл. машиностроения. – 2016. – Т. 19, № 3. – С. 5–11.
6. *Significance* of loss correlations in performance prediction of small scale, highly loaded turbine stages working in Organic Rankine Cycles / P. Klonowicz, F. Heberle, M. Preißinger, D. Brüggemann // Energy. – 2014. – Vol. 72. – P. 322–330.
7. *Kurzrock, J. W.* Experimental Investigation of Supersonic Turbine Performance / J. W. Kurzrock // Am. Soc. Mech. Eng. – 1989. – 89–GT–238.
8. *Комплекс* програм розрахунку тривимірних течій газу в багатовінцевих турбомашинах «FlowER»/ С. В. Єршов, А. В. Русанов // Свідоцтво про державну реєстрацію прав автора на твір, ПА № 77; 19.02.96 – Державне агентство України з авторських та суміжних прав. – 1996. – 1 с.
9. *Русанов, А. В.* Математическое моделирование нестационарных газодинамических процессов в проточных частях турбомашин / А. В. Русанов, С. В. Ершов. – Харьков: Ин-т пробл. машиностроения им. А. Н. Подгорного НАН Украины, 2008. – 275 с.

Поступила в редакцию 20.11.16