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Ключові слова: *ORC, проточна частина, аналітичний метод профілювання, просторова течія, чисельне моделювання, модифіковане рівняння Бенедикта-Вебба-Рубіна з 32 членами.*

УДК 621.165:532.6

DEVELOPMENT OF THE 500 KW AND 1 MW ORC TURBINE FLOW PARTS

Розглянуто декілька варіантів проточних частин осьових турбін потужністю 50 кВт та 1 МВт для когенераційної установки, що використовує як робоче тіло силіконове масло (MDM). Єдиним геометричним обмеженням для проектування цих турбін було мінімальна висота лопатки – 20 мм. Остаточні тривимірні розрахунки всіх ступенів турбіни проведено з урахуванням реальних властивостей робочого тіла на основі модифікованого рівняння стану Бенедикта-Вебба-Рубіна. Газодинамічна ефективність розроблених проточних частин турбін задовольняє вимоги, що ставляться до енергетичних машин подібного роду.

Introduction

A promising technology for small scale cogeneration systems is Organic Rankine Cycle (ORC) [1]. Main components of this cogeneration cycle are a heat source, eg. ecological boiler fit to combust different kinds of biomass or biofuels, intermediate heat cycle to extract heat from flue gases to thermal oil as a heat carrier, and the main cycle including the evaporator, turbine driving a generator, recuperator, condenser and circulating pumps for the working medium and thermal oil. Depending on the available heat source and particular application there are variety of working media that can be used for ORC. The solution offers a possibility to apply low temperature heat sources, allows utilisation of different types of fuels, and also a modular construction which facilitates adaptation of the CHP unit to the required power range. One can think of micro CHP units dedicated for individual households of total heat capacity up to 20kWt and electric power up to 4kWe as well as small CHP modules dedicated for communal energy centres of total heat capacity 5MWt and electric power 1MWe.

The main topic of this work was creating of two types of axial turbine flow parts with electric power up to 0.5 and 1 MW for ORC cogeneration plant. MDM (silica oil) is used as the working media. The properties of the working media are described using the REFPROP software complex. For the numerical analyses software complex IPMFlow are used. Final 3D calculations of all turbine stages are provided with accounting the real properties of working media. The real properties are described using an analytical interpolation method for approximating the modified Benedict-Webb-Rubin equation with 32 members.

1. Method for design and gasdynamic evaluation of the turbine flow system

The design of the turbine flow system consists of several elements: computation of basic geometric and flow characteristics using 1D methods, building 3D geometry of the flow path, 3D calculations and optimisation.

Computation of basic geometric and flow characteristics.

This is made with the help of 1D equations of conservation of mass and rotalpy, appropriate treatment of velocity triangles and correlations for kinetic energy losses [2].

The search for the optimum geometric characteristics proceeds from a great number of variants taking into account the assumed flow and constructional restrictions:

- properties of the working medium (incorporated in the form of the Tamman equation), inlet/exit parameters;
- stage reactions (minimum and maximum values), blade heights (minimum and maximum values), stator exit angles (minimum and maximum values), rotor inlet angles in the relative frame (minimum and maximum values), rotor exit angles in the absolute frame (minimum and maximum values), maximum stator and rotor exit Mach numbers;

– mid-span radiuses at rotor inlet/exit (minimum and maximum values), stator and rotor throat areas (minimum and maximum values), rotational speed (minimum and maximum values).

As a result of computations, geometric and gasdynamic characteristics are obtained referring to the maximum possible turbine power: flow angles and velocities in absolute and relative frame, mid-span radiuses, blade heights, mean values of thermodynamic parameters, rotational speed and turbine power.

Building 3D geometry of the flow system

This is made by means of parametrization and analytical profiling of the blading system [3, 4]. The blades are defined by plane profiles (Fig. 1) described by leading and trailing edges, suction and pressure curves. The leading and trailing edges are circle arcs. The suction and pressure curves are polynomials of the 5-th and 4-th order, respectively:

$$y(x)=\sum_{i=0}^{5}a_i x^i\ ,\ a_i=\text{const}\,. \tag{1}$$

$$y(x)=\sum_{i=0}^{4}a_i x^i\ ,\ a_i=\text{const}\,. \tag{2}$$

The input data for building the profile cascade are: b_x – profile width, α_1 – camber line inlet angle; r_1 – leading edge radius; α_{2ef} – cascade effective angle; r_2 – trailing edge radius, t – cascade pitch; α_{2s} – geometric angle; $\Delta\alpha_1$, $\Delta\alpha_2$ – leading and trailing edge angles.

Coefficients of the suction curve (1) are found from the following set of equations:

$$\begin{cases} y_s'(x_{1s})=\text{tg}(\alpha_1+\Delta\alpha_1) \\ y_s''(x_{1s})=\{y_{s,0}''\} \\ y_s(x_O)=y_O \\ y_s'(x_O)=\text{tg}(\alpha_o) \\ y_s(x_{2s})=y_{2s} \\ y_s'(x_{2s})=\text{tg}\{\alpha_{2s}\} \end{cases},$$

where x_{1s}, y_{1s}, x_{2s}, y_{2s} are coordinates of the tangency points with the leading/trailing edge circles, α_{2s} and y_0'' are varying parameters chosen in a way to provide the assumed value of throat O and assure the minimum curvature. The throat can be calculated from the cascade pitch and effective angle

$$O=t\cos\alpha_{2ef}\,.$$

After the determination of the suction surface as well as leading and trailing edges, coefficients of the pressure curve (2) are found from the following set of equations:

$$\begin{cases} y_p(x_{1p})=y_{1p} \\ y_p'(x_{1p})=\text{tg}(\alpha_1-\Delta\alpha) \\ y_p''(x_{1p})=\{y_{p,0}''\} \\ y_p(x_{2p})=y_{2p} \\ y_p'(x_{2p})=\text{tg}\alpha_{2p} \end{cases},$$

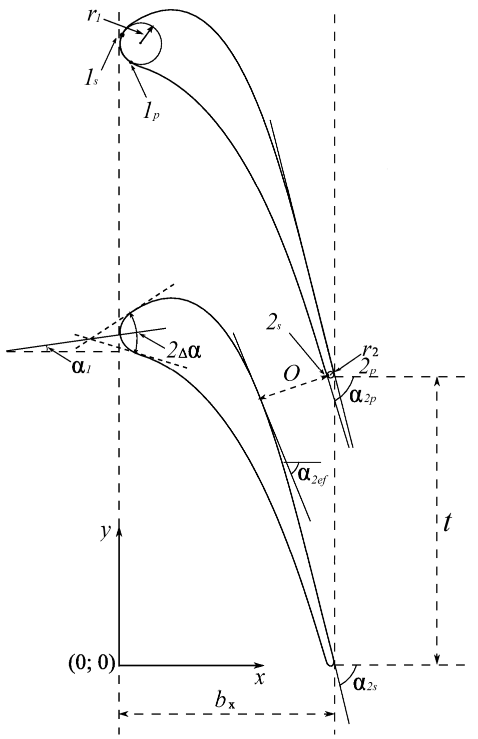


Fig. 1. Cascade of profiles

where x_{1p}, y_{1p}, x_{2p}, y_{2p} are coordinates of the tangency points with the leading/trailing edge circles, which can be found from the assumed angle α_1-$\Delta\alpha$ at the leading edge and angle α_{2p} at the trailing edge. The angle α_{2p} is found from the range between α_{2ef} and α_{2s} so as to assure the minimum curvature of the pressure curve.

In order to reduce the number of free parameters, initial data are given in three cross-sections. The cascade pitch is evaluated from the cross-section radius and blade number. Building a profile in any sections is based on quadratic approximations assuring a monotonous change of the profile surface.

Method for solving 3D flow

Flow computations are made with the help of a code ***IPMFlow*** [5, 6], which draws on the following mathematical models: Reynolds (Favre) averaging of Navier-Stokes equations, SST turbulence model of Menter, implicit quasi-monotonous high-order ENO scheme. The results of computations obtained from the code ***IPMFlow*** provide flow details and global characteristics of turbine blading systems [7, 8].

Methodology for determining the constants of the Tamman equation for the one-dimensional calculation

From the known parameters of flow at the inlet and the static pressure at the outlet of the flow part, using software complex REFPROP [9] defined the isentropic parameters at the output of the flow part. According to these parameters to perform one-dimensional calculation, the constant of the Tamman equation of state are defined as follows:

$$R = \frac{p_1 - p_2}{\rho_1 T_1 - \rho_2 T_2}; \quad p0 = R\rho_1 T_1 - p_1; \quad c_p = \frac{h_1 - h_2}{T_1 - T_2}; \quad \gamma = \frac{1}{1 - R/c_p}.$$

Constants of the Tamman equation of state determined individually for each stage.

Interpolation-analytical approximation of the modified Benedict-Webb-Rubin equation of state to account for the properties of real working fluids

This method was for the first time applied in [10] for 3D calculations of steam flow. Thermodynamic properties of water and steam were found from the equation of state IAPWS-95 [11].

The thermally modified Benedict-Webb-Rubin equation of state of with 32 members [12] has the form:

$$\begin{aligned} P = {} & \rho RT + \rho^2\left[G(1)T + G(2)T^{1/2} + G(3) + \frac{G(4)}{T} + \frac{G(5)}{T^2}\right] + \rho^3\left[G(6)T + G(7) + \frac{G(8)}{T} + \frac{G(9)}{T^2}\right] \\ & + \rho^4\left[G(10)T + G(11) + \frac{G(12)}{T}\right] + \rho^5[G(13)] + \rho^6\left[\frac{G(14)}{T} + \frac{G(15)}{T^2}\right] + \rho^7\left[\frac{G(16)}{T}\right] \\ & + \rho^8\left[\frac{G(17)}{T} + \frac{G(18)}{T^2}\right] + \rho^9\left[\frac{G(19)}{T^2}\right] + \rho^3\left[\frac{G(20)}{T^2} + \frac{G(21)}{T^3}\right]\exp(\gamma\rho^2) \\ & + \rho^5\left[\frac{G(22)}{T^2} + \frac{G(23)}{T^4}\right]\exp(\gamma\rho^2) + \rho^7\left[\frac{G(24)}{T^2} + \frac{G(25)}{T^3}\right]\exp(\gamma\rho^2) + \rho^9\left[\frac{G(26)}{T^2} + \frac{G(27)}{T^4}\right]\exp(\gamma\rho^2) \\ & + \rho^{11}\left[\frac{G(28)}{T^2} + \frac{G(29)}{T^3}\right]\exp(\gamma\rho^2) + \rho^{13}\left[\frac{G(30)}{T^2} + \frac{G(31)}{T^3} + \frac{G(32)}{T^4}\right]\exp(\gamma\rho^2), \end{aligned}$$

where R – individual gas constant, γ - specific heat ratio, G - BWR constants, P – pressure, ρ – density, T – temperature.

For the simulation of spatial viscous flows it is necessary to know thermodynamic dependencies for the following functions: u – internal energy; $h = u + \frac{p}{\rho}$ – enthalpy; c_v, c_p – isochoric and isobaric heat capacity; S – entropy; a – sonic speed. To determine these dependencies, differential equations of thermodynamics [13], equation (1) and a dependence for the Helmholtz free energy f are used.

Under this approach, the required thermodynamic functions were determined by the dependencies with the dimensionless compressibility coefficients for the corresponding thermodynamic functions determined by interpolation from a pre-calculated arrays of the base points. This approach has significant advantages in comparison with the "direct" interpolation of the thermodynamic function. This is due to the fact that the range of variation of dimensionless compressibility coefficients is relatively narrow and they are more monotonic functions compared with the other thermodynamic functions. Therefore, an acceptable accuracy of interpolation can be achieved with a much smaller dimension of the arrays of base points. Also to reduce the dimension of the array without a loss of accuracy, independent variables - pressure and density - are considered in a logarithmic scale. Third-order polynomials are used for the interpolation of the compressibility coefficients.

2. The calculation of basic geometric and gasdynamic characteristics of the turbine flow part

The calculation was performed from the first stage by methods described above. Was selected the maximum possible drop for stage (taking into account the selection of the optimal values of u/c for such a heat difference and the degree of reactivity of the first stages to 15%), subject to subsonic flow in the gaps (M <0,9). After calculating the first stage the obtained at the output flow values were taken as the flow parameters in the second stage. This procedure is repeated for all subsequent stages until the parameters have not been reached at the outlet of the turbine.

For developing the turbine flow parts were get the next initial data [14]:

– power 500 kW / 1 MW;
– the properties of the working fluid – software complex RefProp;
– inlet parameters: pressure – 1200 kPa; temperature - 553.5 K;
– outlet parameters: pressure – 17 kPa; rotational speed – 3000 rev/min;
– geometric constraints: minimum height of the blade – 20 mm.

3. 3D calculations of the designed 500 kW flow part

With the help of software package IPMFlow was made stage-to-stage 3D calculation of the flow part. The calculation is performed for a grid with the total number of cells in one stage more than 1 million. (about 500 thousand. cells in one blade). Fig. 2 is a view of designed using 3D methods flow part. After that there was provided the final calculations of all stages with taking into account the real properties of working fluid using the method described before. Fig. 3, 4 shows visualization of the flow in the flow part.

The total capacity of the flow part amounted to 0.6 MW, kinetic energy losses amounted to 6.73%, and the efficiency 94.05%.

4. 3D calculations of the designed 1 MW flow part

Fig. 5 is a view of designed using 3D methods flow part. Fig. 6-7 shows visualization of the flow in the flow part.

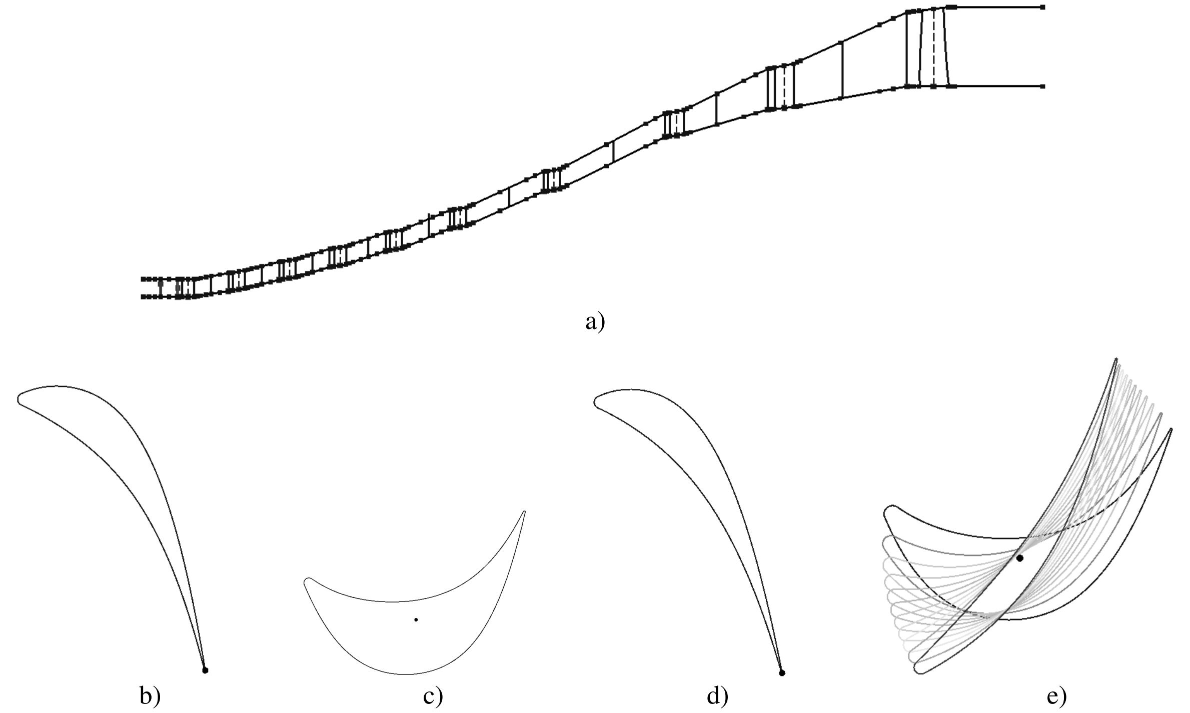


Fig. 2. View of the flow part designed using 3D methods:
a) – meridional section; b) – stator of 9th stage; c) – rotor of 9th stage;
d) – stator of 10th stage; e) – rotor of 10th stage

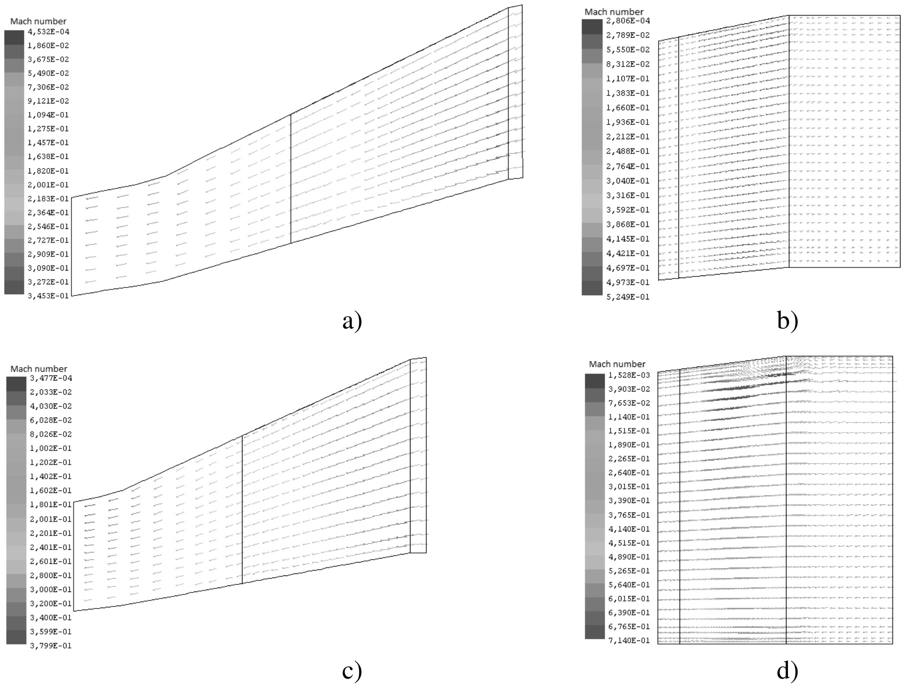


Fig. 3. Velocity vectors on the average meridional section:
a) – stator of 9th stage; b) – rotor of 9th stage; c) – stator of 10th stage; d) – rotor of 10th stage

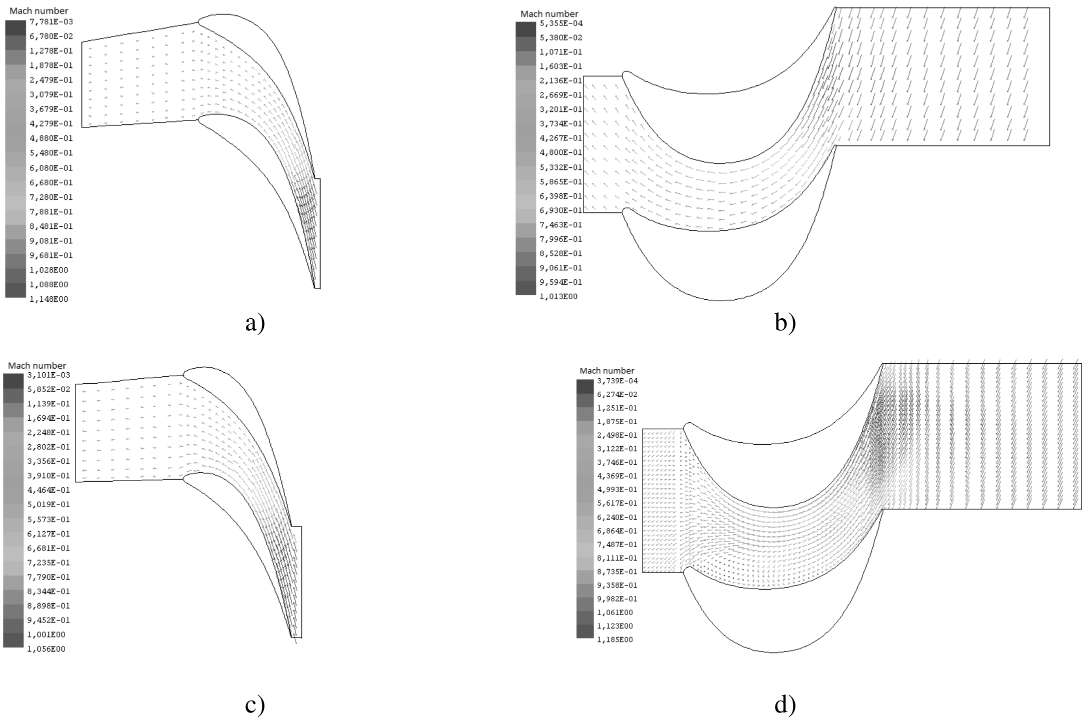


Fig. 4. Velocity vectors on the average tangential sectio:
a) – stator of 9th stage; b) – rotor of 9th stage; c) – stator of 10th stage; d) – rotor of 10th stage

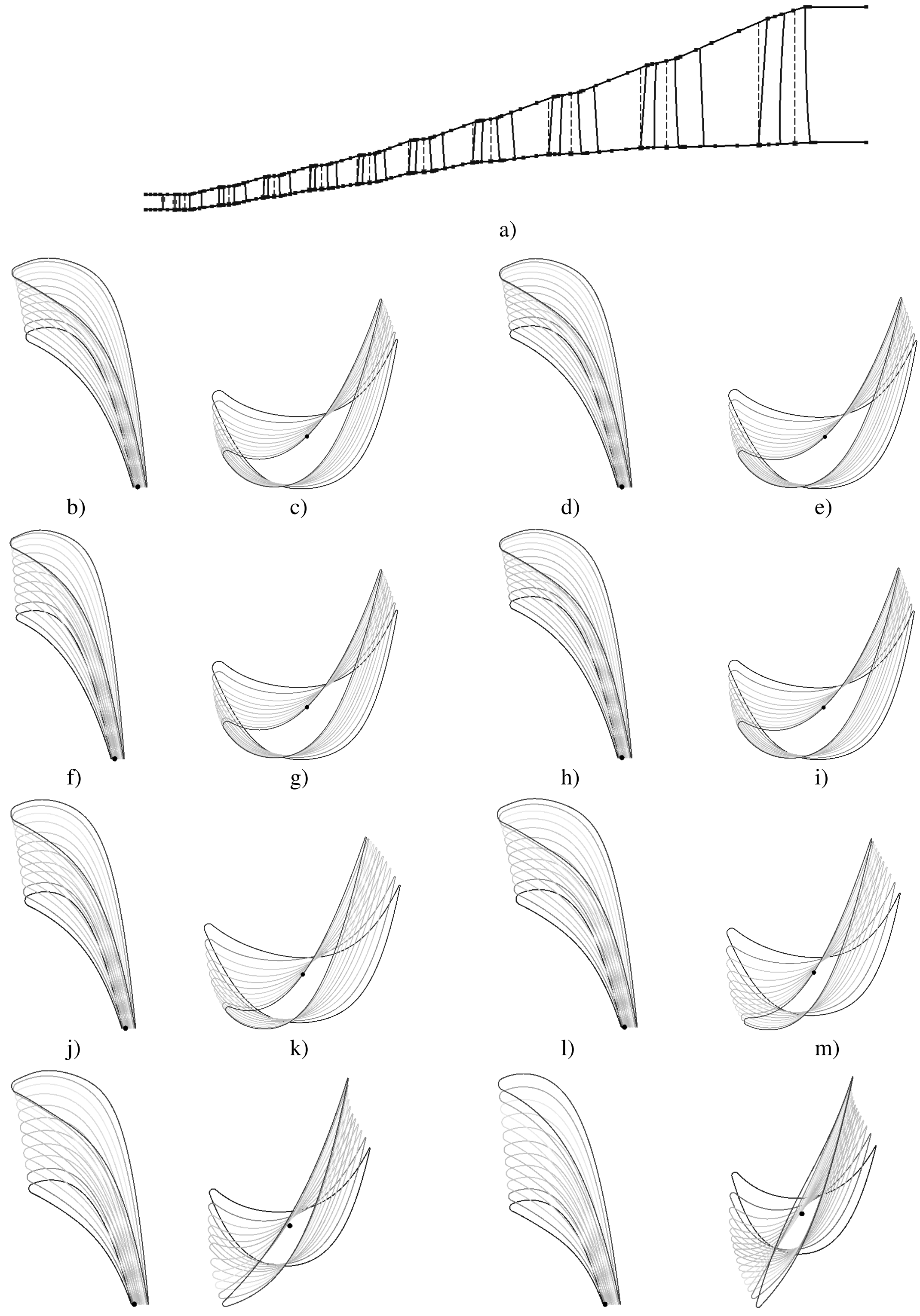
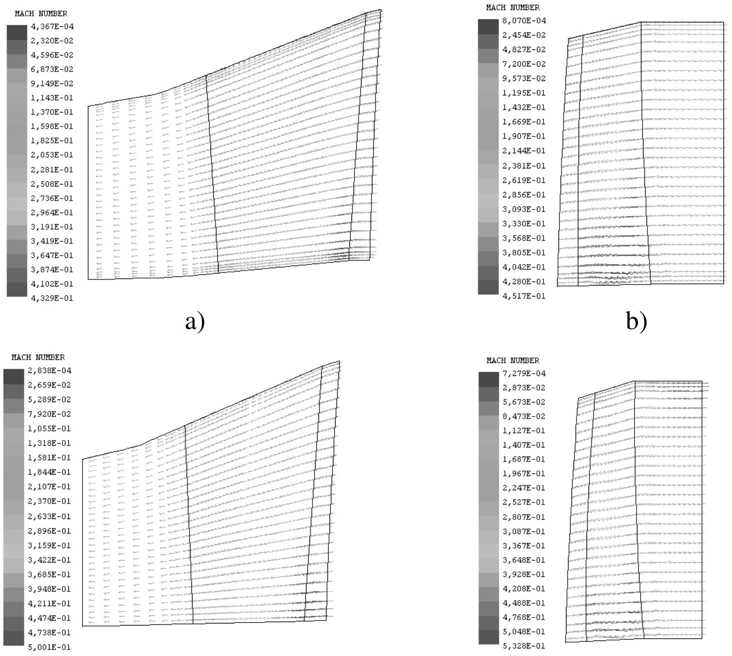
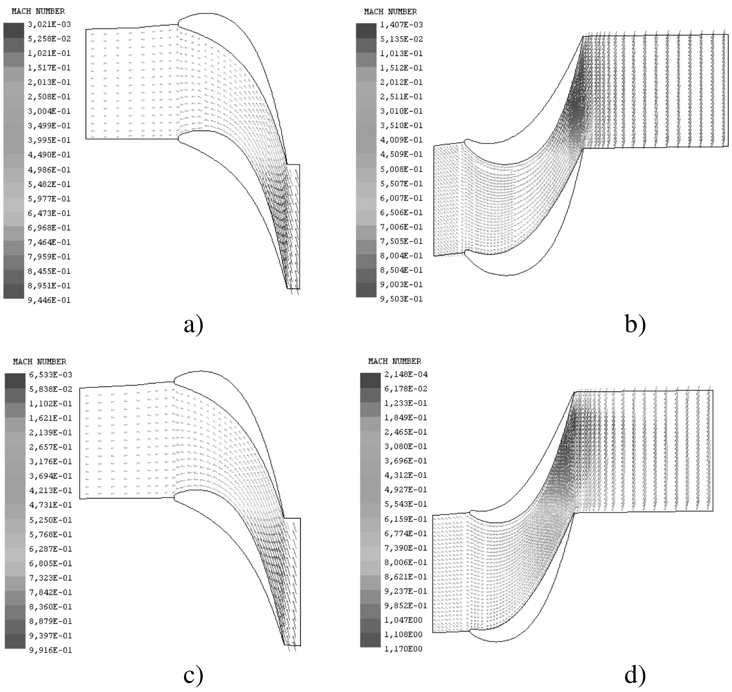


Fig. 5. View of the flow part designed using 3D methods:
a) – meridional section; b) – stator of 3rd stage; c) – rotor of 3rd stage; d) – stator of 4th stage; e) – rotor of 4th stage; f) – stator of 5th stage; g) – rotor of 5th stage; h) – stator of 6th stage; i) – rotor of 6th stage; j) – stator of 7th stage; k) – rotor of 7th stage; l) – stator of 8th stage; m) – rotor of 8th stage; n) – stator of 9th stage; o) – rotor of 9th stage; p) – stator of 10th stage; q) – rotor of 10th stage



a) b) c) d)

Fig. 6. Velocity vectors on the average meridional section:
a) – stator of 9th stage; b) – rotor of 9th stage; c) – stator of 10th stage; d) – rotor of 10th stage



a) b) c) d)

Fig. 7. Velocity vectors on the average tangential section:
a) – stator of 9th stage; b) – rotor of 9th stage; c) – stator of 10th stage; d) – rotor of 10th stage

The total capacity of the flow part is 1.21 MW, kinetic energy losses is 5.98%, and the efficiency – 95.19%.

Conclusion

Based on authors experience it can be argued that the resulting flow part with acceptable accuracy will correspond to the initial data and have satisfactory efficiency indicators.

With the help of 3D methods obtained the final form of the flow part of the turbine in which there are practically no separated flows, low loss of kinetic energy, obtained high efficiency of the flow part.

The total capacity of the designed flow parts are 0.6 MW and 1.21 MW respectively, and the efficiency is up to 95.19%.

References

1. *Duvia, A.* ORC plants for power production from biomasss from 0.4 to 1.5 MWe / A. Duvia, M. Gaia // Technology, efficiency, practical experiences and economy, Proc. 7th Holzenergie Symposium, ETH Zürich. – 2002.
2. *Щегляев, А. В.* Паровые турбины / А. В. Щегляев. – М.: Энергия, 1976. – 358 с.
3. *Русанов, А. В.* Метод аналитического профилирования лопаточных венцов проточных частей осевых турбин / А. В. Русанов, Н. В. Пащенко, А. И. Косьянова // Восточно-Европейский журнал передовых технологий. – 2009. – № 2/7 (38). – С. 32–37.
4. *Русанов, А. В.* Профилирование радиально-осевых турбин с использованием современных компьютерных технологий/ А. В. Русанов, О. И. Шатравка, А. И. Косьянова // Восточно-Европейский журнал передовых технологий. – 2009. – № 4/4 (40). – С. 58–62.
5. *А.с.* Комплекс програм розрахунку тривимірних течій газу в багатовінцевих турбомашинах «FlowER»/ С. В. Єршов, А. В. Русанов. – Державне агентство України з авторських та суміжних прав, ПА № 77; 19.02.96. – 1 с.
6. *Русанов, А. В.* Математическое моделирование нестационарных газодинамических процессов в проточных частях турбомашин / А. В. Русанов, С. В. Єршов. – Харьков: ИПМаш НАН Украины, 2008. – 275 с.

7. *Lampart, P.* Validation of 3D RANS Solver With a State Equation of Thermally Perfect and Calorically Imperfect Gas on a Multi-Stage Low-Pressure Steam Turbine Flow / P. Lampart, A. Rusanov, S. Yershov // J. of Fluids Eng. – 2005. – Vol. 127. – P. 83–93.
8. *Lampart, P.* Increasing flow efficiency of high-pressure and low-pressure stream turbine stages from numerical optimization of 3D blading / P. Lampart, S. Yershov, A. Rusanov // Engineering Optimization. – 2005. – Vol. 37. – P. 145–166.
9. *REFPROP,* National Institute of Standards and Technology Standard Reference Database Number 23. – Available from:: http://www.nist.gov/srd/nist23.htm.
10. *Русанов, А. В.* Интерполяционно-аналитический метод учета реальных свойств газов и жидкостей / А. В. Русанов // Восточно-Европейский журнал передовых технологий. – 2013. – № 3/10 (63). – С. 53–57.
11. *IAPWS,* Revised Release on the IAPWS Formulation 1995 for the Thermodynamic Properties of Ordinary Water Substance for General and Scientific Use. – Available from: http://www.iapws.org.
12. *Younglove, B. A.*, Thermophysical Properties of Fluids II Methane, Ethane, Propane, Isobutane, and Normal Butane / B. A. Younglove, J. F. Ely // J. Physical and Chemical Reference Data. – 1987. – Vol. 16, No. 4. – P. 577–798.
13. *Нащокин, В. В.* Техническая термодинамика и теплопередача / В. В. Нащокин. – М.: Высш. шк., 1980. – 496 с.
14. *Opracowanie* kanału przepływowego turbiny osiowej ORC na czynnik roboczy MDM 500 kW i 1 MW z łopatkami kształtowanymi wzdłuż wysokości kanału / R. Rusanov, M. Szymaniak, Ł. Jędrzejewski, P. Bagiński. – Nr arch. 1063/2014, Gdańsk, IMP PAN.

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Ключові слова: *турбогенератор, обмотка ротора, тепловий стан, метод скінченних елементів.*

УДК 621.1.016.7

К ВОПРОСУ МАТЕМАТИЧЕСКОГО МОДЕЛИРОВАНИЯ ТЕПЛОВОГО СОСТОЯНИЯ РОТОРА ТУРБОГЕНЕРАТОРА МОЩНОСТЬЮ 550 МВТ, ОХЛАЖДАЕМОГО ВОДОРОДОМ

Виконано моделювання теплового стану вузлів ротора синхронного турбогенератора потужністю 550 МВт з безпосереднім охолодженням обмоток воднем. Температурне поле ротора досліджено за допомогою методу скінченних елементів у тривимірній постановці. Наведено кореляції для визначення коефіцієнтів теплопередачі, що були отримані різними вченими під час проведення серії експериментів і обрано ту, що забезпечує відповідність розрахункового розподілу температур у роторі до випробувального.

Введение

Турбогенератор мощностью 550 МВт относится к генераторам большой мощности. Схема охлаждения представлена на рис. 1.

Основные узлы ротора турбогенератора: вал, обмотка возбуждения, уложенная в продольные пазы в теле ротора, бандажные кольца, удерживающие лобовые части обмотки от действия центробежных сил, центробежные компрессоры с двух сторон вала, обеспечивающие непосредственное принудительное охлаждение обмотки ротора водородом.

Ротор исследуемого генератора мощностью 550 МВт двухполюсный, обмотка возбуждения уложена в 36 пазах и выполнена из 18 концентрических катушек (по 9 катушек на каждый полюс). На каждом полюсе имеется по одной малой катушке, состоящей из 5 витков. Остальные катушки – большие, выполнены из 7 витков. Виток состоит из двух медных проводников П-образной формы, образующих канал для прохождения водорода.