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METHODS FOR DESIGN OF AXIAL TURBINES FOR ORC COGENERATION UNIT WORKING WITH MDM

The paper describes two methods for the design of blading systems of axial turbines for an Organic Rankine Cycle (ORC) cogeneration unit working with silica oil MDM. The algorithms are based on mathematical models of various levels of complexity – from 1D to 3D. Geometry of flow parts is described with the help of analytical methods of profiling using a limited number of parameters. The 3D turbulent flow model is realised in the software complex *IPMFlow*, which is developed based on the earlier codes *FlowER* and *FlowER-U*, or in software complex *ANSYS*. Examples of developed turbines for a 500 kW machine are presented.

Key words: cogeneration unit, Organic Ranking Cycle, CFD, axial turbine.

Introduction

A promising technology for cogeneration based on local energy resources is Organic Rankine Cycle (ORC) technology. In this Rankine-like technology the power unit is driven by a vapour of a working medium other than water. There is a variety of available working media, which can be effectively selected for a wide range of operating conditions. The medium considered in this paper – MDM belongs to the class of siloxanes and its properties make it suitable for applying in a middle temperature power plant cycle. From the point of view of thermodynamics it reveals a significantly different behavior than the ideal gas [1]. Therefore, in order to obtain a proper working point of the turbine and avoid the unnecessary losses, an appropriate state equation is used in this paper whose coefficients in the assumed expansion region are found from tables of medium properties.

With an ever-growing shortage of energy resources one of the main requirements for modern turbine plants is their high efficiency. The increased efficiency of turbomachinery is possible as a result of their gas-dynamic perfection, for example optimisation of plane sections of the blades, spatial blade profiling, special profiling of meridional contours, etc. [2–4] with the help of computational fluid dynamics (CFD) methods.

An approach to the design of flow parts of axial turbine stages based on the use of mathematical models of various levels of difficulty is presented in this paper. To describe the geometry of flow parts a method of analytical profiling is provided, which uses a limited number of parameters. The paper describes two methods of turbine design, the first method was developed at the A.N. Podgorny Institute for Mechanical Engineering Problems of National Academy of Sciences of Ukraine (IPMach NASU) in Kharkov, the second - at the Szewalski Institute of Fluid-Flow Machinery, Polish Academy of Sciences (IFFM PAS) in Gdansk. A 3D turbulent flow model for the first method is realised in the program complex *IPMFlow*, which originates from the programs *FlowER* and *FlowER-U*. A 3D turbulent flow model for the second method is found from the program complex *ANSYS*.

An algorithm for the designing of turbine flow parts developed at IPMach NASU

An algorithm for the designing of turbine axial and radial-axial types is introduced in Figure 1. The first step is a preliminary calculation of geometrical characteristics of the flow part using relatively simple methods, based on solving of one-dimensional or quasi-axial-symmetric equations [5, 6].

To construct the full three-dimensional geometry of the flow part on the basis of obtained geometric characteristics, analytical methods of profiling of flow parts are used. These methods allow us to build turbine stages of axial and radial-axial types.

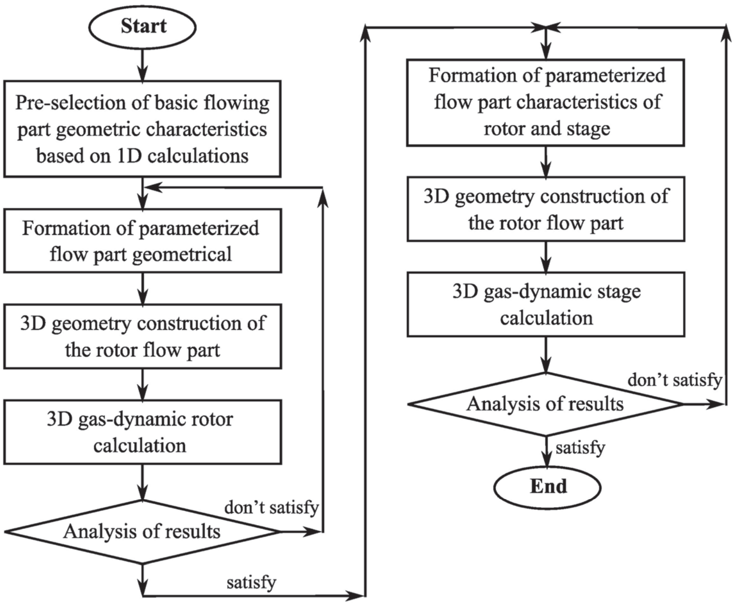


Fig. 1 – Flowchart of flow part stage designing

Calculations of three-dimensional flows in the resulting flow parts of turbine stages are performed using the program complex *IPMFlow*. Most of 3D calculations are performed on coarse grids (with a relatively small number of cells). The refined grids are used only for final calibration calculations. The algorithm of designing the flow part is automated. Several methods for solving optimization tasks such as the Nelder-Mead method and genetic algorithms are used to generate new values of variable parameters (geometric characteristics of the flow part). An automated approach usually requires several hundreds of iterations [5, 7]. Stochastic methods allow for searching of global extremes, but this usually involves increased computational costs [8]. In the given examples, it took up to one hundred 3D calculations to obtain the final form of one stage.

Building of 3D geometry of the flow part

The program developed at IPMach NASU is used for blade profiling (fig. 2). Using this program, we can build 3D blade profiles of different complexity, such as shown in figure 3.

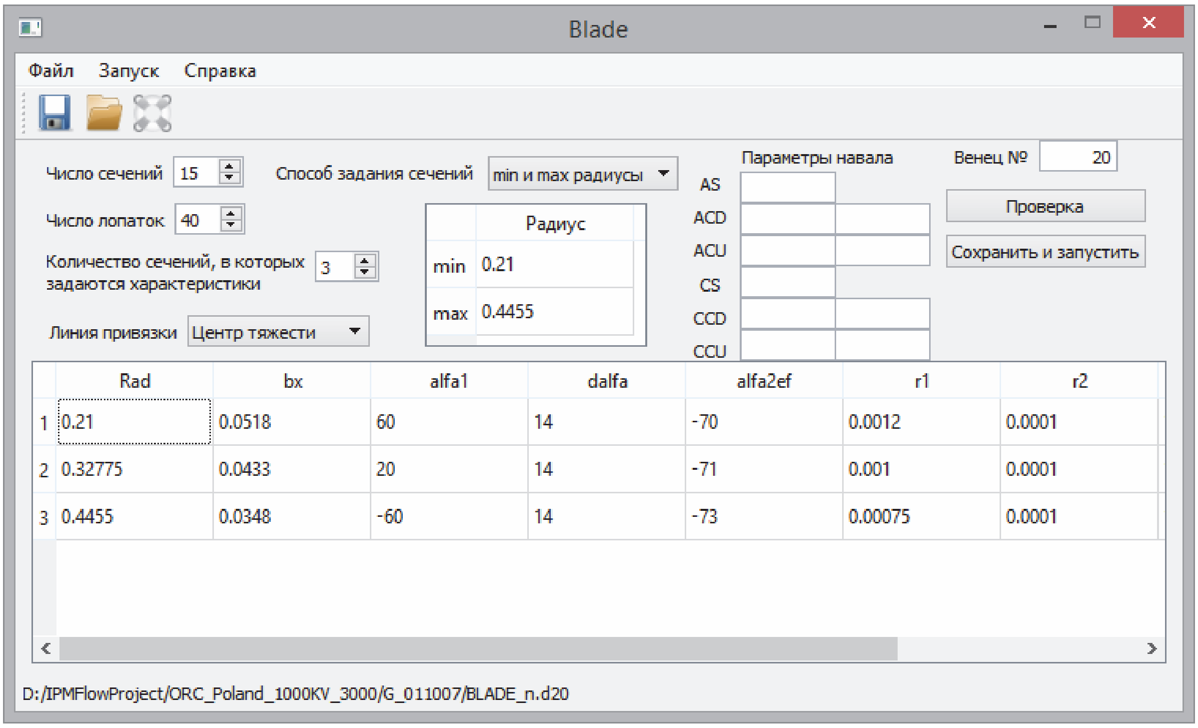


Fig. 2 – Program for blade profiling

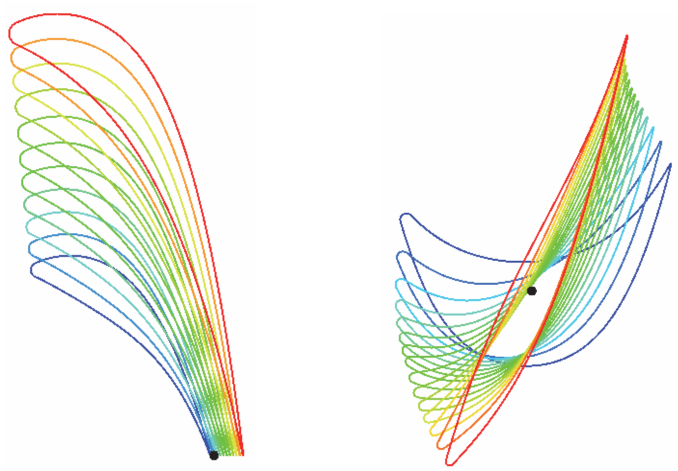


Fig. 3 – View of built profiles of stator and rotor

For building geometry of axial flow turbine blades in this program, the method of parameterization and analytical profiling described in [9] is used. The blade is given by an arbitrary set of flat profiles, each of which is considered in the Cartesian coordinate system with the x-axis parallel to the axis of the turbine and the y-axis coinciding with the cascade front (Fig. 4).

The blades are defined by plane profiles (Fig. 4) 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:

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

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

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_{2c} – "bevel" angle of pressure curve, $\alpha_\infty = \alpha_{2s} + \alpha_{2c}$; $\Delta\alpha_1$, $\Delta\alpha_2$ – leading and trailing edge angles. $1p$, $2p$, $1s$, $2s$ are points of conjugation of the input and output edges with suction and pressure side curves, respectively (Fig. 4).

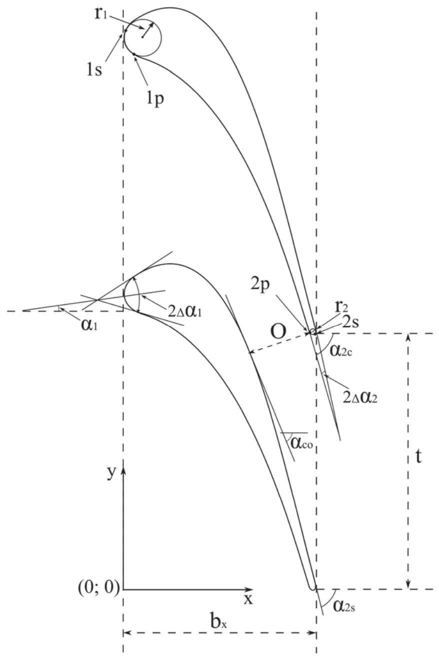


Fig. 4 – Profile cascade

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

$$\begin{cases} y_s'(x_{1s}) = \mathrm{tg}(\alpha_1 + \Delta\alpha_1) \\ y_s''(x_{1s}) = \{y_{s,0}''\} \\ y_s(x_O) = y_O \\ y_s'(x_O) = \mathrm{tg}(\alpha_o) \\ y_s(x_{2s}) = y_{2s} \\ y_s'(x_{2s}) = \mathrm{tg}\{\alpha_{2s}\} \end{cases}; \quad (3)$$

where x_{1s}, y_{1s}, x_{2s}, y_{2s} are coordinates of the tangency points with the leading/trailing edge circles, which are determined by the given angle $\alpha_1 + \Delta\alpha_1$ at the inlet edge. Parameters α_{2s} and y_0'' in equation (3) are chosen in a way to provide the assumed value of throat O and assure the minimum curvature of the curves (1).

The throat can be calculated from the cascade pitch and effective angle

$$O = t\cos\alpha_{2ef}. \quad (4)$$

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

$$\begin{cases} y_p(x_{1p}) = y_{1p} \\ y_p'(x_{1p}) = \mathrm{tg}(\alpha_1 - \Delta\alpha) \\ y_p''(x_{1p}) = \{y_{p,0}''\} \\ y_p(x_{2p}) = y_{2p} \\ y_p'(x_{2p}) = \mathrm{tg}\alpha_{2p} \end{cases}, \quad (5)$$

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 $\alpha_1 - \Delta\alpha$ 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 (Fig. 4) or $\alpha_{2s} - \Delta\alpha_2$ is given.

3D flow calculation method

For numerical investigations of flow, the software complex ***IPMFlow*** is used, which is the development of the software systems ***FlowER*** and ***FlowER-U***. It implements the following elements of the mathematical model: the Reynolds-averaged non-stationary Navier-Stokes equations, SST differential two-parameter model of turbulence of Menter, implicit quasi-monotone high-order ENO-scheme. To account for real thermodynamic properties of the working fluid, the Tammann equation of state or the modified Bennedict-Webb-Rubin equation of state with 32 coefficients are used. The results of computations obtained from the code ***IPMFlow*** have the necessary reliability in the qualitative structure of the flow and in the quantitative characteristics of the isolated turbine cascades and turbine as a whole [10, 11].

Example of design of the flow path for a cogeneration ORC turbine

The designed ORC turbine is a 500 kW machine operating on MDM as a working medium. The inlet parameters were determined as: pressure – 12 bar, temperature 553.5 K. For the cogeneration ORC installation, the temperature in the condenser was set at 363 K at the saturation pressure of 0.17 bar. The ORC cycle is equipped with a recuperator to increase the cycle efficiency. The resulting pressure drop in the turbine blading system is equal to 70.

The obtained flow path consists of 10 stages. The minimum blade height (for the first stage) is equal to 20 mm. The last stage has 3D shaped stator and rotor blades. The preliminary design was obtained from stage-to-stage calculations of the flow part using 1D methods. 3D calculations were performed on a grid of more than 1 million cells per one stage (about 500 thousand cells in one blade-to-blade passage). A view of the design showing the meridional section of the turbine flow path and hub-to-tip sections of the last stage rotor is presented in Fig. 5. Fig. 6 shows flow visualisation in subsequent turbine stages. The flow is regular. The kinetic energy losses in the turbine are relatively low and amount to 6.7 %, with the leaving energy loss included but the tip leakage loss not included.

The method for the designing of turbine flow parts developed at IFFM PAS

The in-house program allows one to design multistage axial turbines by means of one-dimensional meanline method. Additionally, it creates the geometry of the blades including three-dimensional shaping of the blade (blade twisting).

The program accepts the following design parameters: p_{0t}, T_{0t}, p_{sk}, $\dot{m}$, n, L_s. Additionally, one must specify the initial distributions of the enthalpy drop ratios, degrees of reaction and the axial widths of the blade rows. Optionally, there is a possibility of defining the distribution of the partial admission along the stages. The 1D module of the program is shown in Fig. 7.

In order to obtain a proper working point of the ORC turbine and to avoid the unnecessary losses, an appropriate equation of state must be used in the machine design process. The method described here uses a Helmholtz free energy equation of state adopted for the family of siloxanes [12] implemented in the NIST Refprop library in version 9.1 [13].

The losses in the blade channels are approximated on the basis of three optional correlations: Craig-Cox [14], Traupel [15] and Filipov [16]. The first of them is set as the default model according to the conclusions presented in the literature [17]. After calculations, the meridional flow path of the turbine is plotted (Fig. 2) together with the velocity triangles for the stages (Fig. 8).

In the next stage of turbine design, the flow efficiency will increase during the process of optimisation. This process can be performed by means of a few possible algorithms:

- Genetic algorithm [18, 19].
- Implicit filtering [20].
- Nelder-Mead method [21, 22].
- Simulated annealing [23].
- Hooke-Jevees method [24].

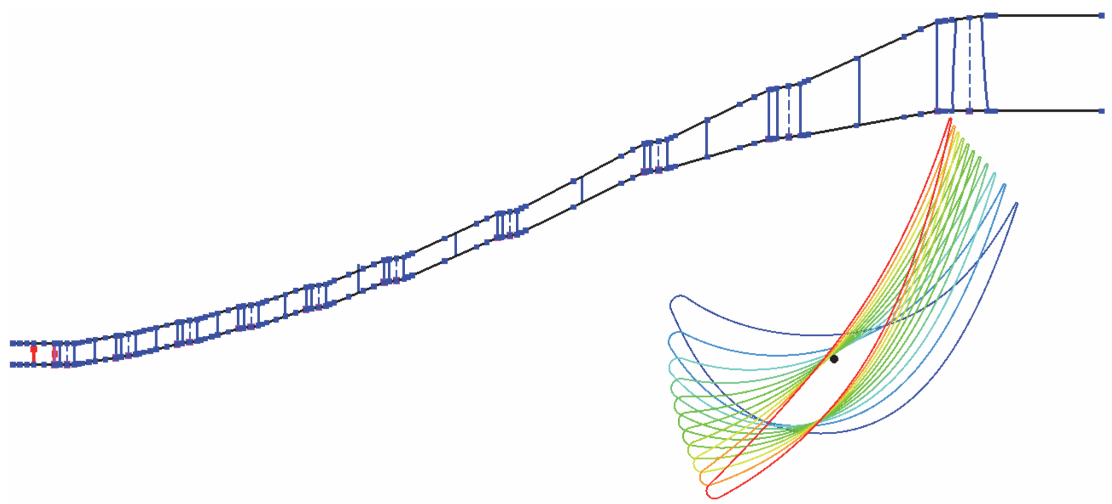
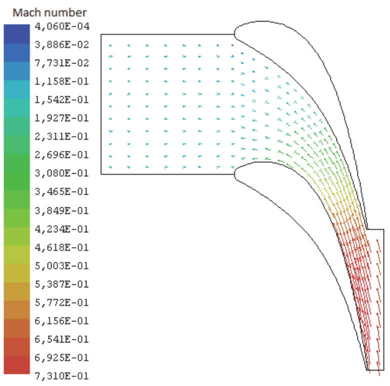


Fig. 5 – View of the meridional section of the turbine flow path and hub-to-tip sections of the last stage rotor



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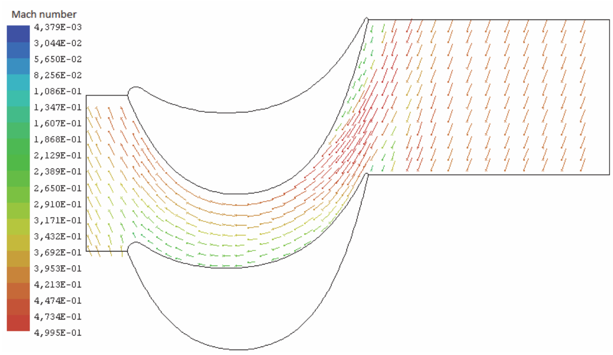

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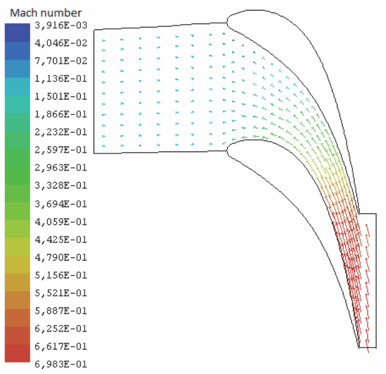

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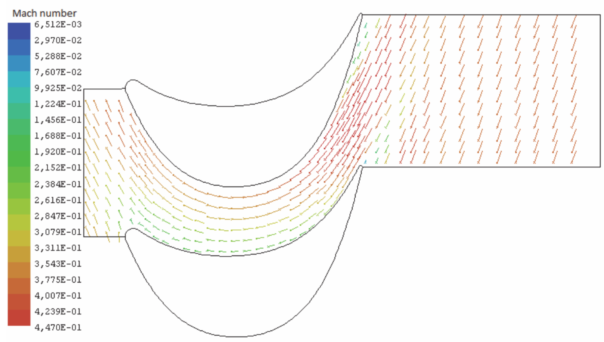

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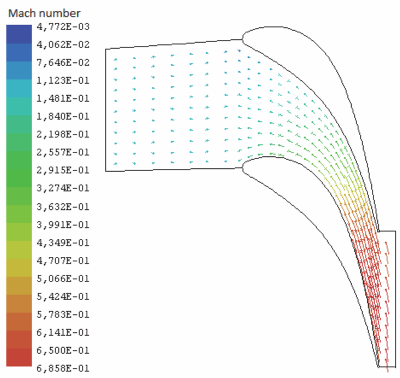

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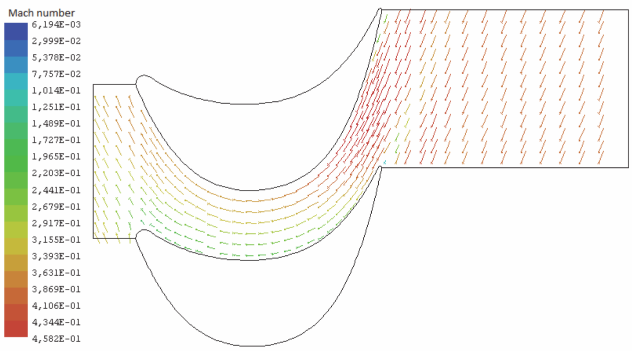

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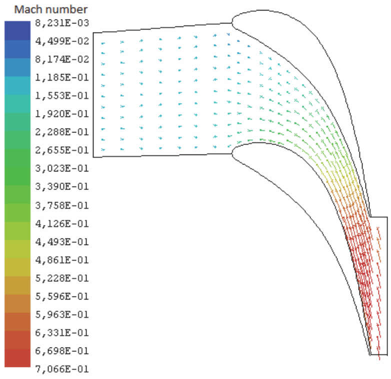

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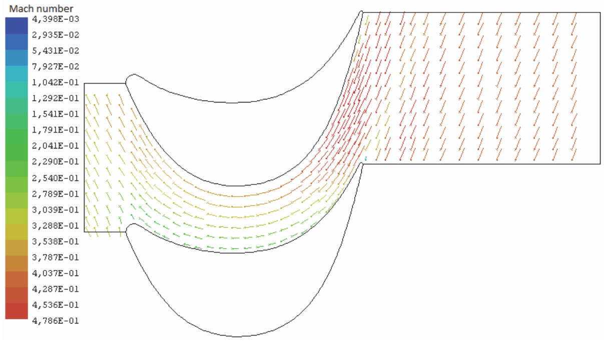

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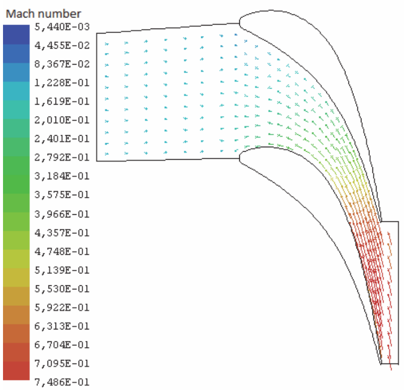

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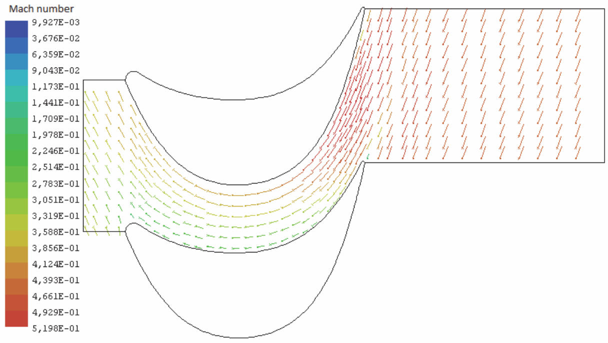

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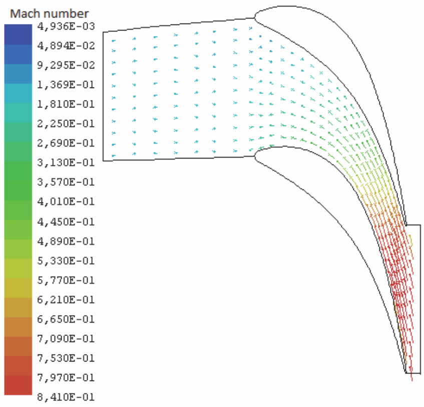

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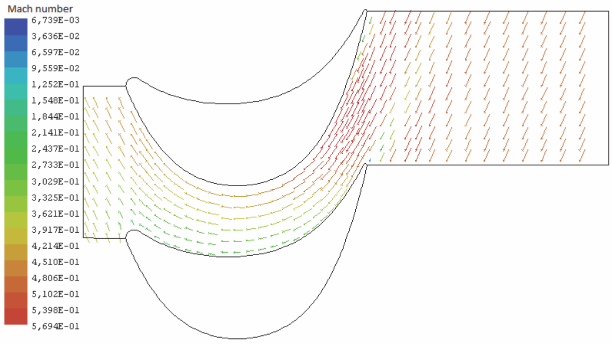

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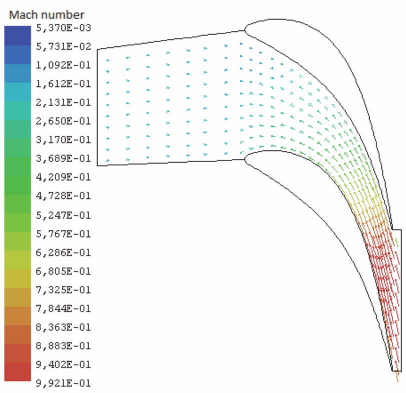

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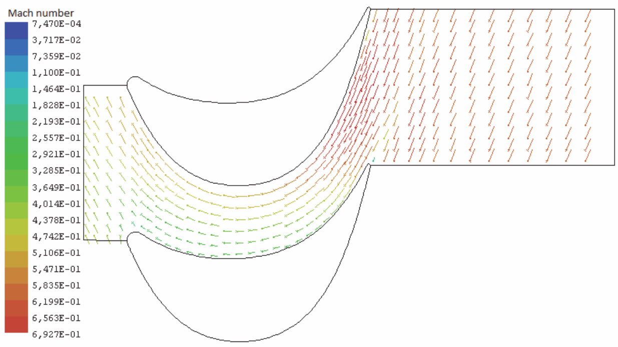

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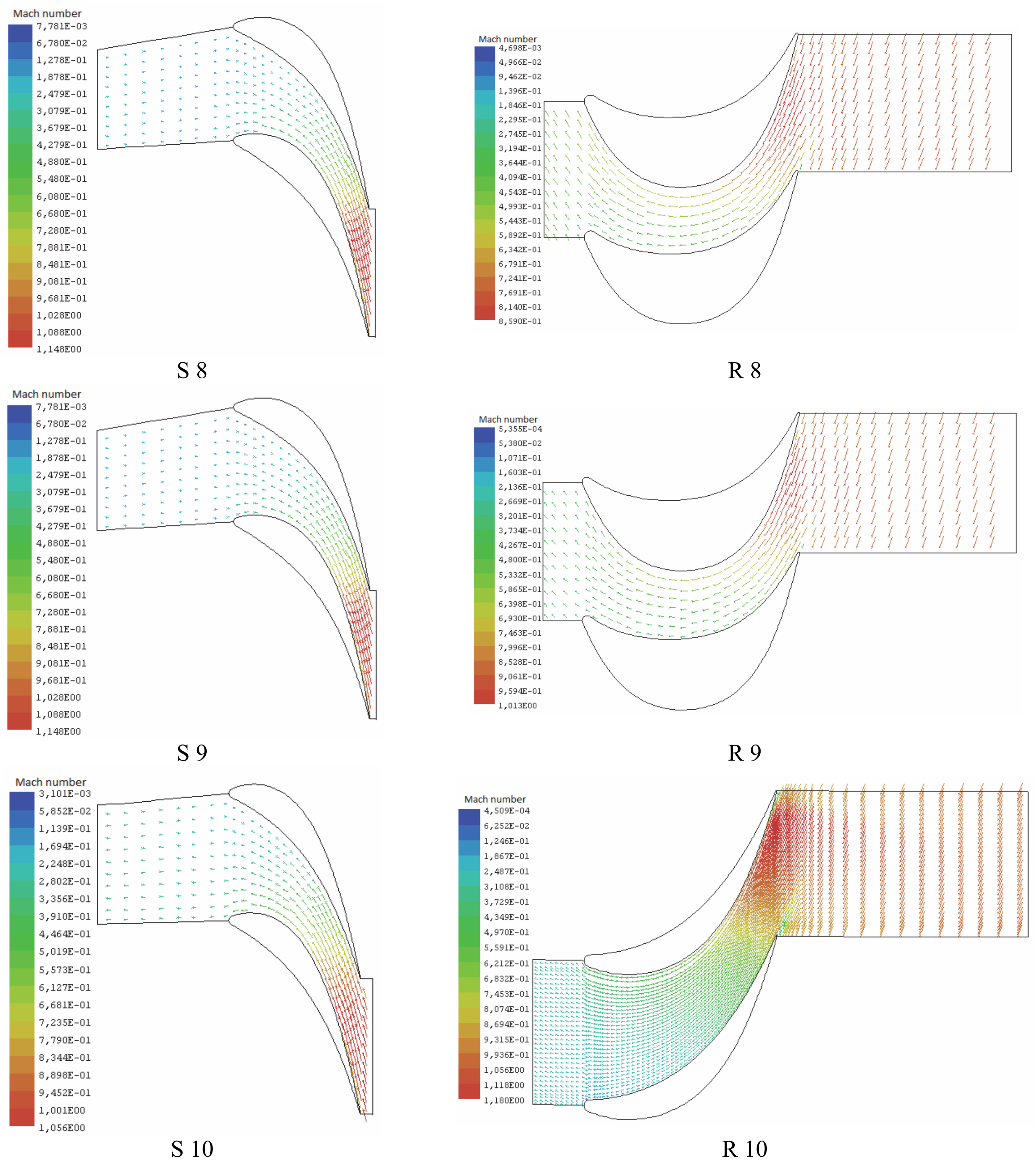

Fig. 6 – Velocity vectors at the mid-span blade-to-blade section

Before this process starts, the optimisation boundaries are defined (isentropic enthalpy drop in stage, degree of reaction, hub diameters, blades length) together with the penalty function and its sensitivity, which contains Mach number, angle α_1 and α_2, degree of reaction and a ratio of the rotor mean diameter to the blade length. Convergent and good performance of the process depends on the appropriate set of these parameters. The optimisation program still deserves deep study. In the future, the hybrid algorithm of Bees Optimisation with the Nelder-Mead method will be implemented, tested and improved [25, 26]. An optimization module is shown in Fig. 9.

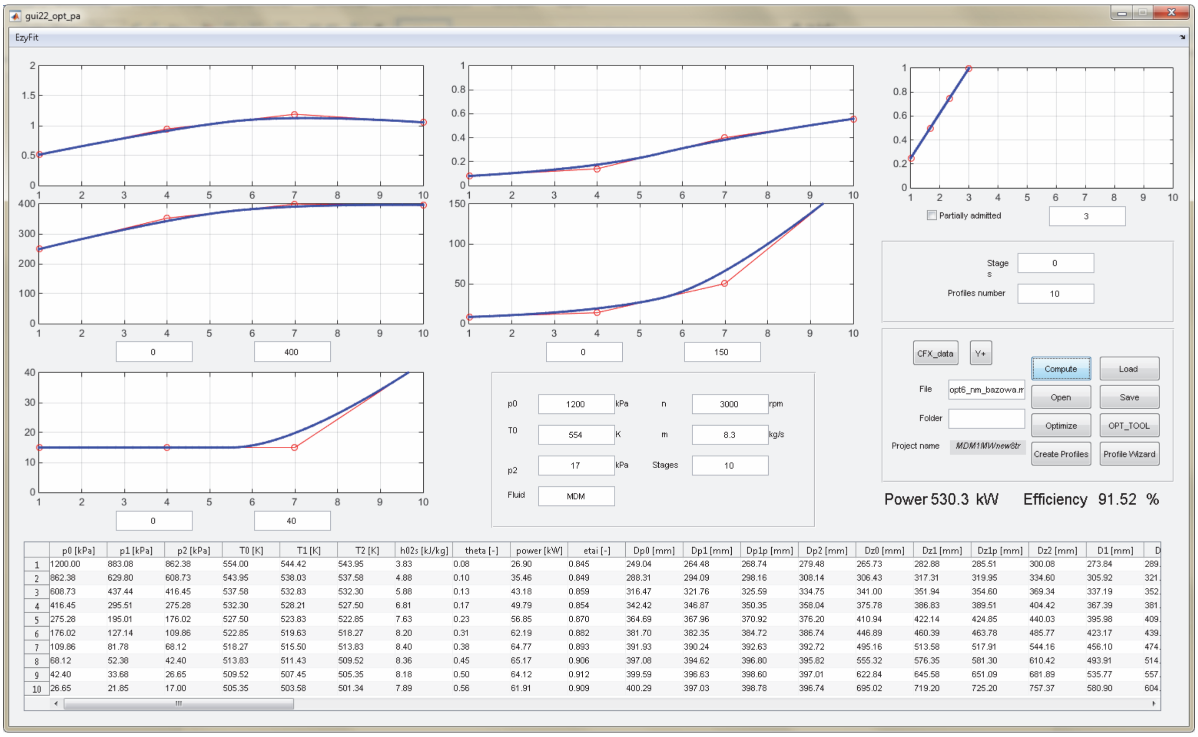

Fig. 7 – The 1D module of the design program

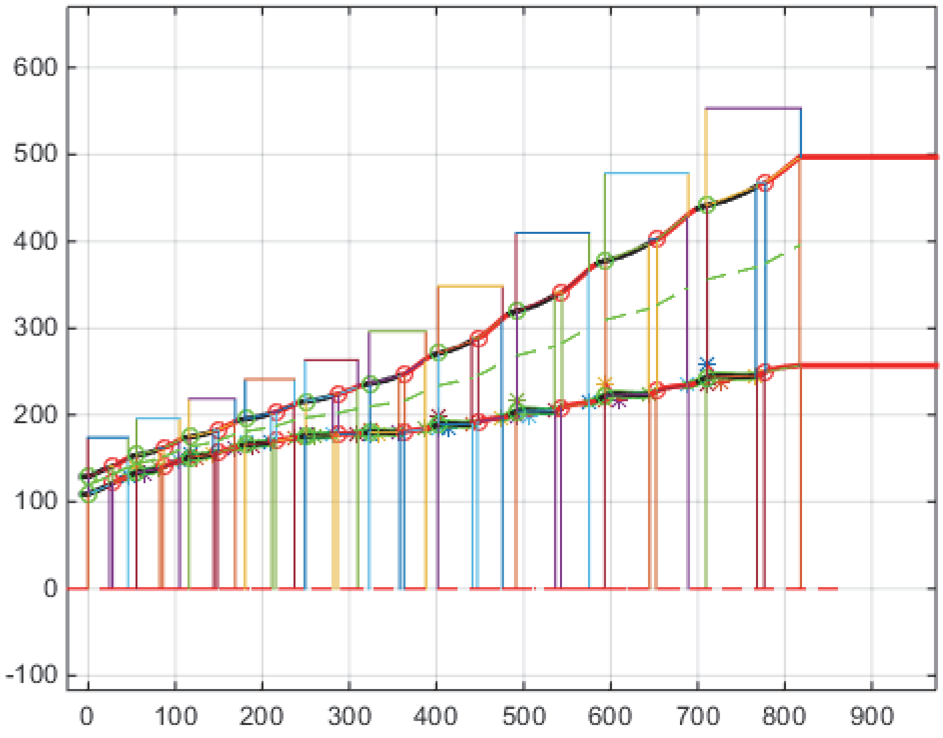

Fig. 8 – An example of the meridional flowpath plotted by the design program

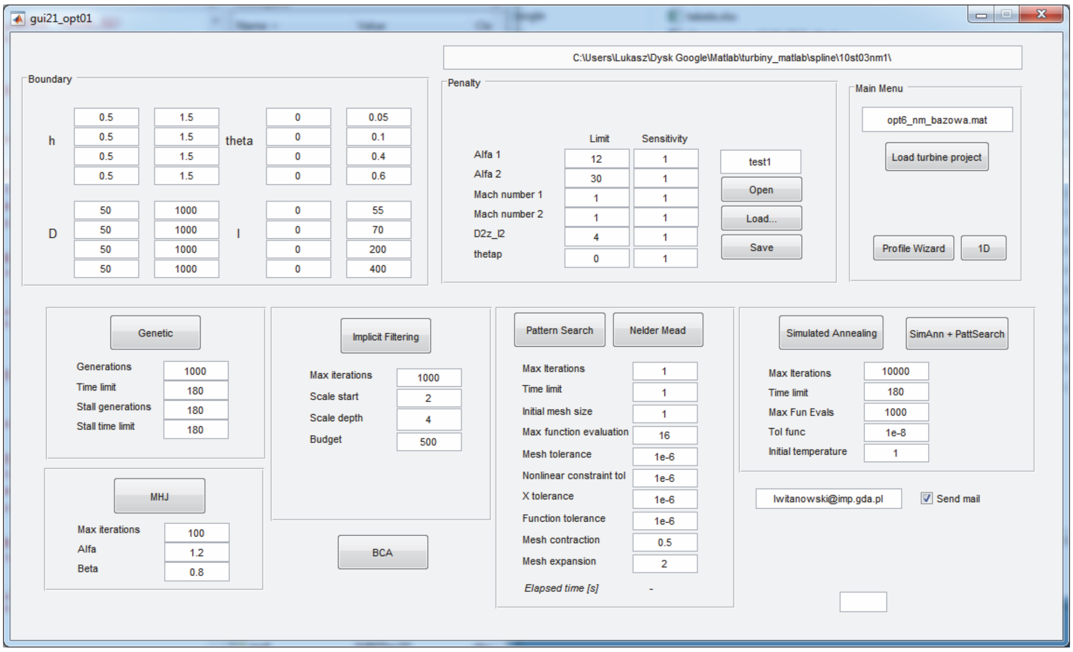

Fig. 9 – Optimization section/module of the design program

The next stage of the design is the creation of the blades geometry. This is made by means of the profile module of the design program. The form of this module is presented in Fig. 10.

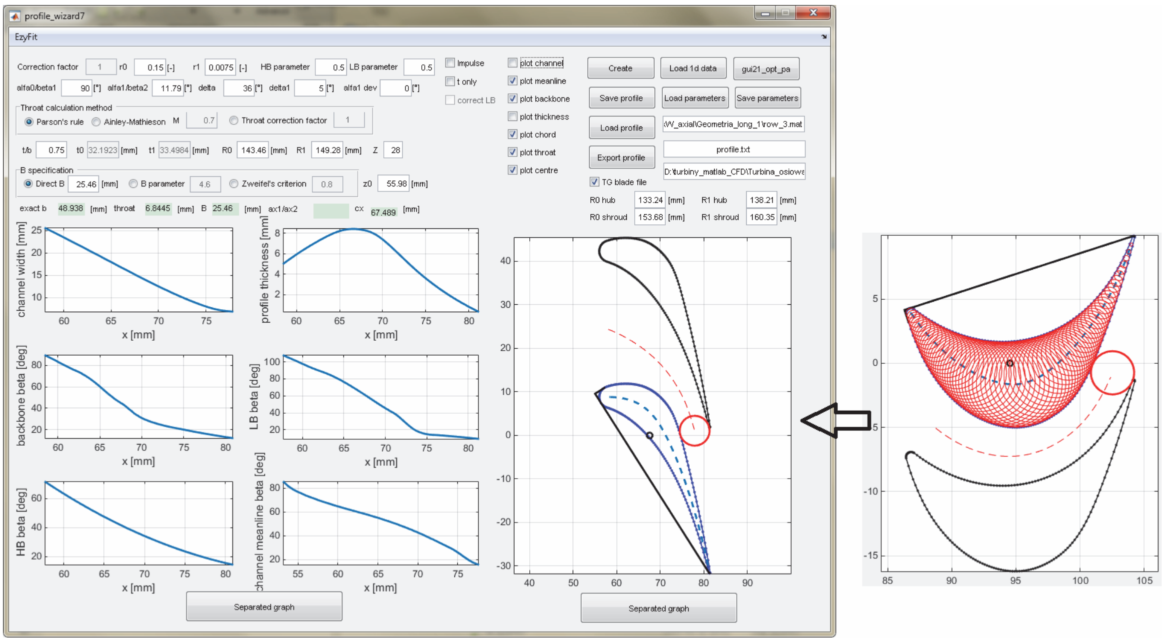

Fig. 10 – The form of the blade geometry module of the design program – example of a stator and rotor

The module acquires values of the angles from 1D calculations and creates the geometry. The pressure side of the profile is a 3rd order b-spline curve, while the suction side consists of two 3rd order b-splines connected at the throat of the blade channel. The leading and trailing edges of the blade are circular arcs. The resultant distributions of the blade angles, blade thickness and channel width are plotted in the form.

In the case of a significant change of the parameters along the blade (small D/l ratio), there is a possibility of designing the twisted blades. The distribution of the angles along the blade is computed by means of the radial equilibrium equation [27],

$$-\frac{c_u^2}{r} + c_r \frac{\partial c_r}{\partial r} + c_z \frac{\partial c_z}{\partial z} = -\frac{1}{\rho}\frac{\partial p}{\partial r} + F_r\,. \tag{6}$$

After neglecting the mass force F_r and radial component c_r, the equation (6) simplifies to the form

$$\rho \frac{c_u^2}{r} = \frac{\partial p}{\partial r}\,; \tag{7}$$

which can eventually be transformed to

$$c_z \frac{\partial c_z}{\partial z} + \frac{c_u}{r}\frac{\partial (r c_u)}{\partial r} = 0\,. \tag{8}$$

The circumferential component obeys the relation

$$c_u r^n = \text{const} \tag{9}$$

and the exponent n is a constant value. Examples of the change of profiles along the blade height is presented in Fig. 11. It is possible to export the geometry to a CAD environment in order to make the technical documentation.

3D flow calculation method

Once the geometry is created, the CFD simulations can be performed. The computational grids are created in the *ANSYS TurboGrid* software [28], which is capable of generating high quality hexahedral meshes of the blade channels. After the domain discretization is performed, the RANS simulations are made by means of *ANSYS CFX* software. The CFX code uses second order space discretization. The three-dimensional compressible flow of the MDM is computed using single stator and rotor channels with the application of periodicity conditions. Between the blade rows a mixing plane is applied, which circumferentially averages the parameters at the interface. For turbulence modelling, the k–ω SST model with automatic wall function is applied. As the boundary conditions, the total pressure and total temperature at the inlet and the static pressure at the outlet are applied. Additionally, 5 % turbulence intensity is applied at the inlet.

Before results of any investigation can be meaningfully discussed or compared, mesh independence study must be performed in order to establish and verify the quality and repeatability of the simulations.

Example of design of the flow path for a cogeneration ORC turbine

The designed ORC turbine is also a 500 kW machine operating on MDM as a working medium. Similar to the previous design, the obtained flow path consists of 10 stages. The preliminary design was obtained from stage-to-stage calculation of the flow part using 1D methods. Final 3D calculations were performed on a grid of 19 mln elements in total. Sample results of the simulations, including Mach number and static pressure contours as well as velocity vectors are shown in Fig. 12 to 15. As seen from the meridional section, endwall contours are slightly different than in the previous design. The flow is regular. The kinetic energy losses in this design are also low and amount to 90.5 % (by the outlet speed) with the leaving energy loss included but the tip leakage loss not included.

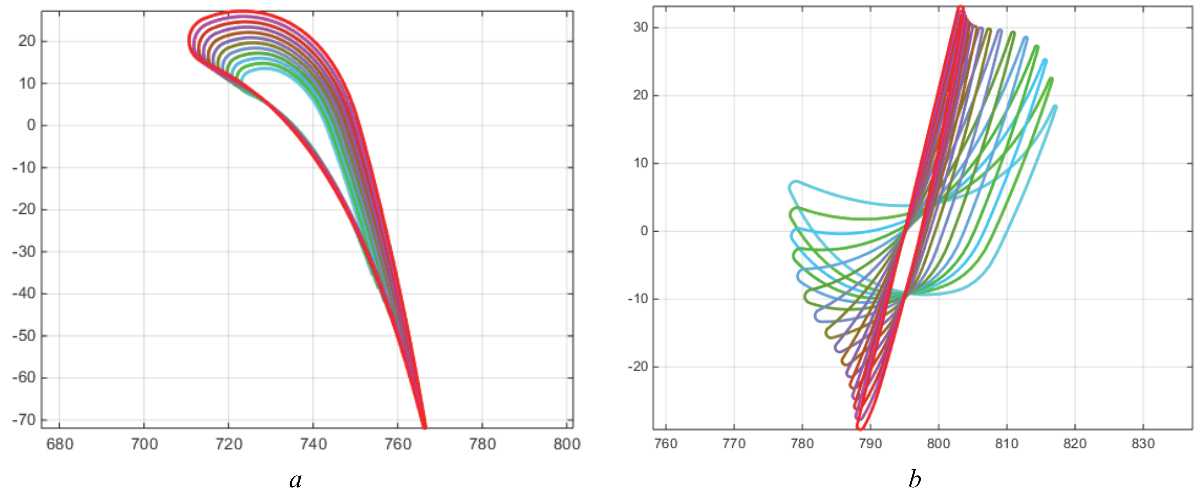

Fig. 11 – The change of the profiles along the blade:
a – example of stator; *b* – example of rotor

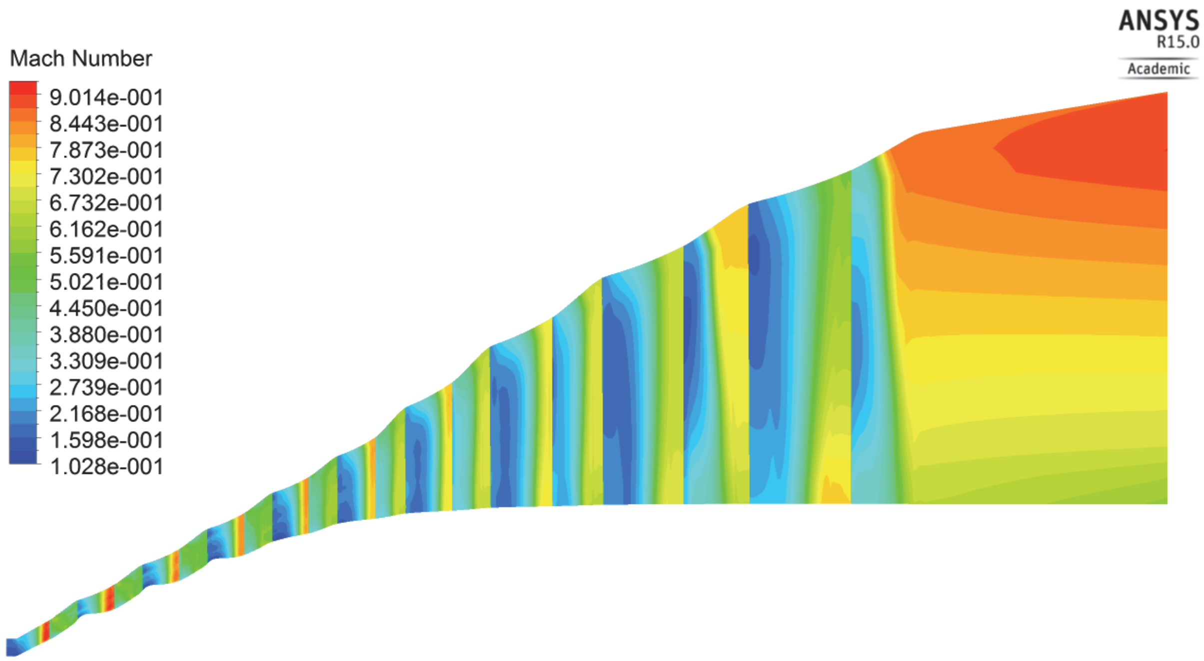

Fig. 12 – Mach number distribution in the turbine (circumerentially averaged)

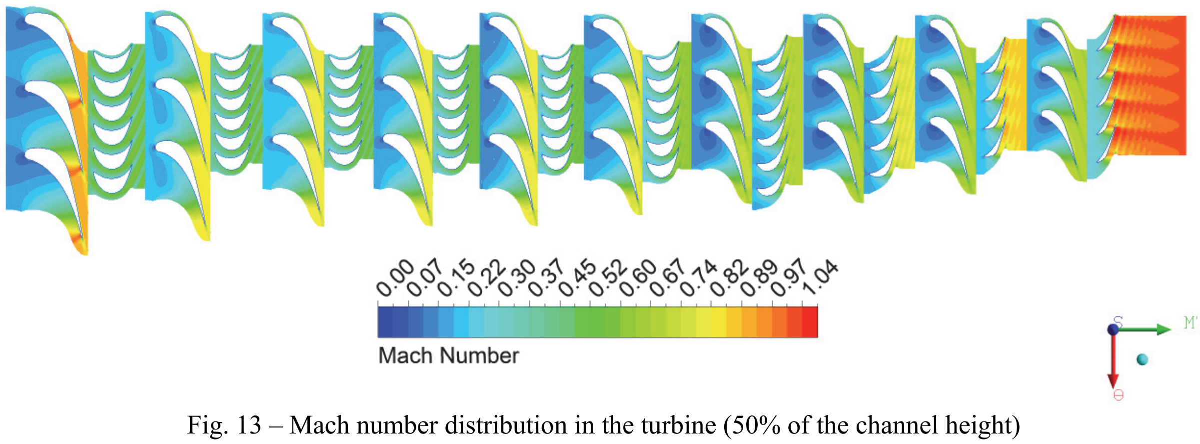

Fig. 13 – Mach number distribution in the turbine (50% of the channel height)

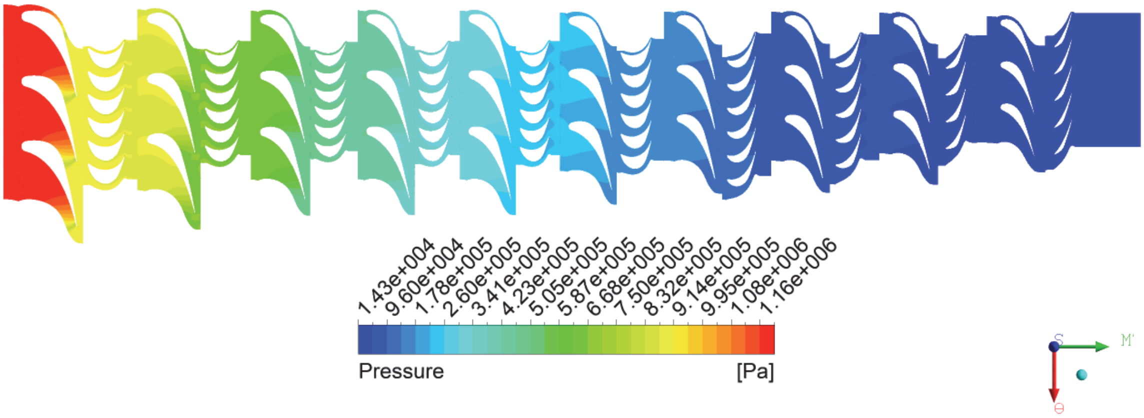

Fig. 14 – Pressure distribution in the turbine (50% of the channel height)

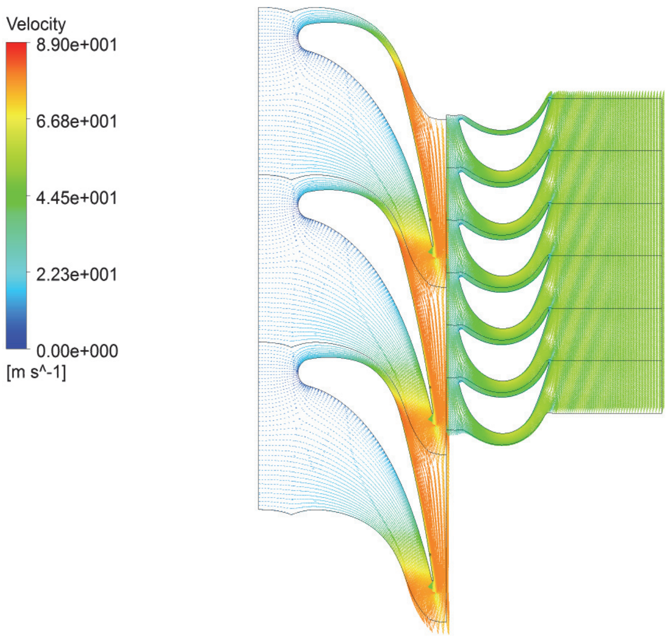

Fig. 15 – Velocity vectors in a stage (50 % of the channel height)

Conclusions

The described design methods enable the elaboration of an axial turbine for an ORC cogeneration unit. The methods draw on mathematical models of various levels of complexity – from 1D to 3D. The 3D turbulent flow model is realised in the software complex *IPMFlow*, which is developed based on the earlier codes *FlowER* and *FlowER-U*, or in software complex *ANSYS*. Two variants of flow path of a 500 kW axial turbine were presented. Both turbine variants exhibit satisfactory flow efficiencies.

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