NUMERICAL AND EXPERIMENTAL INVESTIGATION ON THE ROTARY VANE EXPANDER OPERATION IN MICRO ORC SYSTEM

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ABSTRACT

Volumetric expanders are nowadays used in micro, small and medium power ORC systems. As it was indicated by Bao and Zhao (2013) most often spiral and screw machines are applied. However, it can be seen that the application of rotary vane expanders is also growing (Tchanche et al. (2011)). Rotary vane expanders are particularly interesting because of the many advantages they have. The most important features of such expanders are: very simple construction; high power in relation to the dimensions; the ability to operate in low inlet pressure and wet gas conditions; low weight; lack of clearance volume; lack of steering valves; possibility to construct an oil-free machines; ease of sealing; the ability to operate at low rotational speeds and a low price. As it was described by Gnutek and Kolasiński (2013) power range of the rotary vane expanders is 0.1—7 kW, thus these machines are particularly interesting for micro and domestic ORC systems. Vane expanders used in ORC systems are very similar to these commonly used in pneumatic systems, however, it is necessary to carry out the appropriate adaptation of the machine. This includes special hermetic sealing, lubrication and cooling. As a part of the research works on ORC power systems with vane expanders conducted by the authors on Wrocław University of Technology a research test-stand (comprehensively described by Gnutek and Kolasiński (2013)) was designed and realized. This micro power, R123 based, ORC prototype enables experimental analysis of the vane expander operation under different conditions. In this article authors present the results of numerical simulation of vane expander operation in ORC prototype and compared them with the results of experiment. 3D model of the expander was built and analyzed in ANSYS CFX based on the geometrical data obtained by complete disassembly of the machine. The numerical analysis included the same, as in the case of the experiment, expander operating conditions, i.e. pressure, temperature and R123 flow rate at the inlet and outlet of the expander.

1. INTRODUCTION

As it was indicated by Lund and Münster (2006) effective energy recovery from renewable and waste energy sources is one of the most important present-day problems. Advanced energy systems based on local waste heat, renewables and fossil fuel resources can give the opportunity for an increase of consumers energy safety and continuity of energy supply. However, implementation of local energy systems requires the relevant energy conversion technologies.

One of the promising energy conversion technologies is a ORC system. ORC power systems may differ in power, purpose and technical configuration. Available are: micro power (0.5—10 kW), small power (10—100 kW), medium power (100—500 kW) and a large power (500 kW and more) systems. As it was presented by Vanslambrouck (2009) they can operate as power plants, CHP's and multi-generation systems.

The most important problems connected with ORC system design are the suitable working fluid and expander selection. Expander selection is mainly based on the system power and its purpose. In

general, two types of expanders can be applied in ORC systems. One are the turbines, the others are volumetric machines.
Turbines are mainly applied in an large and medium power ORC systems powered by the heat sources with high thermal capacity (1 MW and more) and temperature (150 °C and more), such as large industrial waste sources e.g. steam boilers or gas turbine exhaust gases. Volumetric expanders are applied mainly in micro and small power systems such as domestic and agriculture plants. One of the most important problems in this case is the dynamic thermal characteristic of the heat source also characterized by small capacity, thermal power and temperature (up to 150 °C). Variation in the heat source properties has a negative influence on the continuity of the system operation and difficulty in system adjustment. Therefore the design and construction of small and micro power ORC systems is very difficult and most of the existing systems are still at the level of prototype or under research. In low and micro power ORCs applicability of turbines is very limited due to the machine operational characteristics requiring large flow of the working medium. Moreover microturbines are very small in dimensions what results in very high rotational speeds and difficulty of a rotor balancing and bearing. The necessity of a very precise parts fitting result in high manufacturing costs. This is the contrary to the aim of small ORC systems which should be simple, cheap and easy to use.
Volumetric expanders are a good option for systems where the low pressures and low working medium flows are expected. In general piston, screw, spiral, vane and the rotary lobe expanders can be applied in small and micro ORC plants. The general advantages of volumetric expanders resulting from comparison with the turbines were comprehensively described by Gnutek and Kolasiński (2013). Vane expanders are especially interesting for small and micro ORC systems. Currently they became a subject of different experimental and numerical scientific analyses (Montenegro et al. (2014)). Micro vane expander has a number of advantages resulting from comparison with other volumetric expanders. The most important are: very simple design, high power in relation to the dimensions, suitability for wet gas conditions, low weight, ease of gas-tight sealing and very low price. There are no ORC-dedicated vane expanders available, however, standard pneumatic air motors can be easily adapted.
As a part of the research works on ORC power systems with vane expanders, conducted by the authors on Wrocław University of Technology, a research test-stand (comprehensively described by Gnutek and Kolasiński (2013)) was designed and realized. This micro power, R123 based, ORC prototype enables experimental analysis of the vane expander operation under different conditions.
The authors decided to carry out the experiment on the test-stand and to compare the results with numerical analysis of the applied vane expander. 3D model of the expander was built and analyzed in ANSYS CFX based on the geometrical data obtained by complete disassembly of the machine. The numerical analysis included the same, as in the case of the experiment, expander operating conditions i.e. pressure, temperature and R123 flow rate at the inlet and outlet of the expander.

2. DESCRIPTION OF THE TEST-STAND, THE EXPANDER AND THE EXPERIMENTAL RESULTS

An experimental test-stand was designed and realized in order to study the influence of the different operational conditions on the operation of the rotary vane expander. Figure 1 shows a simplified construction scheme of the test-stand. The main test-stand components are: the gas central heating boiler (featuring maximal thermal power of 24 kW) (1), the shell-and-tube evaporator (2), the working fluid pump (3), the reservoir of working fluid (4), the plate condenser (5) and the micro multivane expander connected with DC generator (6). The working fluid is R123. The test stand is based on manual control of operational parameters with the help of regulation valves. The manual control helps in simulating different operational conditions e.g. it is possible to change the working medium flow direction in the evaporator from a counter-flow to the co-current. The measurements are carried out using the following methods: temperatures are measured with the use of T-type thermocouples, pressures are measured with the use of tube pressure gauges. The flow rate of R123 as well as the flow rate of cooling and heating water are measured with the use of rotameters (See Fig. 1 for the measurement sensors locations: p – manometer, t – thermocouple, V – rotameter).

The heat source for the system is hot water from the gas central heating boiler (1). The temperature of the heat source can be regulated in the range of 40—85 °C. This allows the evaluation of operational conditions of the ORC power system for variable heat sources.

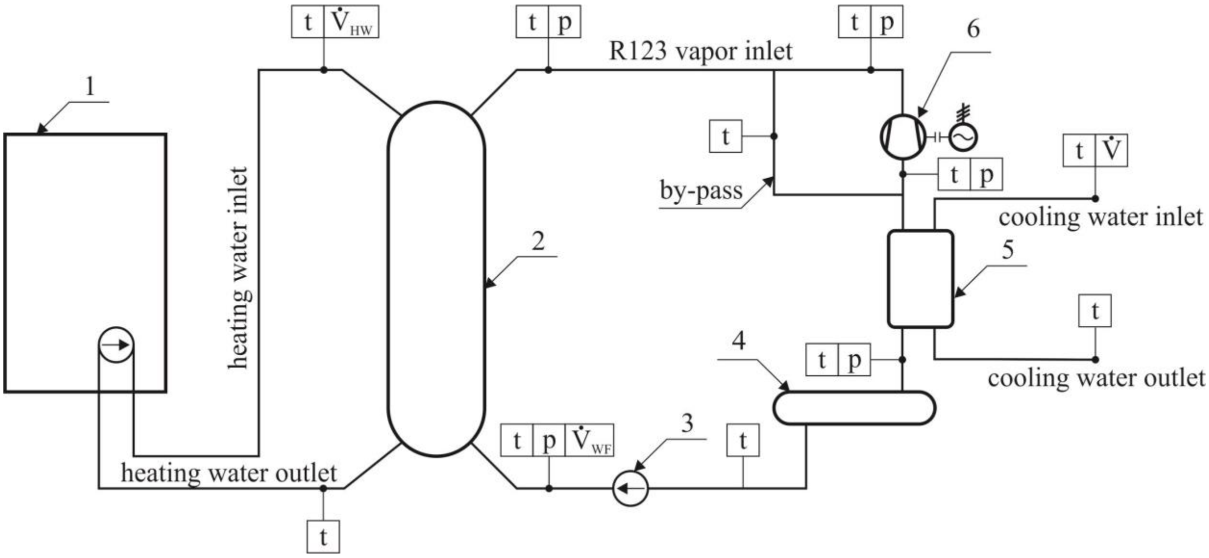

Figure 1: The simplified construction scheme of the test-stand.
1 – gas central heating boiler, 2 – shell-and-tube evaporator, 3 – working fluid pump,
4 – reservoir of working fluid, 5 – plate condenser, 6 – multivane expander with DC generator

A more detailed description of the test stand was presented by Gnutek and Kolasiński (2013). The expansion device is a micro four-vane air motor featuring a maximum power of 600 W. The expander was specially adapted for low-boiling working fluid e.g. the special seals and bearings were used. Moreover, a number of changes in the expander design were made in order to maximize the machine power. The expander is connected by gas-tight clutch with a small DC generator. Figure 2 shows a view of the expander-generator unit disassembled from the test-stand.

Figure 2: The view of the expander-generator unit disassembled from the test-stand

The design of the above described rotary vane expander is presented in figure 3. It consists of cylindrical stationary cylinder of inner diameter 37.5 mm and rotating rotor of outer diameter 34.0 mm. The rotor is placed eccentrically relative to the cylinder. Eccentricity is 1.75 mm. Length of the cylinder is 22.0 mm. Rotor is equipped with four flat vanes, which can move in the slots due to centrifugal forces. Vanes are positioned to the cylinder surface in the right angles. Thickness of each vane is 1.5 mm. Rotary vane expander is fed with R123 working fluid via cylindrical pipe of inner diameter 8.5 mm. Pipe is tangentially located at side surface of the cylinder. On the second side there is identical outlet pipe where gas leaves the expander chamber to the condenser. The operation principle of rotary vane expander is as follows: high pressure at inlet acts on the vanes and results in rotary motion of the rotor. Due to eccentricity and moving blades, working chamber constantly changes its volume from lowest to highest. The working fluid is trapped in chamber which volume increases with increase of angle of rotation. Consequently working fluid expands and exits through the outlet to the condenser. Rotational energy of the rotor is transformed into mechanical work which is the main output of the rotary vane expander.

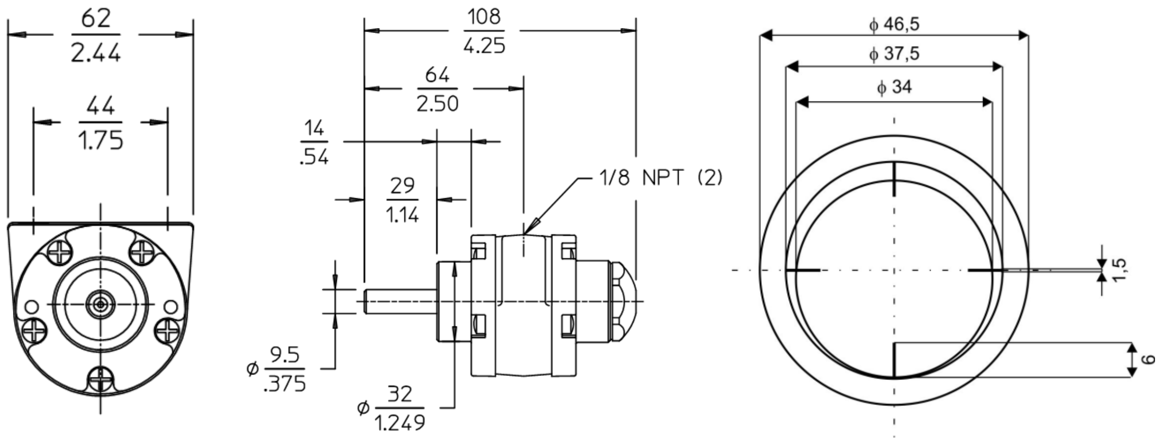

Figure 3: Dimensions of the rotary vane expander being considered.

In order to determine the working fluid thermal properties at the inlet and the outlet of the expander the experiment was carried out on the test-stand. The authors decided to carry out this experiment for the maximum temperature of the heat source (t = 85 °C) and maximum working medium pressure allowed for this expander (p = 0.5 MPa). For this conditions the measured thermal properties of the working fluid at the inlet to the expander were p_{in} = 0.5 MPa and t_{in} = 80 °C. The measured outlet temperature was t_{out} = 65 °C and the outlet pressure was p_{out} = 0.1 MPa. The measurement results together with the corresponding values of specific enthalpy and the specific entropy (determined with the SOLKANE software) are presented in table 1.

Table 1: The experimental results

$\dot{m}_{R123}$	t_{in}	p_{in}	t_{out}	p_{out}	h_{in}	s_{in}	h_{out}	s_{out}
kg/s	**°C**	**MPa**	**°C**	**MPa**	**kJ/kg**	**kJ/kgK**	**kJ/kg**	**kJ/kgK**
0.0615	81	0.5	65	0.1	429.48	1.6788	424.88	1.7472

Presented above experimental results are the input data to the described below numerical analysis.

3. NUMERICAL MODELING

3.1 Numerical domain

In the figure 4 three dimensional numerical domain and mesh are depicted. Additionally the mesh details near the tip of the vanes and inlet/outlet areas are also visible. Numerical domain consists of fluid regions only and housing walls of the rotary vane expander are modelled via boundary conditions. Additionally numerical domain was divided into two areas: stationary inlet and outlet pipes and rotating part including fluid residing in the working chambers of the rotary vane expander. Dimensions of the numerical domain are the same as described in the previous section. The inlet and outlet are in the distance 45 mm from the Y axis of the rotary vane expander. Inner cylinder surface is represented by the CYLINDER surface and outer surface of the rotor by ROTOR surface. Due to small differences between rotor and cylinder diameter it was hard to provide good quality mesh in the case of vane thickness 1.5 mm. In order to overcome this difficulties vane thickness in the model was set up to 0.5 mm. This simplification does not introduce significant error during the calculation because impact of vane thickness on heat transfer and fluid flow is negligible. Additionally it was impossible to use gap value between vane tip and cylinder wall less than 0.15 mm due to strong curvature of the geometry. Therefore for each time step during the simulation gap was hold to 0.3 mm for each vane to avoid creation of negative volumes in the numerical mesh. It results in eccentricity 1.45 mm whereas in the real rotary vane expander it equals to 1.75 mm. Nevertheless, in the reality small gaps always exists due to sealing rings and small gaps should be taken into account in order to model leakage phenomenon (Montenegro et al., 2014). In the present study hexahedral mesh was used for rotor domain with 9 control volumes in the gap and number of hexahedrons equal to 75492. For the inlet and outlet pipes tetrahedral mesh was used with total number of tetrahedrons equal to 109383.

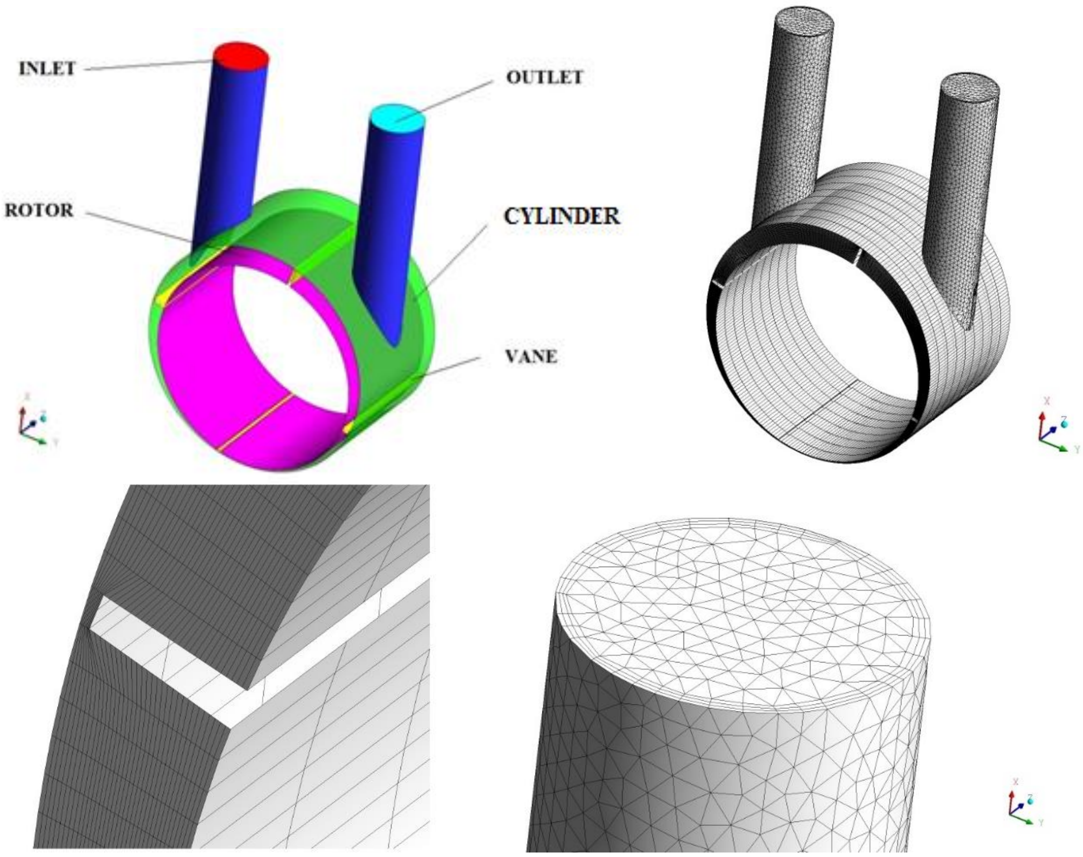

Figure 4: Three dimensional numerical domain of the rotary vane expander and numerical mesh. At the bottom mesh details near the vane tip and inlet/outlet areas

3.2 Numerical model

Non-isothermal, incompressible turbulent flow in the rotary vane expander has been considered. Transport equations of mass, momentum and energy were solved with the use of commercial solver (Ansys, 2014). In the absence of a phase change, radiation and with neglecting dissipation function term these equations can be written in the following vector form

Continuity equation

$$\nabla \cdot \mathbf{U} = 0 \tag{1}$$

Momentum equation

$$\frac{\partial \rho \mathbf{U}}{\partial \tau} + \nabla \cdot (\rho \mathbf{U}\mathbf{U}) = -\nabla p + \nabla \cdot \left[\mu \left(\nabla \mathbf{U} + \nabla \mathbf{U}^T \right) \right] \tag{2}$$

Energy equation

$$\rho c_p \left[\frac{\partial T}{\partial \tau} + \nabla \cdot (T\mathbf{U}) \right] = \nabla \cdot (k \nabla T) \tag{3}$$

where **U** stands for total velocity vector, i.e. difference between velocity vector and moving mesh velocity vector. It is due to the incorporated deforming mesh method (Ansys, 2014) in order to take into account movement of the rotor and vanes. Deforming mesh method consists in calculating in each time step the shape of the moving boundaries. Additionally the nodes distribution in the mesh is calculated according to the following diffusion equation

$$\nabla \cdot \left(\Gamma_{disp} \nabla \delta \right) = 0 \tag{4}$$

where δ is the displacement relative to the previous mesh locations and Γ_{disp} is the mesh stiffness, which determines the degree to which regions of nodes move together (Ansys, 2014). It is so-called displacement diffusion model and preserves the relative mesh distribution of the initial mesh. Deforming mesh method requires additional computational resources, but it provides most accurate results for issues with moving parts. Due to simple geometry of the rotary vane expander for each of the boundary surface equations of motion were specified via CEL expressions (CFX Expression Language). They are basically code routines written in Fortran programming language and can be easily incorporated in the solver. The deforming mesh method is inherently transient because shape of the numerical domain changes constantly and has to be determined for each time step. In the present study rotor moves at constant rotational speed n = 3000 rev/min. The time step used in the simulations corresponds to the rotation of the rotor about 0.1 degree. Very high rotational speed results in fully turbulent flow. In order to turbulent flow standard k-ε model was used. Both convection, temporal and turbulent terms in the transport equations were solved with use of high resolution scheme. For each wall no-slip boundary condition was applied. The vane and the rotor surfaces were treated as adiabatic. On the cylinder and side walls convective boundary condition was imposed with heat transfer coefficient h = 5 W/(m^{2}K) and ambient temperature t_{amb} = 20° C. Working fluid enters the inlet pipe under 0.5 MPa pressure and in temperature 81 °C and exits through outlet in temperature 65 °C. The outlet pressure is equal to 0.1 MPa. In order to provide information transfer between stationary and rotating subdomains, interface boundary condition was used. The working fluid used in the simulation was R123. It was treated as Newtonian with having constant thermo-physical properties determined from the Refprop software for the mean temperature 73 °C and presented in table 2. Calculations were conducted up to five full rotor revolutions and then periodic steady state was assumed. It was reasonable due to small differences between successive fourth and fifth rotation. In terms of residuals, within each timestep calculations were conducted to achieve convergence below 10^{-6}.

Table 2: Thermo-physical properties of R123 used in the simulation

ρ	**c_p**	**μ**	**k**	**Pr**
kg/m^3	**J/(kgK)**	**μPa·s**	**W/(mK)**	**-**
10.00	750.00	12.388	0.01201	0.77

4. NUMERICAL RESULTS AND DISSCUSION

Figure 5 shows the pressure distribution in the working chambers during rotor movement. As it can be seen from this figure the highest pressure of the working fluid is in the inlet port and filled working chamber. During the rotor movement working fluid pressure decreases with the decreasing volume of the working chamber. The lowest pressure is in the outlet port during the evacuation of the working fluid from the machine. Figure 6 shows the working fluid velocity vectors during the rotor movement. The highest velocities can be observed in the 2nd and the 3rd working chamber, where the chamber volume is minimal. Relatively low working fluid velocities can be observed in the inlet and the outlet ports. Also, in this areas, vortices can be observed. These vortices have negative influence on the expander operation, as they result in working fluid pressure changes and mixing, thus the edges of the inlet and the outlet port of the expander should be redesigned and optimized. Figure 7 shows the temperature distribution in the working fluid during the expander operation. As it can be seen in this figure the working fluid temperature decreases during the expansion. In the inlet and the outlet port the local temperature fluctuations are appearing. These fluctuations are resulting from both the leakage between the vane and the cylinder (the hot gas flowing into the 1st expander working chamber mixes with the gas in the 4th expander working chamber and the gas flowing through the outlet port) and vortices appearing in these areas (increased velocity of the fluid and change in the fluid internal energy).

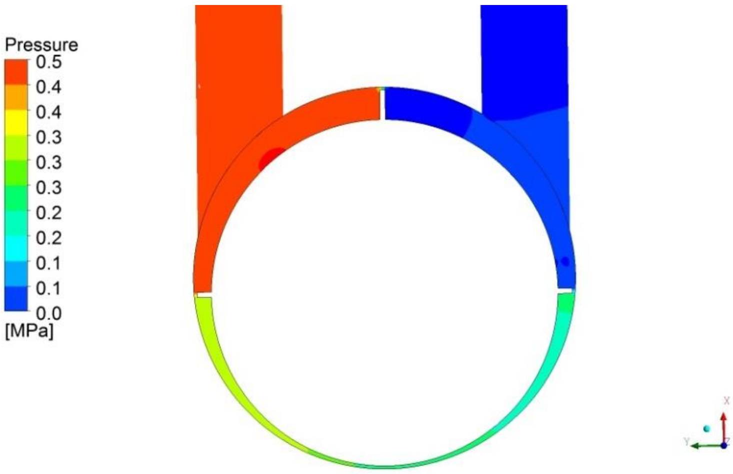

Figure 5: Pressure distribution in the plane for z = 0.011 m

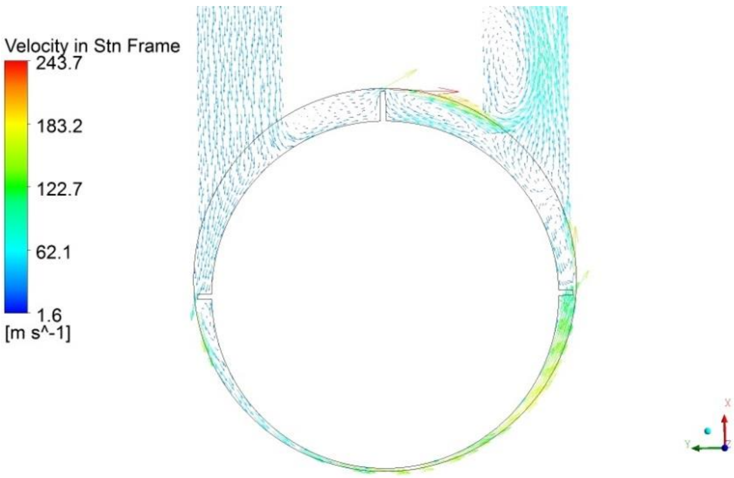

Figure 6: Vectors of velocity in the stationary frame of reference in the plane for z = 0.011 m

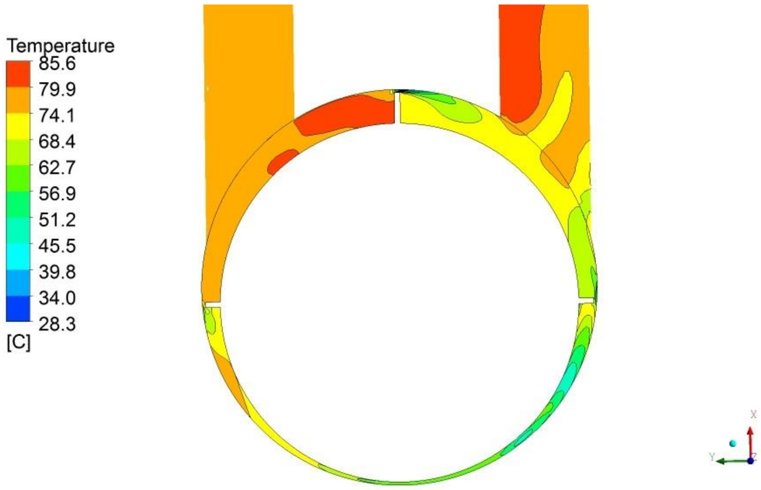

Figure 7: Temperature distribution in the plane for z = 0.011 m

5. SUMMARY AND CONCLUSIONS

In this study the numerical and experimental analysis of the micro-power rotary vane expander operation in ORC system were presented. Numerical analysis was based on 3D model of the expander which was built basing on the geometrical data obtained by complete disassembly of the experimentally tested machine. Numerical analysis of the expander operation was performed in ANSYS CFX, basing on the thermodynamic parameters measured on the test-stand.
The results of the analysis showed the distributions of the pressure, velocity vectors and the temperature of R123 in the expander working chambers. The calculated results show that in the case of volumetric machines the working fluid velocities inside the working chambers are low when compared to the turbines. Moreover, the velocity vectors distribution indicated that in the inlet and the outlet port of the expander large vortices are appearing. These vortices have negative influence on the expander operation, and should be optimized by the change in the expander design. Thus, the sharp edges of the inlet and the outlet port should be redesigned. One of the possible solutions is the change of the inlet and the outlet port diameter to the larger and rounding the edges. Also, the optimization of the expander design should include the change of the rotor diameter to the lower and the length of the cylinder to the larger, in order to increase the volume of the working chambers. Increased volume of the working chambers will result in the increase of the expander power. The authors are currently working on further numerical and experimental analyses concerning the optimization of the volumetric expanders design.

NOMENCLATURE

The nomenclature should be located at the end of the text using the following format:

δ	relative displacement	(–)
Γ	mesh stiffness	(–)
μ	dynamic viscosity	(μPa·s)
ρ	density	(kg/m^3)
τ	time	(s)
c	specific heat	(J/kgK)
h	specific enthalpy	(kJ/kg)
h	heat transfer coefficient	(W/m^{2}K)
k	thermal conductivity	(W/mK)
$\dot{m}$	mass flow	(kg/s)
p	pressure	(Pa)

Pr	Prandtl number	(–)
s	specific entropy	(kJ/kgK)
T	temperature	(K)
t	temperature	(°C)
U	total velocity vector	(–)

Subscript

amb	ambient
in	inlet
out	outlet
p	related to the isobaric process
R123	related to the R123 working fluid

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