ANALYSIS OF HEAT TRANSFER WITHIN ROTARY VANE EXPANDER

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ABSTRACT

Thanks to their positive features, i.e., simple design with a small number of moving parts, easy manufacturing, and low production and purchase costs, multi-vane expanders are promising for application in micro-power organic Rankine cycle systems. These expanders are simple in design, small in dimensions, inexpensive, and feature low gas flow capacity and expansion ratio. The processes of heat exchange inside the working chambers of the multi-vane machine have a significant impact on its operation and efficiency. Therefore, they are undoubtedly important from a scientific and practical point of view. The literature review shows that to date the processes of heat exchange taking place in multi-vane machines have not been analyzed in detail, therefore there is a lack of experimental and modelling results related to these processes. Although some analytical models related to these phenomena were formulated, there is a lack of comprehensive model analysis. The heat transfer process in a multi-vane machine is also significantly affected by the phenomenon of the boundary layer scraping by a vane. The result shows that this condition leads to disturbances in the fluid velocity profile in the gap between the vane and the cylinder. In this paper, the discussion of heat transfer models and results of experimental research and modelling on the heat transfer in multi-vane expander are reported. Experimental results show that the heat transfer direction changes during the machine operation. Modelling was focused on determining the heat transfer coefficients in the expander under various flow conditions. The comparison of the modelling results suggests that applicability of models available in the literature and applied for modelling the heat transfer in other machines and devices with rotating vanes (e.g., scraped surface heat exchangers) is limited in the case of estimating the heat transfer processes in multi-vane machines.

1 INTRODUCTION

Expander is a key subassembly of the ORC system. By the type of the applied expander, ORCs are classified into two categories: turbine-based systems and systems in which volumetric expanders are applied. Turbines are mainly applied in high-power ORC systems, while smaller ORCs are often utilizing volumetric expanders. Features of these two types of the expanders are different. Compared to turbines, volumetric expanders feature a lower range of operating pressure and lower working fluid flows. What is more, the lower cycle frequency, lower rotational speed, the possibility of obtaining larger expansion ratios in one stage, and ease of hermetic sealing are their positive attributes. Piston, screw, scroll, multi-vane, and rolling piston expanders can potentially be applied in ORCs. In most of the cases, these expanders are at the level of lab prototypes, or are under research. In some of the cases (like, e.g., screw and scroll expanders), the design of the expander is complicated, and the manufacturing process is difficult. Some screw and scroll expanders are made in oil-free versions and can expand wet gas. Compared to scroll and screw expanders, piston expanders have a simpler design. On the other hand, it requires lubrication and valve timing.
Recently, an increased interest is observed in multi-vane expanders designed for application in modern domestic ORC (organic Rankine cycle) systems of low or micro power (ranging from 1 to 3 kW_e). Multi-vane expanders, have a very simple design, a promising ratio of machine power to external dimensions, and their costs are low. If special construction materials are applied, it is possible to

eliminate the need for lubrication. The multi-vane expander features lower vapor consumption and a smaller range of operating pressures compared to the other types of volumetric machines and turbines and can be easily hermetically sealed, what is more, this type of expander can operate in moist gas conditions. Multi-vane expanders feature power ranging from a few hundred W up to 10 kW and shaft rotational speeds of 1000–4000 rpm, while the maximum value of the pressure at the inlet to the multi-vane expander is about 10 bar (Kolasiński, 2019). Research on the use of multi-vane expanders in ORC systems was started in the 70s and are continued today in many scientific units worldwide. In the 80s of the 20th century, Badr *et al*. (1984) from Cranfield University investigated a comprehensive experimental and theoretical analysis on multi-vane expanders. In the 90s of the 20th centuries, Gnutek and Kalinowski (1994) from the Wrocław University of Technology carried out research on the possible application of a multi-vane expander in the ORC system. As a part of the above-described research, the test-stands were developed, experimental data were collected, and a number of mathematical models describing the operation of the machine (i.e., working chambers volume change, variability of forces and friction). Thanks to the possibilities given by computer simulation, it is possible to provide accurate simulations of the multi-vane expander operation and provide guidelines for their optimization. For example, these issues were treated in (Rak *et al*., 2018).
To date, the processes of heat exchange taking place in these machines have not been analyzed in detail, therefore there is a lack of experimental and modelling results related to these processes. Although some analytical models related to these phenomena were proposed (these models are described in detail later in this paper), there is still a lack of comprehensive model analysis. The processes of heat exchange inside and outside the machine have a significant impact on its operation and efficiency, which is why they are undoubtedly important from a scientific and practical point of view. These phenomena are very complex because, in the multi-vane expander working chamber, various thermodynamic and hydrodynamic processes occur simultaneously (e.g., circulation of the working fluid in the chamber, flow impact on the surface of machine parts or leakage between adjacent working chambers (through the gaps between the top of the vanes and the cylinder, and through other leaks). The heat transfer process in a multi-vane machine is also significantly affected by the phenomenon of the boundary layer scraping by a vane (Gnutek, 1997). In a result, this leads to disturbances in the velocity profile of the fluid in the gap between the vane and the cylinder.

2 HEAT EXCHANGE PROCESSES IN A MULTI-VANE EXPANDER

During the operation of a multi-vane expander, different physical phenomena are proceeding simultaneously in the working chambers. The important phenomena are working fluid flow through the working chambers, working fluid circulation in working chambers and its impact on machine parts, internal and external leakage of the working fluid and friction. The working fluid temperature is changing during the processes (depending on the character of the machine operation, i.e., temperature decrease in the case of cooled compressors and expanders expanding hot gases or temperature increase in the case of expanders working with cold gases). Therefore, in some of the cases, a cooling (or heating) the machine by means of cooling (or heating) fluid is needed. The above-described processes are influencing the conditions of the heat transfer in the device. Therefore, the following heat transfer rates within the machine can be listed: Q_{g1} - heat transfer rate delivered to the working fluid by heating fluid; Q_{g2} - heat transfer rate taken from the working fluid by cooling fluid; Q_s - heat transfer rate dissipated from the surface of the machine to the surroundings; Q_{t1} - heat transfer rate generated in the result of friction and transferred to gas; Q_{t2} - heat transfer rate generated in the result of friction and transferred to the parts of machines; Q_t=Q_{t1}+Q_{t2} - heat transfer rate generated in the result of friction; Q_k - heat transfer rate transferred by convection from (or taken by) gas to the working chamber walls. Figure 1 shows the cross-section of multi-vane expander with marked main subassemblies (1 – cylinder, 2 – rotor, 3 – vane, 4 – vane slot) and an illustration of the heat transfer processes which occur in the multi-vane expander.

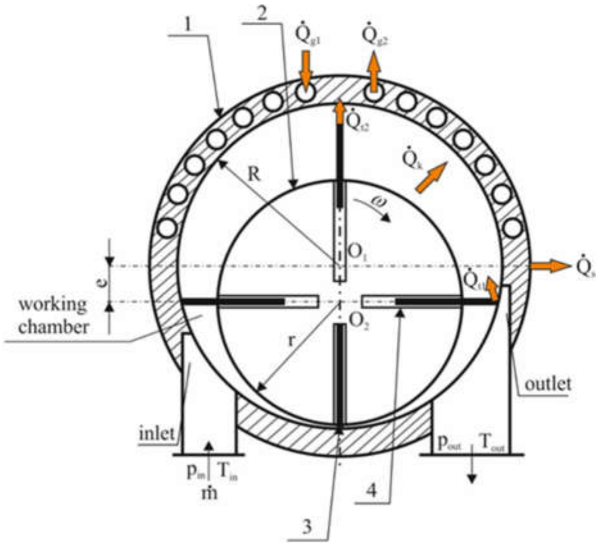


Figure 1: Cross-section of a multi-vane expander with marked main heat transfer rates and subassemblies.
1 – cylinder, 2 – rotor, 3 – vane, 4 – vane slot

The heat transfer rate emitted (or absorbed) from (by) the machine surface to (from) the environment can be expressed by Equation (1),

$$\dot{Q}_s = h_s\left(T_c - T_{air}\right)A \tag{1}$$

where h_s - heat transfer coefficient, film conductance from (to) the machine surface to (from) the environment, T_c - temperature of the machine surface, T_{air} - temperature of the surroundings, A - heat transfer surface (outer surface of the machine).
The heat transfer rate to the gas by the heating fluid can be calculated from Equation (2),

$$\dot{Q}_{g1} = \dot{m}_1 c_{p1}\left(T_{1in} - T_{1out}\right) \tag{2}$$

where m_1 - the mass flow of the heating fluid, c_{p1} – specific heat capacity of the heating fluid, T_{1in} - temperature of heating fluid at the inlet to the machine, T_{1out} - temperature of heating fluid at the outlet of the machine.
The heat transfer rate taken from the gas by the cooling fluid can be calculated from Equation (3),

$$\dot{Q}_{g2} = \dot{m}_2 c_{p2}\left(T_{2in} - T_{2out}\right) \tag{3}$$

where m_2 - the mass flow of the cooling fluid, c_{p2} – specific heat capacity of the cooling fluid, T_{2in} - temperature of cooling fluid at the inlet to the machine, T_{2out} - temperature of heating fluid at the outlet of the machine.
As shown in (Gnutek, 2004), the heat of friction can be generalized using Equation (4),

$$Q_{fe} = \left(1 - K_{fe}\right)L_{fe} \tag{4}$$

where Q_{fe} - friction heat emitted during the operation of the moving element, L_{fe} - work needed to overcome friction, K_{fe} - total input of non-thermal components of friction in energy expenditure, K_{fe} = 0.04 - 0.08 (Mamontov, 1961).
Given the contact surface of the rubbing elements, Equation (5) expressing the friction heat source output (expressed in W/m^2) can be written as (Mamontov, 1961),

$$\dot{q}_{\mathrm{fe}} = \left(1 - K_{\mathrm{fe}}\right) N_{\mathrm{fe}} \frac{1}{A_{\mathrm{s}}} \tag{5}$$

where N_{fe} - power consumed to overcome the friction of the element, A_s - contact surface of the rubbing elements.
From a scientific perspective, the most interesting heat transfer phenomenon in a multi-vane machine is the heat exchange process in a rotating working chamber. As shown in (Gnutek, 2004), to calculate the heat transfer coefficient h, the models for convective heat transfer applied to flow around flat surfaces can be used, with the assumption that this surface has a length of l which is equal to the circumference of the working chamber, and the speed of the flowing working fluid is determined using the Equation (6),

$$w = \omega r \tag{6}$$

Considering the above assumptions, it is possible to apply the following criterion relation for the laminar boundary layer (for $Re_l < 10^5$) (Gnutek, 2004),

$$Nu = 0.66\,\mathrm{Re}_{\mathrm{l}}^{0.5}\,\mathrm{Pr}_{\mathrm{p}}^{0.33}\,\varepsilon_{\mathrm{r}} \tag{7}$$

For the turbulent boundary layer, the following relation can be used (Gnutek, 2004) (the model is further referred to as ZG)

$$Nu = 0.037\,\mathrm{Re}_{\mathrm{l}}^{0.8}\,\mathrm{Pr}_{\mathrm{p}}^{0.43}\left(\frac{\mathrm{Pr}_{\mathrm{p}}}{\mathrm{Pr}_{\mathrm{s}}}\right)^{0.25} \tag{8}$$

where: $Re_l=\omega l^2/\nu$ - Reynolds number for the flat surface (length corresponding to the circumference of the working chamber); $Pr_p=\eta c_p/k$ - Prandtl number (at fluid temperature); Pr_s - Prandtl number (calculated for wall temperature); $\varepsilon_r=(\mu_m/\mu_w)^{0.14}$ - a factor depending on changes in the physical properties (i.e., kinematic viscosity) of the working fluid with a temperature change from the initial value of t_m to final value t_w.
The average value of the heat transfer coefficient from the walls of the working chamber into the gas can be determined using Equation (9),

$$h_{\mathrm{s}} = Nu\frac{k}{l} \tag{9}$$

In the case of multi-vane machines, the phenomenon of scraping the boundary layer by a vane has a significant impact on heat transfer processes. The Nusselt number, in this case, can be determined from the following relation (Gnutek, 2004),

$$Nu_{\delta} = 0.0228\,\mathrm{Re}_{\delta}^{0.75}\,\mathrm{Pr}^{0.33} \tag{10}$$

where: $Nu_\delta=h\delta/k_g$.
The Reynolds number, in this case, can be calculated from Equation (11),

$$\mathrm{Re}_{\delta} = \frac{w_{\mathrm{l}}\delta\rho_{\mathrm{g}}}{\eta} \tag{11}$$

where w_l - linear velocity of the vane face moving along the cylinder, δ - boundary layer thickness for free flow of the considered surface, η - dynamic viscosity of fluid.

The phenomenon of scraping the boundary layer through movable elements disrupting fluid flow occurs, among others, in scraped surface heat exchangers. One of the authors analyzed these issues in his previous research and presented the results in (Błasiak *et al.*, 2016; Błasiak *et al.*, 2017). Based on the analyzes carried out, he formulated the following criterion equation for binding Nusselt, Reynolds and Prandtl numbers for such heat exchange conditions

$$Nu = 1.765\,\mathrm{Re}_{\mathrm{m}}^{0.496}\,\mathrm{Pr}^{0.33} \tag{12}$$

Due to the similarity of the occurring phenomena, the authors also decided to use this correlation in the present analysis (the model is further referred to as B-P). In the literature, there are no available Nusselt correlations dedicated strictly to multi-vane machines. For this reason, direct comparison of the presented model with models from the literature is not possible. Nonetheless, to provide some comparison, the authors selected the machine in which the heat transfer conditions are the most analogous. Such a machine is a scraped surface heat exchanger (Błasiak *et al.*, 2016; Błasiak *et al.*, 2017) where scraping vanes continuously scrape the inner wall of the cylinder and thus intensify the heat transfer rate. In a multi-vane expander, the situation is similar with the difference that the purpose of the vanes is not to enhance heat transfer but to separate adjacent working chambers. Table 1 summarizes the equations which have been used in the simulations. These correlations have been applied for the calculation of the Nusselt number. The models are further abbreviated and referred to according to the first names letter of the authors.

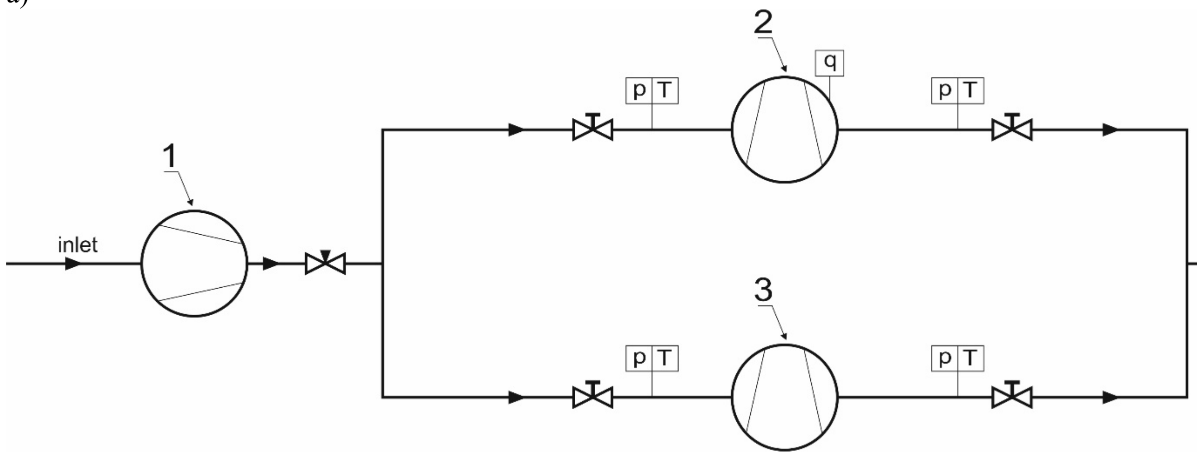
Table 1: Heat transfer models.

Model	Correlation for the Nu	Remarks	Reference
ZG	$Nu = 0.037\,\mathrm{Re}_{\mathrm{l}}^{0.8}\,\mathrm{Pr}_{\mathrm{p}}^{0.43}\left(\dfrac{\mathrm{Pr}_{\mathrm{p}}}{\mathrm{Pr}_{\mathrm{s}}}\right)^{0.25}$	$\mathrm{Re}_{\mathrm{l}} > 10^5$	(Gnutek, 2004)
K-K	$Nu = \dfrac{hD}{k} = \dfrac{k}{\delta}\dfrac{D}{k} = \dfrac{D}{\delta}$	–	(Kern *et al.*, 1958)
K-L-H	$Nu = \dfrac{2}{\sqrt{\pi}}\sqrt{n\,\mathrm{Re}_{\mathrm{D}}\,\mathrm{Pr}}$	$\mathrm{Re}_D = \dfrac{nD^2\rho}{\eta}$	(Kool *et al.*, 1958; Latinen *et al.*, 1959; Harriott, 1959)
P-B	$Nu = \dfrac{1}{\dfrac{\delta}{D} + \dfrac{1}{\sqrt{\dfrac{8}{\pi}\mathrm{Re}_D\,\mathrm{Pr}}}}$	–	(Penney *et al.*, 1967)
P-B$_{\mathrm{exp}}$	$Nu = 1.123\,\mathrm{Re}_{\mathrm{m}}^{0.78}\,\mathrm{Pr}^{0.33}\left(\dfrac{\eta_{\mathrm{b}}}{\eta_{\mathrm{w}}}\right)^{0.18}$	$Re_r > 400$ ethylene, glycol	(Penney *et al.*, 1969)
B-P	$Nu = 1.765\,\mathrm{Re}_{\mathrm{m}}^{0.496}\,\mathrm{Pr}^{0.33}$	Pr < 1	(Błasiak *et al.*, 2017)
E	$Nu = 0.11\,\mathrm{Pr}^{0.35}\left(0.5\,\mathrm{Re}_{\mathrm{w}}^{2}\right)^{0.35}$	–	(Gnutek, 2004)

3 EXPERIMENTAL SETUP DESCRIPTION

To experimentally investigate the heat transfer processes in multi-vane expanders, the test-stand was designed and realized. The scheme of this test-stand is presented in Figure 2(a) while Figure 2(b) shows the general view of test-stand. The test-stand uses an air compressor (1) as the working fluid source. Different types of volumetric expanders can be tested using the test-stand. Two expanders (2) and (3) can be simultaneously assembled in a test-stand.

a)



b)

Figure 2: An experimental test-stand
a) scheme, b) general view.

For the present study, the test-stand was equipped with a micro multi-vane expander featuring the maximum power of 300 W, which is designed to be applied in ORC system and was adapted for low-boiling working fluid, i.e., vanes made of cured silicone doped with graphite, seals made of Teflon and stainless steel bearings were used. More technical details related to this expander are presented by Kolasiński (2019). A needle valve regulates the air flow rate through the expander. The pressure transducers (featuring the measurement accuracy of ±1%) are applied for the working fluid pressure measurement (see p in Figure 2a) at the inlet and the outlet of the expander, while the working fluid temperature measurement at the inlet and at the outlet of the expander was proceeded by using PT100 resistance sensors (see, T in Figure 2a) featuring the measurement accuracy of ±0.1 °C). The heat flow in the expander is measured using the GSKIN heat flux density sensor glued to the cylinder of the expander (see q in Figure 2a). Heat flux density sensor is featuring the measurement accuracy of ±3%. Measurements are recorded by a data recorder. The working principle of the test system is very simple. Air from the compressor (1) is forced through pipelines to the inlet of the multi-vane expander (2). The mass flow rate of air can be regulated during the experiment using the needle valve. After the expansion, the air flows to the surroundings through the outlet pipeline. The experiment is started at a steady state, i.e., when the expander is in the thermodynamic equilibrium with the surroundings. The compressor is switched off, the inlet valve is opened, and the air is forced from the compressor reservoir to the inlet of the expander. In present experiment expander is not insulated and operates at idle. The authors performed series of experiments using the test-stand. To the knowledge of the authors, this experiment was the first attempt of measuring the heat transfer in multi-vane machine. Below, the selected experimental data are presented and discussed.

Figure 3a shows the air absolute pressure variation at the inlet of the expander vs the experimental time, while Figure 3b shows the air absolute pressure variation at the outlet of the expander vs the experimental time. The course of air pressure change at the inlet of the expander is constantly decreasing during the experiment due as the air reservoir is emptying. The course of the air pressure change at the outlet of the expander is different. In the first period of the experiment, pressure growth can be observed. Then, after reaching the peak value, the pressure is constantly decreasing and stabilizing at the ambient pressure level at the end of the experiment. The growth of the pressure visible in the first period of the experiment is probably caused by internal air leakages between the working chambers. In the further experimental time, the pressure of the air decreases and for this reason the internal leakages are limited.

a) b)

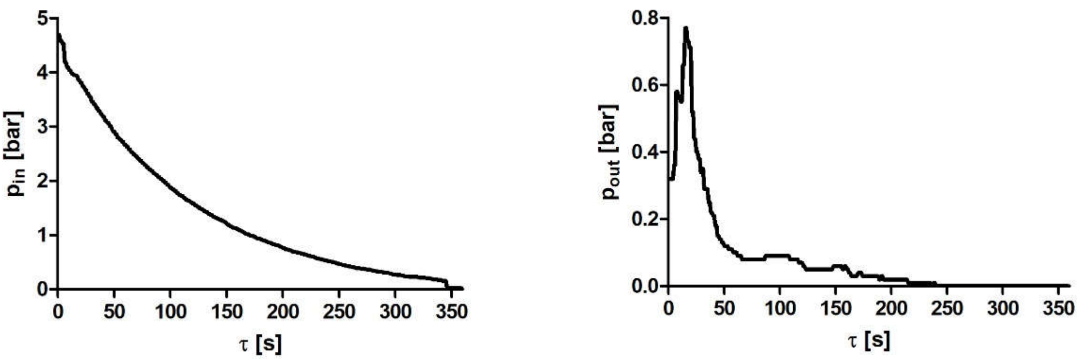


Figure 3: The variation of the working fluid absolute pressure vs the experiment time.
a) Pressure at the inlet of the expander, b) pressure at the outlet of the expander

Fig. 4a shows the air temperature variation at the inlet of the expander vs the experimental time, while Fig. 4b shows the air temperature variation at the outlet of the expander vs the experimental time.

a) b)

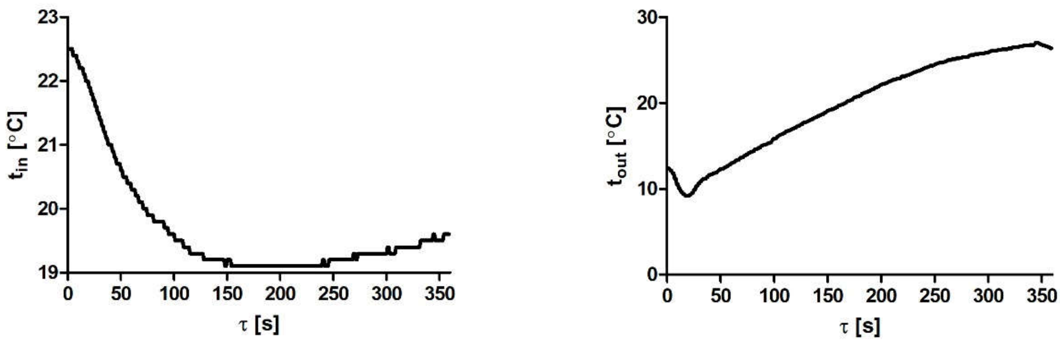


Figure 4: The variation of the working fluid temperature vs experimental time.
a) temperature at the inlet of the expander, b) temperature at the outlet of the expander

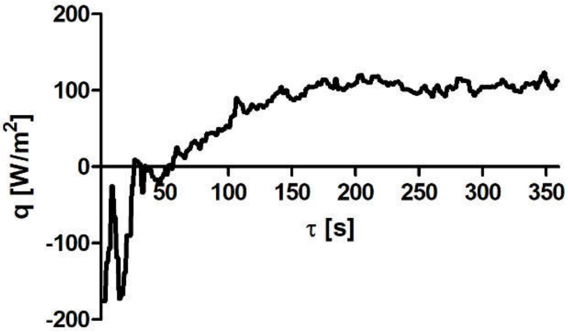


Figure 5: The variation of the heat flux density vs experimental time.

The course of air temperature change at the inlet of the expander is decreasing in the first part of the experiment and after reaching the minimum value is growing in the further part of the experiment. The course of the temperature change at the outlet of the expander is different. After a temperature drop, which is visible in the first part of the experiment, the temperature is constantly growing and stabilizes at the end of the investigation. During the first part of the experiment, the air temperature is dropping as a result of expansion what is visible in Figure 4a and Figure 4b. The course of change of the air temperature at the inlet and outlet of the expander in this part of the experiment is caused by the intensive cooling of the expander's subassemblies by the expanded gas. This effect is visible also in Figure 5, which visualizes the measured heat flux vs experimental time. In the first part of the experiment, the heat flux is negative, therefore heat is flowing from the surroundings to the machine. In the second part of the experiment, the cooling effect is limited as the air pressure at the inlet to the expander drops, and the internal heat sources (i.e., generated by friction between the expander's subassemblies) are giving heat to the subassemblies. These subassemblies are heating the expanding air. This effect is visible also in Figure 5. In the second part of the experiment, the heat flux is positive, therefore heat is flowing from the machine body to the surroundings.
For the same range of pressure and temperature as in the experiment, the authors computed the time averaged Nusselt numbers from the models listed in Table 1 (see Table 2). As can be seen, the mathematical models give the values of the Nusselt number in a very wide range. Additionally, some of them differ by an order of magnitude. Thus, it exists a strong demand on taking deeper research on this issue. The mathematical models have to be validated experimentally what is the goal of the future work.

Table 2: A comparison of the time averaged Nusselt numbers.

p_{in}	ZG	K-K	K-L-H	P-B	P-B$_{exp}$	B-P	E
1	119.6	937.5	583.6	103.8	116.1	131.9	88.5
2	157.6	937.5	693.4	120.8	152.0	156.5	112.7
3	193.2	937.5	787.5	134.8	185.3	177.6	134.7
4	227.3	937.5	871.8	147.0	217.2	196.4	155.3
5	260.4	937.5	948.9	157.8	247.9	213.7	174.8
6	292.5	937.5	1020.5	167.6	277.7	229.7	193.6

4 SUMMARY AND CONCLUSIONS

In this paper, the experimental and modelling survey results on heat transfer inside a multi-vane expander have been presented. The experiments were proceeded using a specially implemented test-stand equipped with a micro multi-vane expander designed to be applied in ORC systems. Experimental results showed the course of pressure, temperature and heat flow change during expander operation with changing working fluid pressure at the inlet of the expander. This results gave an insight into the heat transfer phenomena proceeding in the expander and showed that the heat flow direction during gas expansion depends on the operating parameters of the expander (i.e., gas pressure and temperature). In the first part of the experiment machine was intensively cooled by expanded gas and heat was flowing from the surroundings to machine. In the second part of the experiment machine elements were heated due to friction between the expander's subassemblies and heat was flowing from the machine body to the surroundings.

Additionally to the experimental research, the time-averaged Nusselt number has been calculated and compared with available correlations from the literature for similar heat transfer conditions that occur in a multi-vane expander. The comparison of the modelling results suggests that the applicability of models available in the literature and applied for modelling the heat transfer in other machines and devices with rotating vanes (e.g., scraped surface heat exchangers) is limited in the case of estimating the heat transfer processes in multi-vane machines. Therefore, further work on this topic should be focused on experimental research on heat transfer conditions in these machines and developing Nusselt number correlations that are specifically valid for multi-vane expanders. Further experiments and numerical modelling can achieve this goal.

REFERENCES

Badr, O., O'Callaghan, P. W., Hussein, M., Probert, S. D., 1984, Multi-vane expanders as prime movers for low-grade energy organic Rankine-cycle engines, *Applied Energy*, vol. 16, no. 2, p. 129-146.

Błasiak, P., Pietrowicz, S., 2016, Towards a better understanding of 2D thermal-flow processes in a scraped surface heat exchanger, *Int. J. Heat Mass Transfer*, vol. 98, 240–256.

Błasiak, P., Pietrowicz, S., 2017, An experimental study on the heat transfer performance in a batch scraped surface heat exchanger under a turbulent flow regime, *Int. J. Heat Mass Transfer*, vol. 107, 379–390.

Gnutek, Z.., Kalinowski, E., 1994, Application of rotary-vane expanders in systems utilizing the waste heat, *International Compressor Engineering Conference at Purdue,* Purdue University, no. 2, p. 129-146.

Gnutek, Z.., 1997, *Sliding-Vane Rotary Machinery. Developing Selected Issues of One-Dimensional Theory*, Wrocław University of Technology Publishing: Wrocław, Poland.

Gnutek, Z., 2004, *Gazowe objętościowe maszyny energetyczne—Podstawy,* Wrocław University of Technology Publishing: Wrocław, Poland.

Harriott, P., 1959, Heat transfer in scraped-surface exchangers, *Chem. Eng. Prog. Symp. Ser.*, vol. 55, 137-139.

Kern, D.Q., Karakas, H.J., 1958, Mechanically aided heat transfer, *Chem. Eng. Prog.*, vol. 55, 141-168.

Kolasiński, P., 2019, Application of the multi-vane expanders in ORC systems - a review on the experimental and modelling research activities, *Energies*, vol. 12, 2975.

Kool, J., 1958, Heat transfer in scraped vessels and pipes handling viscous material, *Chem. Eng. Res. Des.*, 253-258.

Latinen, G.A., 1959, Discussion of the paper, "correlation of scraped-film heat transfer in the votator", *Chem. Eng. Sci.*, vol. 9, 263-266.

Mamontow, M., 1961, *Voprosy termodynamiki tiela pieremiennoj massy,* Obrongiz: Moscow, Russia.

Penney, W.R. ; Bell, K.J., 1967, Close-clearance agitators-part 2. Heat transfer coefficients, *Ind. Eng. Chem.*, vol. 59, 47-54.

Penney, W.R. ; Bell, K.J., 1969, Heat transfer in a thermal processor agitated with a fixed clearance thin flat blade, *Chem. Eng. Prog. Symp. Ser.*, vol. 65, 1-11.

Rak, J., Błasiak, P., Kolasiński, P., 2018, Influence of the Applied Working Fluid and the Arrangement of the Steering Edges on Multi-Vane Expander Performance in Micro ORC System, *Energies*, vol. 11, 892.