PERFORMANCE OF A SMALL-SCALE ORGANIC RANKINE CYCLE SYSTEM USING A REGENERATIVE FLOW TURBINE: A SIMULATION ANALYSIS

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ABSTRACT

Organic Rankine Cycles (ORCs) have an interesting potential in small-scale power production using low-temperature heat sources. Despite their unique applications in power production from low-temperature heat sources, the widespread use of small-scale ORC systems is still challenging especially due to high primary cost resulting in long-term payback periods. The reason goes back to several inherent technical limitations especially related to the available expander machines, which are usually accounted for a significant amount of the total investment cost of such systems. In this paper, a Regenerative Flow Turbine (RFT) adopted in a small-scale ORC prototype is investigated by means of a modelling study. The characteristic curves of the considered turbine have been obtained using CFD analysis to evaluate its performance in an ORC unit working with R245fa. Results of CFD have shown that performance of the RFT has been improved considerably using R245fa and real gas model instead of air and ideal gas model, and total-to-static isentropic efficiency could reach up to 45 % at the best. The system shows an overall performance that is comparable to that of ORC systems despite being penalized because of low isentropic efficiency of the RFT; however, the main advantages of RFTs are considerably low investment costs and high reliability which are important factors in the design of such small-scale power systems.

1. INTRODUCTION

In the last decade, rising attention has been paid on micro and small-scale Organic Rankine Cycles (ORCs) to produce electricity using low-temperature heat sources from renewables or waste heat. Many of these studies have focused on the expansion machine, which is the key component of an ORC unit not only in terms of performance but also concerning its reliability and capital cost.
Among the different types of expanders, regenerative flow (RF) turbo-machines have shown unique characteristics which make them suitable for some specific applications. Research on RF turbo-machines have been started some decades ago, and theories have been developed to understand the performance of RF pumps (RFPs) (Meakhail and Park, 2005, Quail *et al.*, 2011, Song *et al.*, 2003, Yoo *et al.*, 2005), blowers (RFBs) (Badami and Mura, 2010), and compressors (RFCs) (Engeda and Raheel, 2003) using semi-empirical models. Whether a RF turbo-machine derives the fluid (pumps or compressors) or is derived by the fluid (turbines), the operating principle of the machine is the same. In a RF turbo-machine, while part of the flow follows a peripheral path through the channel, the other part enters the impeller pockets and returns to the channel several times. This secondary flow (regenerative flow) exchanges momentum between the impeller and the mainstream (in the peripheral

direction) resulting in a helical trajectory in the peripheral direction and generating a pulse pressure variation inside the flow (Nejad *et al.*, 2017).

Although a RF turbo-machine is considered as a kinetic machine (Karlsen-Davies and Aggidis, 2016) and it is similar to a centrifugal turbo-machine, in RF turbo-machines the pressure changes in the peripheral direction rather than in the radial direction (Quail *et al.*, 2012). At low flow rates, the circulatory velocity, which represents the swirl of the flow and its normal vector is in the peripheral (tangential) direction, increases together with the number of completed swirls of the flow (Karlsen-Davies and Aggidis, 2016). This means that high heads (hydraulic pressure) are provided by RFCs and RFPs when they work at low flow rates while the trend of the reduction in power with flow rate shows a straight head-capacity curve (Badami and Mura, 2010). On the other side, they are less efficient at low flow rates (Karlsen-Davies and Aggidis, 2017, Nejad *et al.*, 2017, Raheel and Engeda, 2002).

RF turbo-machines inherit some characteristics of positive displacement machines without the issues of wear and lubrication (Raheel and Engeda, 2002). Similar to positive displacement machines, they can handle two-phase flows to some extent (Karlsen-Davies and Aggidis, 2016), which makes them suitable to operate with hot or volatile liquids as a pump (Quail *et al.*, 2011), or with a few degrees of superheating in small-scale power plants as a turbine. For a similar reason, RFCs can be employed in small-scale cryogenic cycles. In addition to these characteristics, self-priming, low Net Positive Suction Heads, compact design, high-temperature capability, noiseless operation, high reliability, and low manufacturing costs are other merits of RF turbo-machines compared to other turbo-machines at the same working conditions (Karlsen-Davies and Aggidis, 2016, Raheel and Engeda, 2002). Nevertheless, low efficiencies, typically between 35% and 50% (Quail *et al.*, 2010), have restricted the large adoption of RF turbo-machines, especially in power systems.

In general, regenerative flow turbines (RFTs) have not been extensively investigated in the literature because of the large diffusion of higher efficiency turbines in power systems resulting in RFTs being crossed out from the list of options. Despite the limited isentropic efficiency, the rising interest in small-scale ORC systems in the last decade is giving chances to RFTs also thanks to their low construction costs compared to radial flow turbo-machines and even volumetric expanders. Indeed, investment costs are accounted as one of the major barriers to widespread adoption of ORC units in the residential energy market (Pereira *et al.*, 2018). For their peculiar characteristics, RFTs can be considered a low-cost and viable alternative for the expanders in small-scale waste heat recovery systems.

So far, only a few researchers have focused on RFTs by means of experimental and numerical studies with the objective of improving their efficiencies. For instance, (Bartolini and Salvi, 1996) experimentally evaluated the performance of an RFT designed to be used in gas pipelines instead of conventional expansion valves. In a similar study, (Balducci and Bartolini, 1992) conducted a test campaign using the same RFT. However, to the best of the authors' knowledge, RFTs have not been tested or simulated in the literature using organic fluids, and their performance in ORC systems needs to be studied further. Hence, the performance of the RFT with R245fa, a popular organic fluid in ORC systems, is investigated in this paper using the CFD analysis in ANSYS Fluent©. Then, an own code developed in MATLAB© is used to assess the performance of an ORC unit with such expander machine. Compared to the experiments, the computational analysis is advantageous due to its high repeatability, lower costs and reduced time. In the case of RFTs, it is especially useful to observe the complex swirling flow (Meakhail and Park, 2005). The RFT studied has the same geometry studied by (Balducci and Bartolini, 1992) experimentally, and by some authors of this paper numerically (Moradi *et al.*, 2019), which in both studies air was used as the working fluid. Indeed, the main goal of this work is to introduce an alternative solution for expander machines in ORC units along with the main challenges in its engineering design.

2. NUMERICAL MODEL

2.1 Organic Rankine Cycle (ORC)

The first step for the design of an ORC system is to select the suitable organic fluid considering the temperature of the heat source, the critical temperature and pressure of the organic fluid, and other criteria such as zero ODP, low GWP, and low flammability index.

In this study, an ORC system is developed to represent the performance of the simulated RFT in a low-temperature power system by means of a MATLAB© code. The heat source is considered steam at 3 bar and 160 ºC at the inlet of the evaporator. At this temperature level, one of the suitable working fluids to be adopted in an ORC unit with scroll expander is R245fa (Quoilin *et al.*, 2012). Furthermore, R245fa has been used in many ORC systems and it has shown good performance in waste heat recovery applications (Declaye *et al.*, 2013, Galloni *et al.*, 2015, Muhammad *et al.*, 2015), and when environmental impacts and safety levels are concerned, it is the most suitable for waste heat recovery systems (Wang *et al.*, 2011). However, R245fa is a high-GWP refrigerant and ongoing researches are trying to replace it with low-GWP fluids (Petr and Raabe, 2015). Anyhow, because of its widespread adoption in small-scale low-temperature ORC systems, R245fa has been considered as working fluid in this analysis.

The design of the ORC system is performed considering the characteristic curves of the RFT. The mass flow rate of the organic fluid is an input of the model whilst the isentropic efficiency of the RFT and its pressure ratio (PR) are obtained from the CFD results. The PR of the pump is considered equal to one of the RFT. To avoid a two-phase flow at the suction of the pump, 5 ˚C sub-cooling is assumed in the model. The temperature difference between the inlet of the cooling water (at 20 ˚C) and the saturation temperature of the organic working fluid in the condenser is considered 25 ˚C resulting in a pressurized condenser, which is beneficial for its operation.

The heat exchangers are modelled using the known inlet temperature and pressure and the design outlet temperature of both streams neglecting pressure drops. The outlet temperature of the organic fluid stream is calculated using the assumed superheating and sub-cooling in the model of the evaporator and the condenser respectively as reported in Table 1. The mass flow rate of the organic fluid stream is taken from the range of the mass flow rate adopted in CFD simulations of the RFT; then using the energy balance between two streams, the mass flow rate of the other stream is calculated. Furthermore, the model checks the possibility of such a thermal performance using the calculated temperature pinch. To check the temperature pinch in each specific working condition of the heat exchangers, a vector of nodes is created using the equal spacing between inlet and outlet enthalpies of one stream considering the heat exchangers as two counter-flow streams (channels) where the heat is transferred through a boundary (wall). The concept of this approach is similar to the Finite Difference method in which heat exchangers are divided into cells using the known heat transfer area. Instead, the model uses enthalpy variations of one stream to create the vector of cells. Hence, it does not require a specific type of a heat exchanger or its area suitable for the preliminary system analysis. The heat flux in each cell is calculated using the enthalpy difference between consecutive nodes. The calculated energy is assumed to be totally transferred to the adjacent cell of the other stream to calculate the enthalpy of the next node of the other stream. Therefore, the temperature profile of the heat exchanger is obtained.

The minimum temperature difference between two adjacent nodes is set as the temperature pinch of the heat exchanger. Results of the model limit the design temperature of the outlet cooling water to a maximum of 35 ˚C considering a temperature pinch of 10 ˚C as the minimum allowable value for both evaporator and condenser. Figure 1 shows the temperature profiles in the evaporator and condenser and the corresponding temperature pinch at the specified saturation pressures. As can be seen, the pinch in the evaporator is about 3 times higher than the minimum allowable value of the model (30.9 K in the evaporator, and 12.7 K in the condenser). Even though the low-temperature pinch is important to better exploit the heat source from the second law of thermodynamic point of view, no maximum pinch is considered in the preliminary analysis presented in this study.

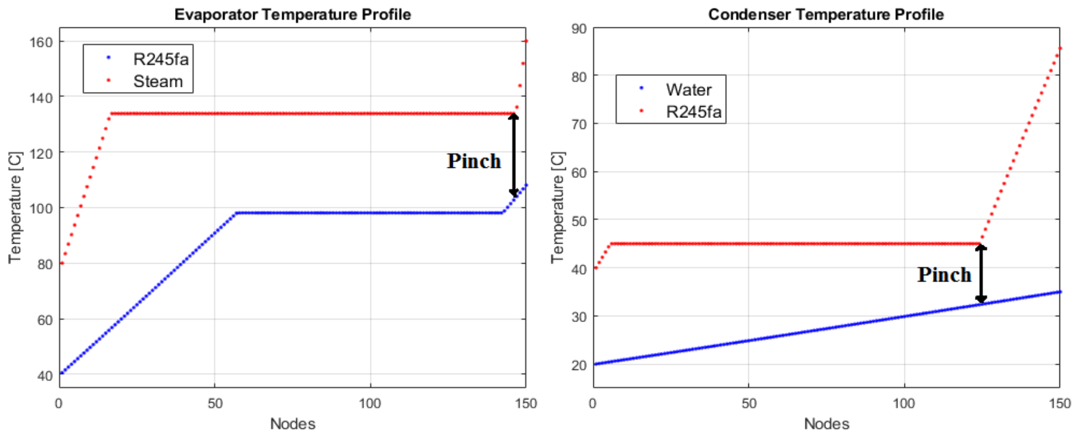

Figure 1: Temperature distribution in the heat exchangers of the ORC unit at $\dot{m}$ = 0.8 kg/s, P_{cond} = 2.94 bar & P_{evap} = 12 bar

Furthermore, the isentropic and the mechanical efficiencies of the pump are considered 75% and 96% respectively (Camporeale *et al.*, 2015), while the pressure drops in the heat exchangers are neglected. The parameters considered in the design of the ORC system are summarized in Table 1.

Table 1: Design parameters of the ORC system

Organic Fluid	R245fa	Heat source inlet	Steam, 3 bar, 160 ˚C
Pump isentropic efficiency	75%	Heat source outlet	Water, 80 ˚C
Pump mechanical efficiency	96%	Cold sink inlet	Water, 20 ˚C
Electrical generator efficiency	97%	Cold sink outlet	Water, 35 ˚C
Sub-cooling at the pump inlet	5 ˚C	Super-heating in the evaporator	10 ˚C

2.2 CFD model of the RFT

The geometry of the RFT (represented in Figure 2a) and the CFD model used in this work are the same as in the previous study (Moradi *et al.*, 2019). The geometry consists of the inlet and outlet ports, the casing, the stripper, and the impeller. In turn, the latter consists of a disk with blades on both sides to better balance the impeller as shown in Figure 2b. The casing covers the impeller on both sides and creates the channel from the inlet to the outlet. The stripper separates the inlet from the outlet of the machine to avoid leakage flow between them while a small gap separates the impeller blades from the stripper body to avoid fracture of the stripper. However, this gap causes some leakage flow from the inlet of the machine toward the outlet and, as a consequence, a minimum gap is recommended in RF turbo-machines (Raheel and Engeda, 2002). Figure 2a shows the halved-geometry of the RFT that is used for the CFD simulations. The symmetry plane crosses through the middle of the impeller, and its normal vector is the axis of the impeller.

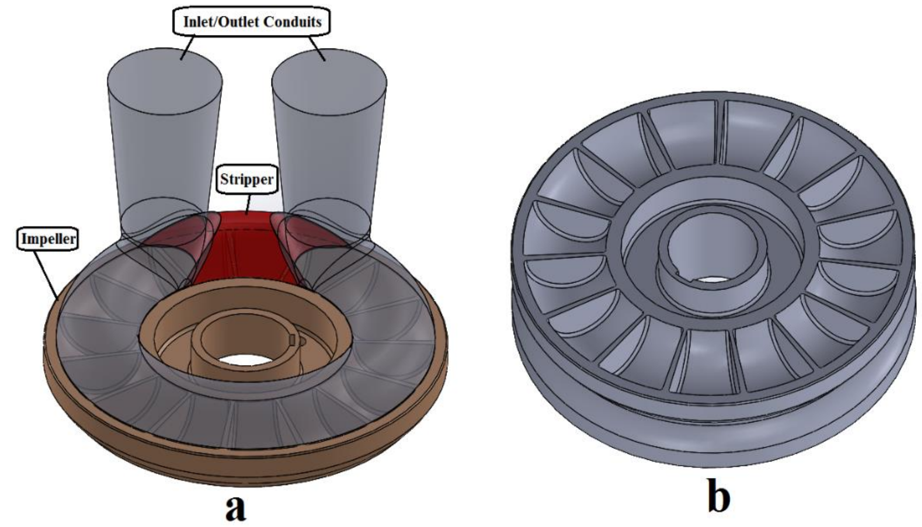

Figure 2: 3D model of the a) RFT, b) impeller

The CFD model was previously validated using experimental data from the literature (Moradi *et al.*, 2019). The same settings of the mesh and of the boundary conditions are adopted in this study, whilst R245fa is considered as working fluid instead of air. The Peng-Robinson real gas model for density has been adopted rather than the ideal gas model as in the previous study. The turbulence model is the Reynolds Stress baseline model (BSL) using k-ω SST (Shear-Stress Transport) as the initial solution. The pressure-velocity coupling is solved using PISO scheme, and PRESTO! is chosen for pressure discretization scheme as suggested for 3-D domains with highly swirling flows. The rotation of the impeller is considered by means of a steady state model, the Moving Reference Frame (MRF) also known as Frozen Rotor approach. Boundary conditions of the CFD model are reported in Table 2. The outlet pressure is assumed fixed and is taken from the saturation pressure of a condenser in a typical ORC unit working with R245fa, and the outlet temperature is estimated using the performance of the RFT in the experiments (with air). In addition, the inlet temperature is set to a typical value of an ORC unit, whilst the range of the mass flow rate is determined from CFD results of the RFT considering the maximum suction pressure of 20 bar, which is equivalent of PR of about 6.5. For the sake of conciseness, details of the mesh and model are not presented in this paper, and interested readers are referred to find them in (Moradi *et al.*, 2019).

Table 2: Boundary conditions of the CFD model

Inlet BCs		**Outlet BCs**		**RPM**
Mass flow (kg/s)	Temperature (K)	Pressure (bar)	Temperature (K)	
0.3-0.9	393	3	363	1500, 3000, 6000

3. RESULTS AND DISCUSSION

Results of the CFD model of the RFT are presented in section 3.1 using dimensional numbers. In the following section, the same characteristic curves are used in the simulation of the ORC system, which its design features are elaborated in the previous section.

3.1 Performance Characteristics of the RFT with R245fa

The performance of the RFT is expressed in terms of the total-to-static isentropic efficiency, the total pressure ratio, and the output power. The total-to-static isentropic efficiency is defined as in Equation (1):

$$\eta_{t,s} = W_{act}/W_{is} = (h_{01}-h_{02})/(h_{01}-h_{2s}) \tag{1}$$

where 1 and 2 indicate the inlet and outlet of the RFT respectively, and 0 indicates total enthalpy. Values of temperature and pressure are obtained using the area-weighted average from CFD results, and then enthalpies that are presented in Equation (1) are calculated using CoolProp fluid database in MATLAB© software.

The PR of the RFT is calculated at different working conditions, and it increases linearly with the mass flow rate as shown in Figure 3. In contrast to traditional turbines, the PR is higher at the lower rotational speed. This trend was also observed by (Balducci and Bartolini, 1992) during their experiments, and in the numerical results with air (Moradi *et al.*, 2019). The reason is related to the intrinsic characteristics of the RFT: the rotation of the impeller blades creates a void effect in the channel. As a result, more gas flows through the channel at higher rotational speeds for a given pressure ratio. Hence, for a given mass flow rate of the gas, a lower pressure ratio occurs at higher rotational speeds.

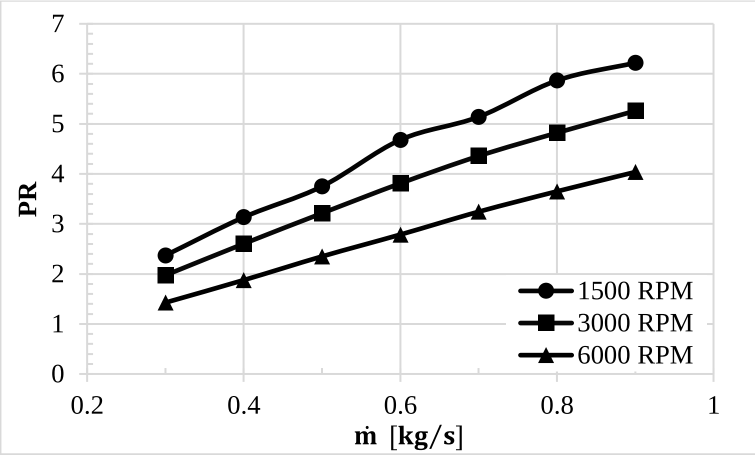

Figure 3: The total pressure ratio by the mass flow rate in different RPMs

The low isentropic efficiency of RFTs is undoubtedly the most important barrier in front of their wide-spread use in power systems. Figure 4 represents the isentropic efficiency of the RFT with varying mass flow rate for different rotational speeds of the impeller (RPMs). The overall isentropic efficiency achieved by the prototype RFT is relatively low compared to commercially available positive displacement expanders or dynamic turbines; however, it is higher compared to the same RFT working with air.

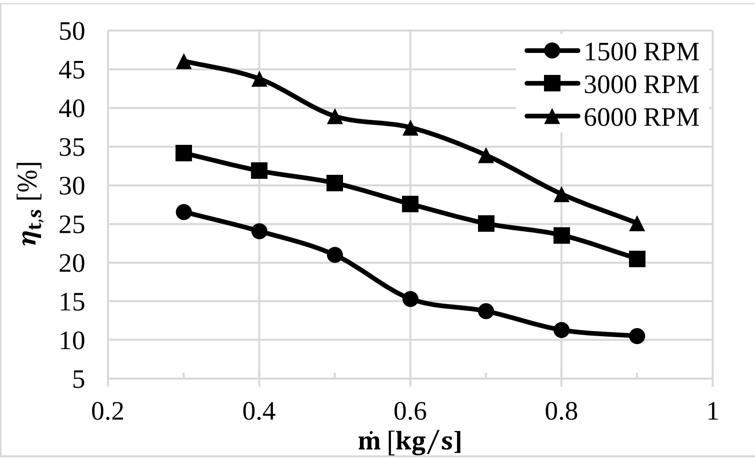

Figure 4: Isentropic efficiency by the mass flow rate in different RPMs

The higher isentropic efficiency of the RFT using R245fa as the working fluid (compared to air) is due to the difference between properties of the two fluids, especially the molecular weight of R245fa that is almost 4.6 times higher than that of the air. As a result, the flow follows less the swirling path and the enthalpy change between inlet and outlet is less when R245fa is the working fluid. This effect can be recognized in Figure 5 that represents streamlines of the flow for air and R245fa at the same working conditions. As can be seen, the flow does not develop the swirl with R245fa as much as with air, and variations in total temperature are considerably less (neglecting the impact of the low-temperature backflow in both cases). In other words, flow blows through the channel with R245fa resulting in fewer losses due to swirl and momentum exchange between the impeller and the mainstream in the channel. In addition, comparison between average circulatory velocity in a plane at the middle of the channel shows that it is almost 1.5 times higher with air compared to R245fa, which confirms the weaker swirl with R245fa as shown in Figure 5.

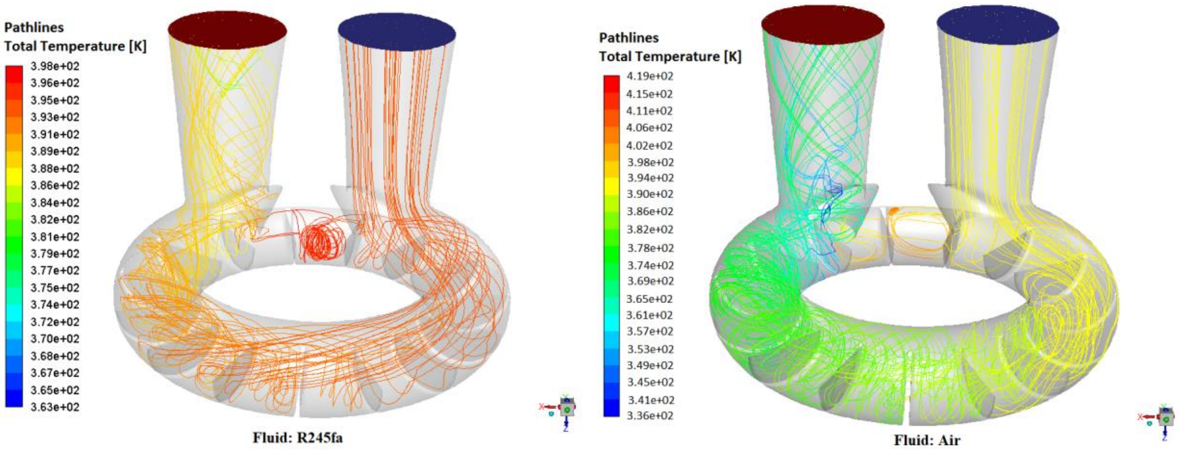

Figure 5: Streamlines colored by the total temperature at 6000 RPM & $\dot{m}$ = 0.3 kg/s for R245fa and air

Figure 6 reports the output power of the machine. In general, the power of the RFT increases almost linearly with an increment of the mass flow rate up to a certain point. After this point, the output power remains almost constant, while the isentropic efficiency still goes down. Therefore, the mass flow rate should not be exceeded to the values higher than this point for each RPM. In addition, the mechanical losses are expected to increase at higher rotational speeds of the impeller and, consequently, both the isentropic efficiency and the output power may be affected negatively at higher RPMs.

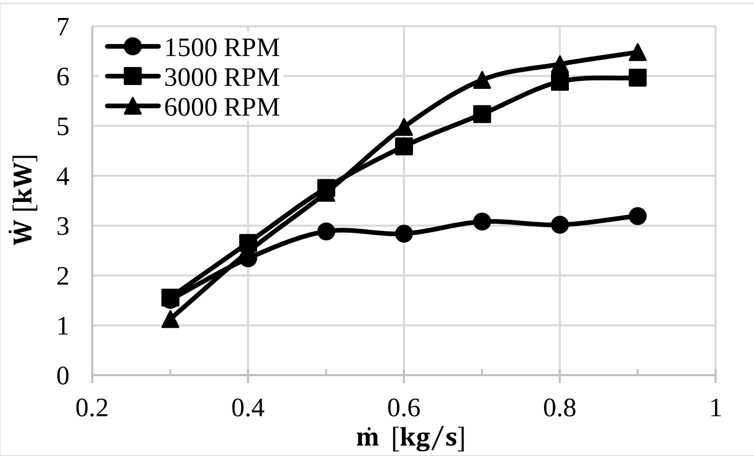

Figure 6: Output Power by mass flow rate in different RPMs

3.2 Performance of the RFT in an ORC system

Before presenting results of the ORC system, it should be noted that the range of the isentropic efficiency of the studied RFT is less than the average values of volumetric expanders or turbo-machines in similar systems. Therefore, it affects the overall performance of the system accordingly.

Results presented in Table 3 are performed at 6000 RPM that the RFT shows higher isentropic efficiency. In addition, the simulation is performed in two mass flow rates of 0.6 kg/s and 0.8 kg/s corresponding to pressure ratios of 3 and 4 approximately.

Table 3: Performance of the ORC system

$\dot{m}_{ORC} = 0.6\ kg/s$		$\dot{m}_{ORC} = 0.8\ kg/s$	
P_{evap}	9 bar	P_{evap}	12 bar
$\dot{W}_{pump}$	0.39 kW	$\dot{W}_{pump}$	0.78 kW
$\dot{W}_{RFT,el}$	4.69 kW	$\dot{W}_{RFT,el}$	6.27 kW
$\eta_{net,el}$	3.2 %	$\eta_{net,el}$	2.9 %

4. CONCLUSIONS

This paper numerically investigated the performance of a RFT using R245fa and the related ORC system. The results can be highlighted as follows:

- The PR of the RFT decreases with RPM, thus confirming the peculiarity of this type of expander.
- The isentropic efficiency of the RFT achieves a peak of more than 45%, explained via a less swirl showed in the CFD results has been found.
- The output power increases with the mass flow rate almost linearly, but up to a certain limit.
- The net electrical efficiency of the ORC unit is around 3%, at the highest RPM of the RFT.

Therefore, the range of the mass flow rate that can be elaborated by the RFT is higher than the one with volumetric expanders but its isentropic efficiency, despite being increased using R245fa, is still low. Nevertheless, low manufacturing costs and high reliability of RFTs can be highlighted further compared to turbo-machines working in this range of mass flow rates. Nevertheless, a re-engineering of the design of the RFT must still be performed to reduce the main losses, especially the leakage flow through the clearance gap between the stripper body and blade tips. In this way, the RFT can reach higher efficiencies and even if they do not reach to the current level of the performance of turbo-machines, they can still be an alternative solution for current technology of expander machines for low-temperature ORC systems. The investigated RFT is still a prototype, and this study revealed the necessity of the future experimental study of such expander devices to be adopted in small-scale ORC systems.

NOMENCLATURE

P	Pressure	(bar)
$\dot{m}$	Mass flow rate	(kg/s)
$\dot{W}$	Power	(kW)
PR	Pressure Ratio	(-)
RFT	Regenerative Flow Turbine	(-)

Subscript

evap	Evaporator
cond	Condenser
act	Actual
is, s	Isentropic
t	Thermal
el	Electrical
0	Total
1 & 2	Inlet & Outlet

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