Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 148 (2018) 66–73

Energy Procedia

www.elsevier.com/locate/procedia

73rd Conference of the Italian Thermal Machines Engineering Association (ATI 2018), 12–14 September 2018, Pisa, Italy

Thermodynamic simulation of a small-scale organic Rankine cycle testing facility using R245fa

Ramin Moradi[a], Roberto Tascioni[a], Emanuele Habib[a], Luca Cioccolanti[b], Mauro Villarini[c], Enrico Bocci[d,]*

[a]*Sapienza University of Rome (RM), Italy*
[b]*eCampus University, Via Isimbardi 10, 22060 Novedrate (CO), Italy*
[c]*La Tuscia University, Via San Camillo de Lellis snc, 22060 Viterbo (VT), Italy*
[d]*Marconi University, Via Paolo Emilio 29, 00193 Rome (RM), Italy*

Abstract

The development of small-scale efficient and low-cost Organic Rankine Cycle (ORC) units using low temperature waste heat for electricity production is attracting a lot of interest nowadays. This paper presents the thermodynamic simulation of a small scale regenerative ORC testing facility. The facility mainly consists of an electric steam generator to produce steam at 170°C and 7.5 bar as hot source, water at 20°C as cold sink and a scroll compressor converted to be used as expander. Simulation was performed by means of MATLAB, and CoolProp external library was used for the thermo-physical properties of the R245fa, the organic working fluid. The performance of the system in a certain range of working conditions has been calculated, obtaining maximum efficiency of 9.6% and net power of 17 kW. Simulations are used to understand the effect of different characteristics of main components on the performance of the system before running the test bench. Thus, allowing the future experimental campaign that will verify the validity of the model.

Keywords: ORC unit; Simulation analysis; Scroll expander;

* Corresponding author Enrico Bocci. Tel.: +390637725-341; fax: +39-06233296906.
E-mail address: e.bocci@lab.unimarconi.it

10.1016/j.egypro.2018.08.020

Nomenclature

$\dot{W}$	Power of pump or expander (kW)	Subscripts and abbreviations	
$\dot{Q}$	Thermal Power in heat exchangers (kW)	ORC	Organic Rankine Cycle
$\dot{m}$	Mass flow rate (kg/s)	PHE	Plate Heat Exchanger
h	Specific Enthalpy (J/kg)	RPM	Rotation Per Minute
P	Pressure (Pa)	CHP	Combined Heat & Power
D	Density (kg/m^3)	is	Isentropic
T	Temperature (K)	su	Suction of expander
SV	Swept Volume of the expander (lit/min)	in	Inlet
r_v	Volume ratio of expander	out	Outlet
Greek Symbols		int	Internal
η	Efficiency (expander)	ref	Refrigerant
ν	Specific Volume (m^3/kg)	c	Cold
ε	Efficiency (thermal)	h	Hot
th	Thermal	exp	Expander

1. Introduction

Energy production from medium and low temperature heat sources has attracted many interests recently. Unlike traditional power plants, ORC systems showed a flexible approach for heat recovery as small-scale power plants [1]. Heat sources can be renewable energy sources including solar, geothermal, and biomass, or waste heat sources such as engine exhaust and industrial waste heat.

When it comes to small-scale energy production, primary costs are a controversial factor since the generated power and efficiency of the cycle is low enough at the present time to limit the widespread adoption of ORC plants. Therefore, efficiency and primary costs of the plant should always be considered simultaneously at the design stage. However, this effect becomes almost negligible when the heat source is free. It is also noteworthy that in a small-scale ORC unit, expander machine allocates most of the capital cost to itself [2]. Hence, the expander machine is the heart of small-scale power plants regarding both energy and economic performance.

Expanders can be categorized in two main groups: volumetric expanders (positive displacement expanders) and dynamic machines (turbo-machines). Volumetric expanders are more suitable for medium and small scale plants because they can operate with lower flow rates, higher pressure ratios, and lower rotational speeds compared to turbo-machines at the same working conditions. They are also less costly [3], and less sensitive to two-phase flow that can occur when adequate superheating degree of the fluid is not achieved.

Converting available positive displacement compressors from refrigeration applications to act as expanders is a feasible option to reduce the investment cost of ORC unit. However, this entails also some drawbacks. Indeed, these machines are initially designed to act as compressors at different pressure ratios and temperatures compared to those of expanders. For example, scroll compressors are usually designed for lower volume ratios with respect to those needed by an ORC unit. This means a reduction in the isentropic efficiency of the expander and the performance cycle in general [4, 5]. Moreover, the working fluid of an ORC system usually differs from the original fluid of the compressor, leading to different leakage losses and nominal pressure ratios of the expander [4]. Several studies have already focused on this topic. For example, Lemort et al [6] presented a semi-empirical model of an open-drive oil-free scroll compressor converted into an expander. The expander was tested in an ORC with HCFC-123 that its critical temperature is about 30^{o}C higher than the working fluid in the present study (R245fa). A number of necessary parameters for their model were obtained from experimental data using genetic algorithm optimization method. Although the model was specific for machine, working fluid, and range of working conditions, it was able to predict performance of the expander with low discrepancy from experimental data. Results of their analysis showed that internal flow leakage is the most significant loss affecting the isentropic efficiency of the machine destructively. In addition, they showed that thermal losses of the expander cannot be neglected, and this effect becomes more important when average temperature of the machine grows up. Quoilin et al [7] studied an ORC with HCFC-123 as working fluid using both numerical and experimental approaches. They used semi-empirical models for scroll expander and plate heat exchangers. All models of the components were compared with experimental data, and the model of the cycle was validated against experiments. Their results showed that the ORC efficiency was decreased because of low efficiency of the positive displacement pump (about 15%), and sub-cooling at the condenser outlet to maintain the NPSH of the pump. They also indicated that there is an optimum rotational speed for the expander for which the sum of leakage and mechanical losses is minimum. Declaye et al [5] conducted an experimental study of an ORC using R245fa, with nominal net power output of 1.8 kWe. A commercial air compressor was converted into an expander, and its performance map was obtained using 74 steady-state experimental points. The maximum efficiency of their system was 8.5% for evaporation and condensation temperatures of 97.5°C and 26.5°C respectively. They showed that the expander efficiency depends on pressure ratio, and that there is an optimum value of pressure ratio for each flow rate and inlet pressure. However, the shaft power increases monotonically with pressure ratio. Galloni et al [2] considered a scroll expander in a non-regenerative ORC to produce electrical power with temperatures of the heat source and sink in the range of 75-95°C and 20-33°C respectively using as R245fa as working fluid. The best performance achieved was about 1 kW electrical power and 9.3% cycle efficiency, which was almost half of the Carnot efficiency working in the same hot and cold temperatures. They found that the cycle performance increased almost linearly with maximum pressure of the cycle, and it has similar trend with decreasing of minimum cycle pressure. Oralli et al [4] presented a deterministic model for scroll expanders that needs exact geometrical details of the scroll machine. Their model was used to study effect of the volume ratio of the expander on its performance and some modifications were also suggested on the geometry of the expander. They assessed impacts of their modifications on the performance of the system using main geometrical features of scroll expanders. They suggested that R404a is the best fluid when no modifications were implemented on the scroll compressor to work as the expander. However, they evaluated the performance of the system without considering any restrictions of the hot and cold source temperatures and maximum and minimum pressure of the system as well. Muhammad et al [8] used R245fa for their experimental campaign of a non-regenerative ORC using a commercial scroll expander to produce 1 kW net electrical power from low grade steam in the range of 1-3 bar. Selection criteria for the working fluid were hot source temperature, thermodynamic performance of the ORC unit and environmental safety of the organic fluid. Their system could operate with thermal efficiency of 5.75% at its best performance. They reported that superheating of the fluid has deteriorating effect on the thermal performance of the system. The majority of studies used semi-empirical models to simulate main components of the ORC systems. In this study, the preliminary evaluation of working conditions of the system is concerned to understand the expected range of performance of the system before running the test facility.

In a previous paper, some of the authors of this work [9] used a commercial scroll compressor as expander in a non-regenerative ORC unit. R410A was used as working fluid during the experiments, while different organic fluids were considered in the subsequent simulation analysis aiming at producing 10 kW electrical power from low-grade heat source. In particular, they found that R245fa was the best working fluid among those in their simulation, in terms of efficiency of the system.

In this paper, R245fa was used in the simulation analysis because of its suitability in the range of available hot source temperatures considered (maximum 170^{o}C), and its low environmental impacts [10]. Results of the simulations will be compared with experimental data in a near future. Such comparison will reveal the accuracy of the proposed model, which has employed specifications of main components of the system provided by manufacturers.

2. Description of ORC facility

The ORC plant consists of the following main components: three Plate Heat Exchangers (PHEs) made by SWEP Company [11], an in-line pump and a scroll expander. Main characteristics of PHEs are described in Table 1.

An electrical steam generator delivers steam at 7.5 bar and 170^{o}C with a maximum flow rate of 275 kg/h at steady state conditions. Water at 20^{o}C is used as cooling medium in the condenser. Figure 1 displays the P&ID of the ORC facility and the mounted rig. As shown in the figure, pump inlet is considered at the lowest elevation in the test rig, and a liquid receiver is used after the condenser to maintain a NPSH>0 at the pump inlet.

Table 1. Main characteristics of plate heat exchangers

Evaporator Model:B200TH	Number of plates Dimensions of projected area	50 243*448.5 mm2
Recuperator Model:B56H	Number of plates Dimensions of projected area	130 243*430 mm^2
Condenser Model:B427L	Number of plates Dimensions of projected area	90 304*567 mm^2

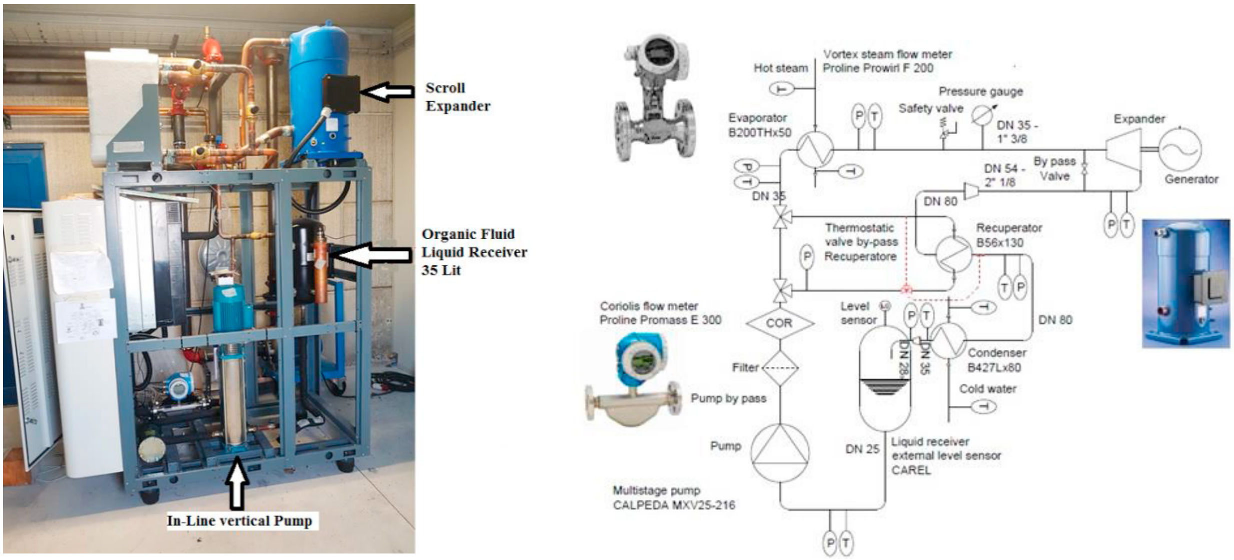

Figure 1. Mounted test rig (left) and P&ID of the ORC test bench (right)

The scroll expander was originally a compressor made by Danfoss [12] for refrigeration applications. Characteristics of the compressor are summarized in Table 2.

Table 2. Main characteristics of the scroll compressor. Danfoss, type SZ380-4

Maximum system test pressure Low Side / High side	25 bar / 41 bar
Maximum differential test pressure	31 bar
Swept volume at compressor outlet (expander inlet)	188 cm^3/rev
Built-in volume ratio	2.8
Net weight	163 kg
Approved refrigerant	R407C

3. Numerical model

Performance of the system is investigated in a certain range of working conditions by modeling the main components of the plant. The following inputs have been considered in the model: (i) inlet temperature and pressure of the hot and cold fluids; (ii) mass flow rate of hot fluid; and (iii) maximum pressure of the ORC. Outlet temperature of the both cooling water and the steam are assumed as controlling variables, thus limiting the number of data required for the evaporator and condenser models.

Data from SSP G7 software provided by the manufacturer were used to evaluate the performance of the evaporator at different working conditions. A linear regression method was used in EES software [13] to predict the heat load of the evaporator as a function of the mass flow rate of the organic fluid, the mass flow rate of the steam, the saturation temperature and the inlet temperature of the organic fluid. Steam inlet temperature and pressure were considered fixed; hence, the correlation was independent from the thermodynamic properties of the steam at the evaporator inlet. Figure 2 shows the regression plot for 4600 data points, with R^2=99.86% and RMS=0.98558.

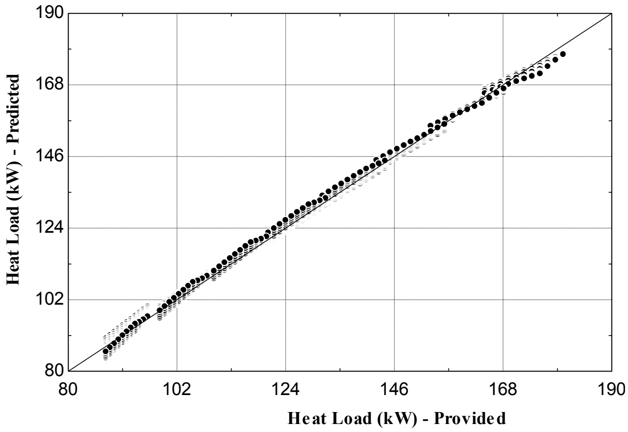

Figure 2. Predicted vs. measured heat load of the evaporator

With reference to the recuperator, the ε-NTU method has been used. The overall heat transfer coefficient (U) of the PHE in different inlet conditions of hot and cold sides was obtained by means of the SSP G7 software. As for the evaporator, a linear regression model in EES software was used to predict the U values. The sensitivity analysis has shown that only mass flow rate of the organic fluid affects the coefficient, and a second order polynomial correlation is able to predict it with good accuracy in the considered range of working conditions.

As regard the condenser, results of the SSP G7 have shown that the PHE is always over-designed when the outlet temperature of the cooling water is limited in the range of 30°C-40°C. Therefore, a correlation was found to predict the over-surfacing of the condenser in the range of working conditions. The performance of the pump has been reported in terms of efficiency and head versus mass flow rate. These data were obtained from the technical specifications provided by the manufacturer of the pump.

The main assumptions of the steady state simulation are:

- Negligible pressure drops and thermal losses in PHEs and piping.
- Evaporation temperature of the organic fluid is considered in the range of 110°C-130°C. In particular, the upper bound is due to limited maximum pressure that can be supplied by the pump, while the lower bound has been set to avoid high temperature difference between the steam and the organic fluid in the evaporator (the saturation temperature of the steam at 7.5bar is around 167.7°C).
- 5°C of sub-cooling of organic fluid at the outlet of the condenser, to assure that fluid enters the liquid receiver at liquid state.
- Temperature difference between cooling water at the inlet of the condenser and the condensing temperature of organic fluid equal to 25°C.

Built-in volume ratio is an inherent characteristic of a volumetric expander. The volume ratio entails a built-in pressure ratio that varies for different fluids and inlet conditions. Since the built-in pressure ratio of the scroll expander in this study is less than the ratio between the maximum and the minimum pressure of the ORC plant, part of the expansion occurs at the exit of the expander, similar to the expansion of gas out of a constant volume. Therefore, scroll expander works in under-expansion mode. As extensively investigated by Lemort et al. [6], under expansion causes a reduction of the isentropic efficiency of the expander, while the output work increases. Their model needs a group of parameters to be identified experimentally to take into account mechanical, thermal, and leakage losses in the performance of the scroll expander. Therefore, it was not possible to use the model in complete form. In this paper, these losses were accounted in the isentropic efficiency of the machine equal to 60%

as average value according to [7, 14, 15]. Equation 1 reports the power output of the scroll expander according to [6].

$$\dot{W}_{exp} = \dot{m}_{ref}\left[\eta_{is}\left(h_{su} - h_{int,is}\right) + v_{in}(P_{int} - P_{out})\right] = \dot{m}_{ref}\left[\eta_{is}\left(h_{su} - h_{int,is}\right) + r_v . v_{su}(P_{int} - P_{out})\right] \quad (1)$$

Where r_v is the built-in volume ratio of the expander, and $h_{int,is}$ corresponds to the specific enthalpy of the fluid at the end of an isentropic expansion between the suction pressure and the pressure obtained from the built-in pressure ratio of the expander. The thermal efficiency of the ORC and the regeneration efficiency of the recuperator were calculated from Equations 2 and 3 respectively.

$$\varepsilon_{th,ORC} = \frac{\dot{W}_{th,net}}{\dot{Q}_H} = \frac{\dot{W}_{th,exp} - \dot{W}_{th,pump}}{\dot{Q}_H} \quad (2)$$

$$\varepsilon_{reg} = \frac{T_{ref,c,out} - T_{ref,c,in}}{T_{ref,h,in} - T_{ref,c,in}} \quad (3)$$

An inverter device mounted on the generator allows the scroll expander to rotate at variable speeds. Rotational speed of the expander can be calculated as in Equation 4.

$$RPM = \frac{\dot{m}_{ref} \times 60000}{D_{ref,su} \times SV_{exp,su}} \quad (4)$$

4. Results and conclusion

The simulation analysis was aimed at finding the best working condition of the ORC for different flow rates of the steam in order to obtain the maximum thermal efficiency and net power. In all the simulations, steam at the outlet of the evaporator has been considered in liquid phase at temperatures < 100^{o}C because of the liquid receiver in the steam loop that works at atmospheric pressure. Outlet temperature of the cooling water was fixed to 35^{o}C in the simulations. This temperature is due to future application of the ORC as CHP unit. Therefore, results of the ORC are one out of a number of possible solutions of the system when it is adapted for future experimental purposes. Figure 3 represents the flow rates of the fluids to satisfy the aforesaid conditions (the steam flow rate is input to the model). Hence, the graph can be used as a guideline to adjust the system for future experiments.

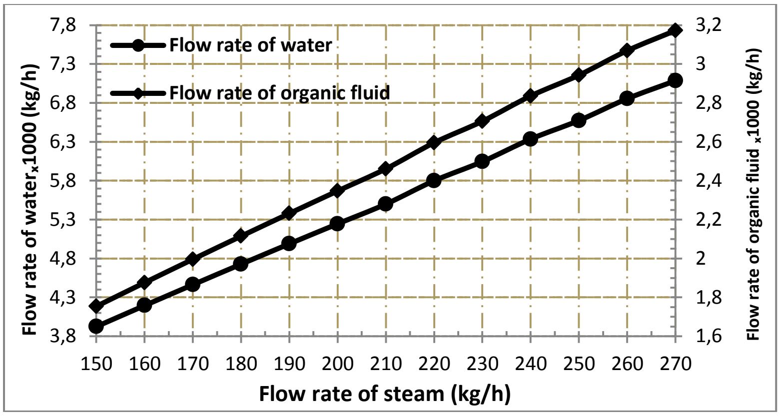

Figure 3. Flow rates of organic fluid and cooling water vs. flow rate of steam

Figure 4a displays the T-s diagram of the cycle with steam flow rate at its maximum thus leading to maximum net power of the system. The point "Internal" corresponds to the outlet pressure of the expander when volume ratio is at its nominal value. Hot and cold source temperatures are shown in the figure using star symbols. Points 3 and 7 are outputs of the recuperator at cold and hot sides respectively. Recuperator efficiency was calculated using Equation 3 in different working conditions, and it was about 65% in all simulations. Figure 4b shows the T-s diagram of the cycle without recuperator for the same flow rate of the steam (270 kg/h). Results showed that both the net power and the thermal efficiency of the system decrease about 16% compared to the regenerated cycle. Regeneration was even more effective when the evaporation temperature of the organic fluid is higher (up to 20% reduction). Therefore, the recuperator plays a fundamental role to improve the performance of the ORC when the system works with high pressure ratios.

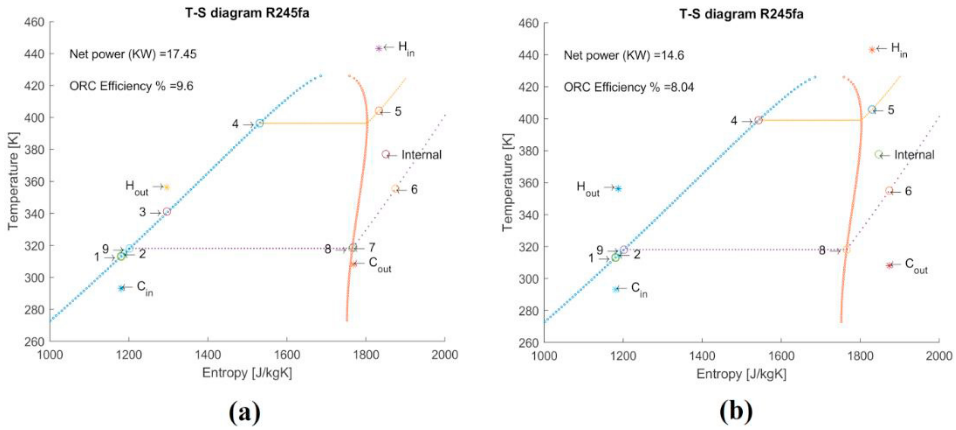

Fig 4. T_S diagram of the ORC at its maximum net power a) with recuperator b) without recuperator

On the left axis of Figure 5, net output power of the ORC is presented with the flow rate of steam. As it can be noticed, the net power of the system increases monotonically with the steam flow rate, while the flow rate of the organic fluid (right axis) has to be increased at the same time to provide enough sub-cooled steam at the outlet of the evaporator. Followed by Equation 4, the rotational speed of the expander is in direct proportion with the flow rate of organic fluid, and it increased from 1100 RPM to about 2350 RPM as the flow rate of steam increased in the range specified in the Figure 5. It is noteworthy that the expander model does not consider the alteration of losses with rotational speed, thus it has been introduced here as indicative. For instance, internal leakage becomes significant when the rotational speed drops below 1500 RPMs [16]. Thus, it is expected to achieve a lower net power than the one reported in the graph for steam flow rate <190 kg/h. Thermal efficiency of the ORC is changed slightly between 9.3% and 9.6% during simulations. Pump efficiency increases with flow rate of the organic fluid from 32% to 43.4%. The condenser also works with about 18% over-surfacing when the flow rate of organic fluid is maximum in the Figure 5, while it goes up to 98% as the flow rate decreases to 1750 kg/h.

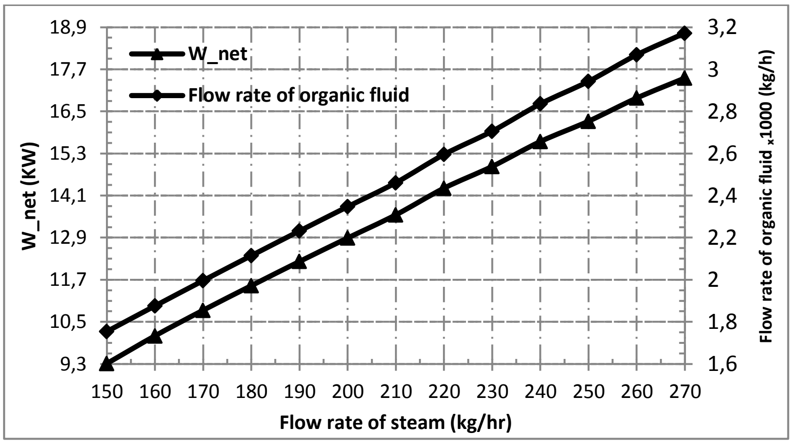

Fig 5. Net power (left axis) and flow rate of the pump (right axis) of the ORC by the flow rate of steam

Results of this study showed adjustments of determinant parameters of the ORC in different mass flow rates of steam considering specific thermodynamic conditions of water at the outlet of the evaporator and condenser. Results showed that even when the steam generator produces steam at maximum flow rate (corresponding to maximum energy in the hot source), main components of the ORC system do not reach to their maximum performance limit; therefore, the system can work with more available energy in the hot source with higher overall performance.

Acknowledgements

The authors kindly acknowledge staff of HiRef Company for their close collaboration. We also appreciate SWEP Company for providing us performance data of their products.

References

[1] M. Kane, D. Larrain, D. Favrat, Y. Allani, Small hybrid solar power systems, Energy, 28 (2003) 1427-1443.
[2] E. Galloni, G. Fontana, S. Staccone, Design and experimental analysis of a mini ORC (organic Rankine cycle) power plant based on R245fa working fluid, Energy, 90 (2015) 768-775.
[3] Ennio Macchi, Organic Rankine Cycle (ORC) Power Systems - Chapter 1. Theoretical basis of the Organic Rankine Cycle, in: Ennio Macchi, Marco Astolfi (Eds.), Elsevier Ltd., 2017.
[4] E. Oralli, Md. Ali Tarique, C. Zamfirescu, I. Dincer, A study on scroll compressor conversion into expander for Rankine cycles, International journal of low-carbon technologies, 6 (2011) 200-206.
[5] Sébastien Declaye, Sylvain Quoilin, Ludovic Guillaume, Vincent Lemort, Experimental study on an open-drive scroll expander integrated into an ORC (Organic Rankine Cycle) system with R245fa as working fluid, Energy, 55 (2013) 173-183.
[6] Vincent Lemort, Sylvain Quoilin, Cristian Cuevas, Jean Lebrun, Testing and modeling a scroll expander integrated into an Organic Rankine Cycle, Applied Thermal Engineering, 29 (2009) 3094-3102.
[7] Sylvain Quoilin, Vincent Lemort, Jean Lebrun, Experimental study and modeling of an Organic Rankine Cycle using scroll expander, Applied Energy, 87 (2010) 1260-1268.
[8] Usman Muhammad, Muhammad Imran, Dong Hyun Lee, Byung Sik Park, Design and experimental investigation of a 1 kW organic Rankine cycle system using R245fa as working fluid for low-grade waste heat recovery from steam, Energy Conversion and Management, 103 (2015) 1089-1100.
[9] Maurizio Cambi, Roberto Tascioni, Luca Cioccolanti, Enrico Bocci, Converting a commercial scroll compressor into an expander: experimental and analytical performance evaluation, Energy Procedia, 129 (2017) 363-370.
[10] E.H. Wang, H.G. Zhang, B.Y. Fan, M.G. Ouyang, Y. Zhao, Q.H. Mu, Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery, Energy, 36 (2011) 3406-3418.
[11] https://www.swep.net.
[12] https://www.danfoss.com/en/.
[13] http://www.fchart.com/ees/.
[14] Manolakosa D, Papadakisa G, Kyritsisa S, Bouzianasb K, Experimental evaaluation of an autonomous low-temperature solar Rankine cycle system for reverse osmosis desalination, Desalination, 74 (2007).
[15] Peterson RB, Wang H, Herron T, Performance of a small-scale regenerative Rankine power cycle employing a scroll expander, Proc. IMechE, Part A: J Power Energy, 82 (2008) 222-271.
[16] T. Yanagisawa, M. Fukuta, Y. Ogi, T. Hikichi, Performance of an oil-free scroll-type air expander, Proc. IMechE Conference on Compressors and their Systems, (2001) 167-174.