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Experimental Study of a 1-kW Organic Rankine Cycle Using R245fa Working Fluid and a Scroll Expander: A Case Study

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ABSTRACT In an organic Rankine cycle (ORC), the study of the cycle efficiencies and the turbine is essential to know the performance in the generation of electrical energy. The proper selection of a working fluid is relevant, because it must be environmentally friendly and compatible with the ORC plant. This article presents an experimental study for the analysis of the cycle and thermal efficiencies on a 1-kW ORC system and the isentropic efficiency of its scroll expander. The test was performed on a 1-kW ORC with scroll expander system with R245fa as the working fluid. Furtheremore, it was considered a working temperature below 100 °C, which is used in waste heat recovery systems, to determine the performance of the ORC. The enthalpy is estimated with the *Coolprop* software. For estimating the cycle and thermal efficiency, the net power and the thermal power, which are supplied to evaporate the working fluid, were considered. The isentropic efficiency of the expander was calculated by the scroll mechanical work and the hydraulic work at the scroll expander. The results show that the mean isentropic efficiency of the fluid in the prototype test for ORC in the generation of 1000 W was about 60%, a promising value for the generation of electrical energy using the residual heat from industry.

INDEX TERMS Energy conversion, fluids, power generation, power systems, waste heat.

NOMENCLATURE

η_{cycle}	Cycle efficiency (%)
η_e	Electrical efficiency of the generator (%)
η_{he}	Heat exchanger efficiency (%)
η_{st}	Isentropic efficiency of the scroll expander (%)
η_{th}	Cycle thermal efficiency (%)
CAMD	Computer-aided molecular design
Cp_{oil}	Specific heat of the thermal oil ($J/g°C$)
HFC	Hydrofluorocarbon
$h_{p,i}$	Working fluid enthalpies to pump input (kJ/kg)
$h_{p,o}$	Working fluid enthalpy at the high-pressure pump output (kJ/kg)
$h_{s,i}$	Working fluid enthalpy to the expander input (kJ/kg)
$h_{s,o}$	Outlet working fluid enthalpies from the expander (kJ/kg)
I_{ac}	Generated alternating current (A)
m_f	Mass flow of working fluid (kg/s)
m_{oil}	Mass flow of thermal oil (kg/s)
ORC	Organic Rankine cycle
Q_{cond}	Condenser thermal power (W)
Q_{evap}	Thermal power in the evaporator (W)
Q_{he}	Thermal power of the thermal oil (W)
s	Entrophy (kJ/kgK)
T	Temperature (°C)
$T_{eh,i}$	Input temperature in the heat exchanger (°C)
$T_{eh,o}$	Output temperature in the heat exchanger (°C)
V_{ac}	120 volts of alternating current

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W_{gen}	Electrical power output (W)
W_m	Scroll mechanical work (J)
W_{net}	Net power (W)
W_p	Power of the high-pressure pump (W)
W_t	Output power measured in the expander (W)
W_T	Hydraulic work (J)

I. INTRODUCTION

In recent years, the demanding level of industrial energy in the world has been increasing due to the advance and development of its processes. For this reason, the energy transition has made the researchers to perform works for fossil fuel reduction using alternative energies [1]. Furthermore, the thermal energy processes are mainly supplied by fixed sources such as incinerators, fire boilers, drying furnaces, exo-thermal processes, purge or exhaust of vapor systems by chimneys, generating the possibility to reuse the residual heat to reintegrate it in the processes, obtaining fossil fuel savings and generating electric energy [2], [3]. However, waste generation energy is increasing and if it is not treated well, it may occur harmful effects on health, environment and social economy [4].

One of the most efficient techniques that has demonstrated advances in the residual heat reuse, is the one which integrates the organic Rankine cycle (ORC) [5]–[8], because it is possible to use in a stable manner, heat sources lower than 100 °C in order to heat a working fluid, taking advantage of its mechanical energy and transforming it into electric energy [9]. The general ORC process has three main stages: the heat source (heating), the conversion system of electric thermal energy (generating) and the heat sinking (condensation) [10], [11].

The generation stage is found within an expander element, which can be classified in two categories: in the turbine (of radial or axial flow) or in the positive displacement devices, such as screw expanders, piston expanders, rotating blades and scroll expander [12]–[14]. Because of its high-efficiency properties, high-pressure ratio, relatively low flow coefficient, the working chamber symmetric design, lower rotation speed and its low manufacture cost, the scroll expanders are mostly used in ORC plants that may generate up to 10 kilowatts of electricity. These expanders can improve the thermodynamic performance due to the opportunity to represent more efficiently, the temperature field and internal asymmetric flow in an ORC [15].

However, fluid selection in ORC systems is important in sizing to improve efficiency in electric power generation. The ideal organic fluid is in which its thermodynamic proprieties are directly related to the heat-sources, resulting in optimal at higher temperatures [16]. Theoretically, this fluid reduces latent heat vaporization by maximizing evaporation pressure and decreasing the amount of overheating [17]. Current studies have shown that it is possible to design or improve a refrigerant by manipulating the molecular structure of existing work fluids to increase performance and efficiency in ORC [18].

One of the most common techniques is using the computer-aided molecular design (CAMD), which simulates the modeling of working fluid with desired parameters which may improve thermal and economic performance in power generation processes [19]. Furthermore, the optimization in the performance of working modelling by CAMD could minimize the specific investment cost of ORC systems [20]. White *et al.* [21] found that the work fluids used to maximize the output power of ORC systems require a higher heat-exchanger area when maximizing the power output from ORC systems is required.

As an alternative to the use of CAMD, prediction work has demonstrated by correlating two fundamental properties of fluids: the maximum efficiency of use and the optimum temperature of the heat source; as a finding, Lukawski *et al.* [22] proposed with this method that the best candidates as working fluids correspond to the group of hydrofluorocarbons (HFCs) for the best performance of the efficiency with the heat source.

The organic fluids are the most used and commonly employed in the refrigeration industry, such as the HFC: R134a, R404a and R245fa [8], [23]. The R245fa shows most thermal efficiency up to approximately 4.6%. Moreover, in comparison of other fluids, such as the R1234ze and R1233zd, R245fa may obtain up to 11.4% more electrical energy [24]. According to conducted studies using R245fa, it exhibits an excellent performance as working fluid in ORC systems, due mainly to the fact that they have demonstrated having a good performance, reaching efficiencies up to 71% [25] and have surpassed other fluids such as the R134a, propane and isobutane due to its safety, ample range temperature, allowing to reach cycle efficiencies higher at 10% with approximate pressures of 13.5 bar [26], [27].

Agromayor and Nord [28] proposed a fluid detection analysis approach which relates variables such as the critical working fluid temperature and the heat source input to determine the optimal fluid function of the critical temperature and the boiling point. Similarly, Quoilin *et al.* [29] compared various working fluids performance through specific elements models, e.g., the expander with the Computerized Fluid Dynamic (CFD) method; in order to obtain the adiabatic and mechanical efficiency and to develop a numeric model, which commonly is valid at an experimental level.

Yang *et al.* [30] demonstrated that the R1233 fluid can replace the R245fa only if the plant vapor is optimized by comparing the performance of these working fluids in a scroll expander, because both fluids present a similar performance in the cycle efficiency. Therefore, the evaporator has to be re-sized according to the R1233zd.

Lemort *et al.* [31] tested a scroll expander model that was adapted to an air compressor with open activator and obtained the 68% of maximum isentrop efficiency by using the R123 as working fluid. The efficiency measured by a numerical study suggests that the internal leaks, the output pressure drop and

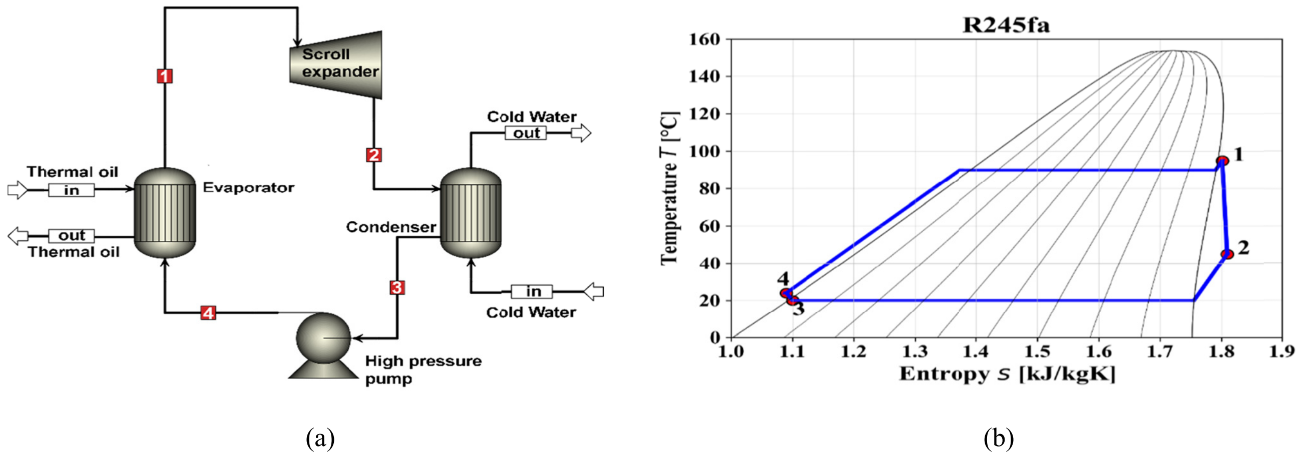

FIGURE 1. a) ORC plant schematic and b) the operation represented in Temperature-entropy (T-s) diagram for R245fa working fluid.

the mechanical missing were the three main roles which deteriorate the scroll performance.

The state of art suggests that it is possible to integrate a scroll expander with a refrigerant fluid while is utilized a small generating plant (up to 10 kW). R245fa and R1233zd fluids could present a better performance because they have a similar cycle efficiency for generating electric energy starting as a fixed industrial heat source.

The modelling of ORC plants are even suitable for various applications which consider thermal energy; Girgin and Ezgi [19] studied the modelling of ORC system for naval surface ship estimating the net power that could work with seven different working fluid, obtaining electric power from 76.3 to 117.6 kW.

This article presents an experimental study for the analysis of the cycle and thermal efficiencies on a 1-kW ORC system and the isentropic efficiency of its scroll expander. It was considered the scroll mechanical work and the hydraulic work at the expander in order to obtain a broad approach of the expander isentropic efficiency. The test was performed on a 1-kW ORC with scroll expander system using R245fa as working fluid. It were considered working temperatures below of 100 °C to determine the performance of the ORC which are used in waste heat recovery systems. The enthalpies calculation in 4 points of the ORC was done. The results show a medium isentropic efficiency during a 1000 W generation testing in the prototype plant.

II. THEORETICAL CONSIDERATIONS

The waste heat systems in the industry offer enormous powers intermittently, therefore one way of using this energy is to linearize it through thermal storage at a lower temperature approximately from 80 to 100 °C [32]. The ORC consists in using a high molecular mass working fluid which causes a drop to a lower enthalpy within the turbine, generating a lower turbine peripheral speed in temperatures below 150 °C [33]. The expansion is caused by thermal energy recovery from an external heat source, such as the exhaust gases. The condenser could be cooled by air, water or a combination of both. The ORC consists mainly of three fundamental stages [34]: the fluid heating, the generation of energy and the fluid condensation. In the fluid heating stage, the thermal energy is obtained from a heat-fixed source of thermal oil (which simulates the residual industrial heat) through heat exchangers causing the working fluid temperature to increase to its boiling point. The generation of energy stage contains a scroll expander coupled to an electric generator to take advantage of the mechanical energy of the fluid at a vapor state to generate energy. Finally, the condensation stage cools the fluid returning it to its liquid stage to begin a new cycle.

Fig. 1(a) shows the schematic for an ORC using R245fa as working fluid and its operation represented in T-s diagram in Fig. 1(b). The working fluid follows the next processes:

–Process 1-2: The isentropic expansion of the refrigerant fluid is made within the expander. The vapor has sufficient pressure to turn the scroll expander. Afterward, the refrigerating vapor loses pressure from the superheated vapor pressure to the saturated vapor pressure [35], [36].

–Process 2-3: Through the heat exchange between the refrigerating agent and the condenser, heat is extracted at constant pressure up until the refrigerating agent is taken to a saturated liquid condition.

–Process 3-4: The isentropic process is performed in the high-pressure pump, where the heat exchange is not considered by working fluid [32]. The pump directs the working fluid to the evaporator in order to increase the fluid working pressure to the evaporation saturation pressure [33]. The saturated liquid is taken until the working fluid enters into the evaporator by the pump, which develops a compression isentropic or adiabatic process.

–Process 4-1: The isobaric process is the working fluid expansion in the evaporator. It is produced by the thermal energy from the heat source to the evaporator at constant pressure.

A. CYCLE EFFICIENCY

In order to obtain the net cycle efficiency η_{cycle} and ignore the heat and pressure loses, it should be taken into consideration the net power of the ORC W_{net} and the thermal power that the organic fluid acquires in the evaporator Q_{evap} supplied by the heat source as follows [29]:

$$\eta_{cycle} = \frac{W_{net}}{Q_{evap}} \tag{1}$$

W_{net} is obtained by:

$$W_{net} = W_t - W_p \tag{2}$$

whereas W_p is the power supplied by the high-pressure pump though the working fluid W_p and W_t the output power measured in the expander. The expression to obtain Q_{evap} is [37]:

$$Q_{evap} = m_f \left(h_{s,i} - h_{p,o}\right) \tag{3}$$

where m_f is the mass flow of working fluid, $h_{s,i}$ is the working fluid enthalpy to the expander input and $h_{p,o}$ is the working fluid enthalpy at the high-pressure pump output. W_t

B. THERMAL EFFICIENCY

When thermal energy is absorbed from the heat source, the working fluid is evaporated, allowing superheated vapor production when fluid quality is 1.

The heat exchanger efficiency η_{he} can be estimated as follows [38]:

$$Q_{he} = m_{oil} Cp_{oil} \left(T_{eh,o} - T_{eh,i}\right) \tag{4}$$

where m_{oil} is the mass flow of thermal oil, Cp_{oil} is the specific heat of the thermal oil, $T_{eh,o}$ is the output temperature in the heat exchanger and $T_{eh,i}$ is the input temperature in the heat exchanger. The heat exchanger efficiency η_{he} is defined by [39]:

$$\eta_{he} = \frac{Q_{evap}}{Q_{he}} \tag{5}$$

The condenser thermal power Q_{cond} is given by:

$$Q_{cond} = m_f \left(h_{s,o} - h_{p,i}\right) \tag{6}$$

where $h_{s,o}$ is the outlet working fluid enthalpies from the expander, while $h_{p,i}$ is the working fluid enthalpies to pump input. The cycle thermal efficiency η_{th} is determined by the energy taken from the evaporator and the electrical power output W_{gen}. This relation could be obtained from the heat exchangers power and the condenser, described by [40].

$$\eta_{th} = \frac{W_{gen}}{Q_{evap}} \tag{7}$$

C. ISENTROPIC EFFICIENCY OF THE SCROLL EXPANDER

The electric power output of the generator W_{gen} is estimated by:

$$W_{gen} = I_{ac} V_{ac} \tag{8}$$

where I_{ac} is the generated alternating current in amps (A) and 120 volts of alternating current (V_{ac}) of the expander [41].

In this particular study, the isentropic efficiency of the scroll expander η_{st} was calculated by the scroll mechanical work W_m and the hydraulic work W_T, expressed by:

$$\eta_{st} = \frac{W_m}{W_T} \tag{9}$$

The mechanical work W_m is calculated by multiplying the generator electrical efficiency η_e got from generator technical data and W_{gen} then:

$$W_m = \eta_e W_{gen} \tag{10}$$

W_T is the hydraulic work at the scroll expander. The energy and mass balance are reduced, to get the work according to the mass flow through the expander is calculated by [42]:

$$W_T = m_f \left(h_{s,o} - h_{s,i}\right) \tag{11}$$

III. METHODOLOGY

Fig. 2 shows the diagram of the piping and instrumentation (P&ID) of the ORC prototype installed in the Universidad Autónoma de Querétaro. The evaporator (EVAP-300) was designed to operate with thermal oil as a heat source from 90 to 150 °C at 95 liters per minute (l/min) flow rate, 20 °C temperature differential between the boiler (CAL-800) and the refrigerating agent R245fa for this case. The refrigerating agent shall be made to circulate by a high-pressure pump (BAP-100) with sliding blades at a rate of 0.08 kg/s and a 10.8 bar discharge pressure. The pump is placed under the liquid receptor (TLR-600) with a static increase of 1 m between the axle of the pumps and the receptor liquid in order to avoid air bubbles. The evaporation temperature in the system is achieved at 110 °C and the evaporation pressure at 11.8 bar.

The maximum operating pressure of the scroll expander is 13.5 bar. The condenser work flow (COND-500) is water-based. The condensation temperature is from 21 to 30 °C at condensation pressure from 1.8 to 2.4 bar. The relief valve is located at the evaporator outlet (EVAP-300) to avoid over-pressure fitted to 12.6 bar. Furthermore, there is a safety and relief valve adjusted at 12.8 bar in the liquid receptor. Finally, the liquid receptor unloading capacity is 1.5 l/s.

Fig. 3 exhibits ORC experimental prototype which integrates the three stages: on the left side is the heating equipment which consists of a thermal oil heater with a 30,000 kcal/h thermal power; in the middle an experimental ORC is shown with a capacity of up to 1 kW and on the right side the cooling tower with a nominal caudal of 56.7 l/min.

For regulating the scroll expander input pressure, a proportional-integral-derivative (PID) controller is integrated into the ORC which activates a globe valve directly with frequency retro-feeding to maintain 60 Hertz (Hz) in the generator. The maximum input pressure is integrated into a PID control in the ORC to regulate the input pressure to the scroll expander. The maximum input pressure of the expander is 12.6 bar; it is crucial to surpass this value in order not to damage the equipment; therefore, an input regulating control

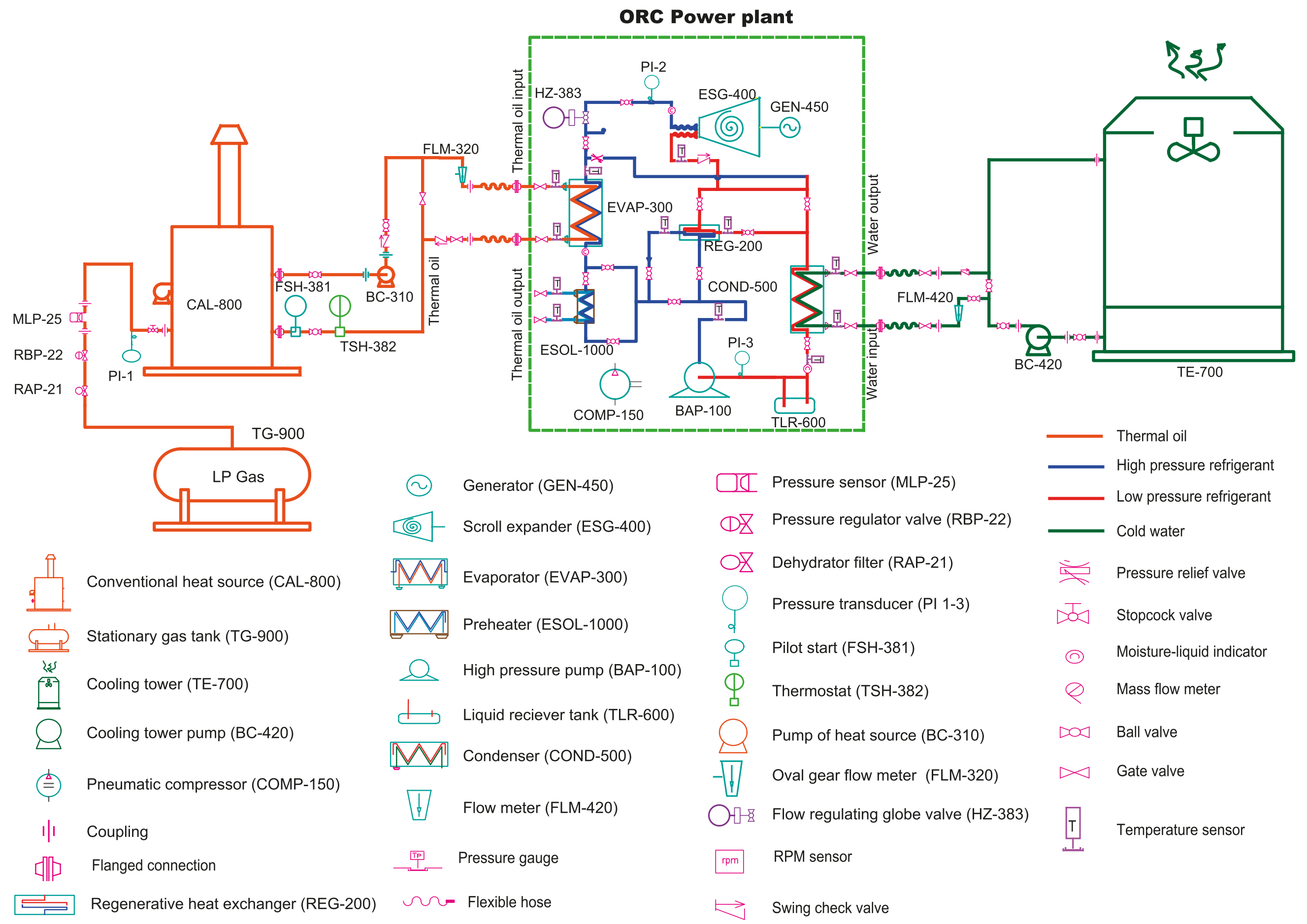

FIGURE 2. Instrumentation diagram and piping (P&ID) of ORC.

FIGURE 3. Electric energy generating ORC plant prototype.

is be implemented. The scheme of the controller is described as per Fig. 4.

The study case is defined with the following operation conditions: 90 °C for operation temperature in the evaporator, maintaining a fixed setting the heat reject at 21 °C.

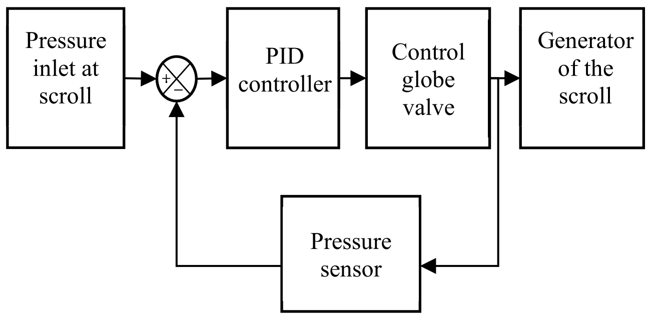

FIGURE 4. Pressure control strategy.

System behavior data is recorded automatically. The saturated vapor pressure is maintained fixed by the PID control. The experimental data is analyzed in order to get R245fa performance models. Enthalpies of the cycle are calculated using *CoolProp* library from Phyton considering temperature and pressure. These models allow to estimate the expander, evaporator and condenser input enthalpies of R245fa high pressure refrigerating fluid in each cycle point. The enthalpies values in four different cycle points are shown in Table 1.

TABLE 1. Enthalpy values in the cycle.

Point	Description		Enthalpy (kJ/kg)
1	expander inlet - evaporator outlet	$h_{s,i}$	480
2	condenser inlet - expander outlet	$h_{s,o}$	430
3	pump inlet - condenser outlet	$h_{p,i}$	230
4	pump outlet - evaporator inlet	$h_{p,o}$	230

TABLE 2. Design parameters of main elements in the experimental setup.

Element	Parameter	Value
Scroll expander (ESG-400)	Efficiency	0.6
	Mass flow of working fluid	0.08 kg/s
	Max speed of the scroll	3600 RPM
	Inlet pressure	11.8 bar
	Inlet temperature	110 °C
	Outlet pressure	5.1 bar
	Outlet temperature	60 °C
Evaporator (EVA-300)	Evaporation temperature	95 °C
	Evaporation pressure	11.66 bar
	Evaporator inlet temperature	30 °C
	Thermal power of the evaporator	14 kW
	Thermal oil caudal	1.27 kg/s
	Oil temperature at the evaporator inlet	120 °C
	Oil temperature at the evaporator outlet	100 °C
Condenser (COND-500)	Condensing temperature	30 °C
	Condensing pressure	1.8 bar
	Organic fluid inlet temperature at condenser	45 °C
	Thermal power of the condenser	17.5 kW
	Heat transfer area	3.06 m^2
	Heat transfer fluid condenser side	Water
	Cooling water flow	0.94 kg/s
	Inlet temperature to the cooling water condenser	28 °C
High pressure pump (BAP-100)	Suction pressure	1.36 bar
	Discharge pressure	11.56 bar
	Pump flow	0.1 L/s
	Electric motor power	260 W
Temperature sensor	Accuracy	≤0.3%
Pressure sensor	Accuracy	≤0.03%
	Total performance (including error and repeatability)	≤0.09%
Flow sensor	Accuracy	±0.1%
	Repeatability	±0.05

The critical parameters for the experimental setup design and the instruments' technical specifications are shown in Table 2.

IV. RESULTS

According to the experimental testing performed with organic working fluid R245fa, results were obtained in a stable state operation point. The operation point is reported in the results matrix in Table 3.

Fig. 5 shows the expander generated speed regarding the input pressure in the expander output. The vapor generation coming from the refrigerating agent is given by the mass flow coefficient of 0.04 kg/s. The supply pressure is adjusted by the proportional flow globe valve and operated through a PID controller. The controller set point is the input pressure.

TABLE 3. Experimental results at selected operating point.

Variable	T_{env} $[°C]$	P_e $[kPa]$	T_e $[°C]$	m $\left[\frac{kg}{s}\right]$	w_{gen} $[W]$	η_s $[\%]$
Result	17	1200	83	0.04	820	8

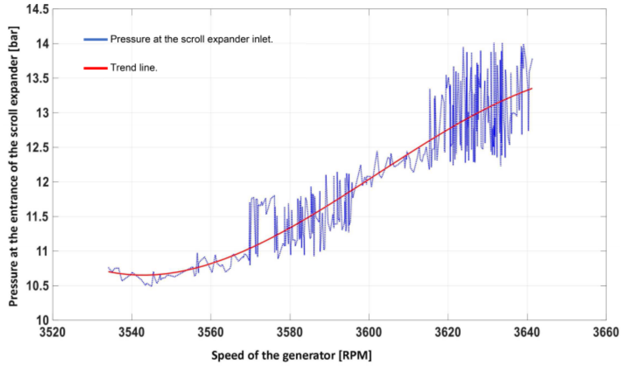

FIGURE 5. The relationship between speed angle and scroll expander pressure.

The maximum input pressure to the expander is 13.4 bar. Furthermore, the scroll expander works at 3600 rpm at about 15 bar, generating 127 Vac at 60 Hz electrical energy and providing 820 W to an inductive load. The test is performed setting the temperature point at 95 °C in the heat source to evaluate the performance according to thermal power stored from heat waste recovery systems.

In order to know the efficiency cycle performance, the expander temperature input is analyzed. The temperature input is a restriction condition in the cycle optimization process when the energy supply comes from fixed heat sources. A temperature interval limits the maximum cycle efficiency into the system under normal conditions, as well as the optimization parameters in other points into the system, such as the condenser. The maximum cycle efficiency in the prototype is 8.29% at 96.6 °C. At higher temperatures, the cycle efficiency increases to get its critical point at 175 °C, which is the maximum inlet temperature in the scroll expander. At this specific point of temperatures, the cycle efficiency tends to increase while the temperature rises. This shows that thermal storage systems could reach an efficiency of approximately 10% due to the working temperature range. If the temperature is kept constant, the efficiency of the cycle does not change because the pump power and the output power are constant according to equation (1). Fig. 6 shows the performance of the cycle efficiency temperature at the outlet of the evaporator (inlet of the scroll expander) and the outlet of the scroll expander.

Fig. 7 shows the relationship between the cycle efficiency and the pressure ratio (pressure drop) measured by the pressure sensors located before and after the expander. It may be observed that as the pressure drop increases the cycle efficiency decreases.

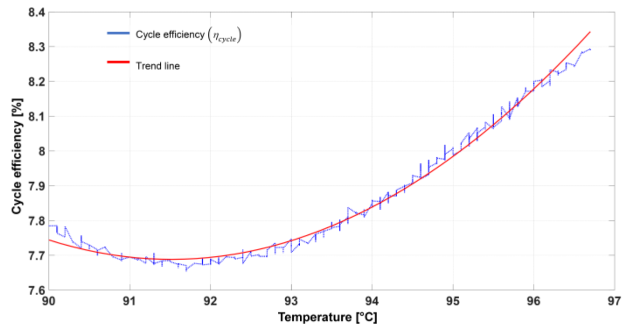

FIGURE 6. **Relationship of the cycle efficiency and expander working flow temperature.**

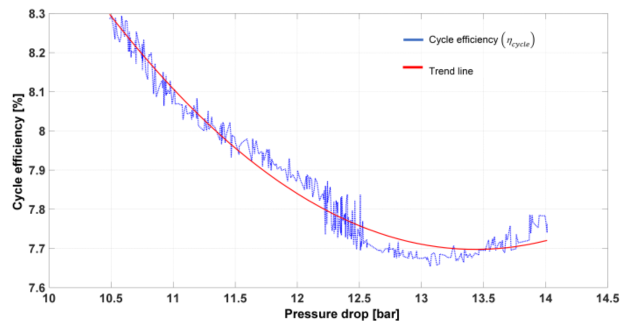

FIGURE 7. **The relationship between fluid pressure drop and cycle efficiency in the system.**

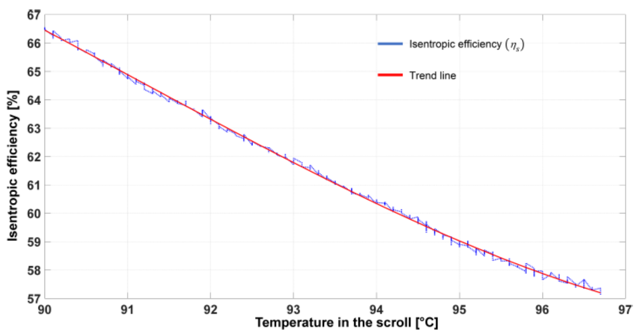

FIGURE 8. **The relationship between inlet temperature the working fluid at the inlet of the scroll expander and isentropic efficiency of the scroll expander.**

The thermal efficiency cycle η_{th}, based upon the gathered data. Also can be obtained, based upon the gathered data. The isentropic efficiency of the scroll expander η_s describes the amount of generated power concerning the amount of power supplied. Then it is possible to know how much-supplied energy is used for electricity generation; this correlation can be observed in Fig. 8, where it is shown that when the fluid temperature increases, the expander isentropic efficiency decreases, i.e., the higher the fluid temperature, the less energy is consumed by the system while maintaining the electrical energy output in the generator, as equation (9); the temperatures are acquired between the outlet of the evaporator and inlet of the scroll expander.

V. CONCLUSION

In this research, the isentropic efficiency of the scroll expander is analyzed using the R245fa refrigerating as the working fluid. The experimental test wis performed in 1-kW ORC prototype in four thermodynamic points: In the expander inlet and outlet, and the pump inlet and outlet.

This study estimated the scroll expander mechanical work considering the generator electrical efficiency and the power generated by the ORC. The mechanical work allowed to show the performance of the isentropic efficiency of the scroll expander according to the temperature of the input vapor. The isentropic efficiency of the scroll expander decreased when the temperature input of the working flow had been increased. In the ORC prototype plant, the R245fa average isentropic efficiency was 61.06% in the expander without considering electromagnetic loses in the generator.

With the acquired data, it was possible to estimate the optimal performance pressure within the operational conditions defined. The PID control set the revolutions according to the optimal pressure point. If pressure setpoint change, the revolutions will be unstable.

The isentropic efficiency increased as pressure decreased, therefore it was necessary to reduce the condensation coefficient, which could be achieved by reducing the water flow in the cooling tower by decreasing the speed of the cooling system pump.

Finally, the experimental study of the isentropic efficiency of the scroll expander by estimating the mechanical work and the hydraulic work within the 1-kW ORC plant from 85 to 100 °C allowed to observe that the generation system could adapt a preheating stage of the thermal fluid by an alternating energy source such as solar, since photovoltaic systems heat up to 60 °C on a typical day, so the fixed source thermal energy supply could be reduced.

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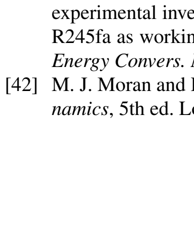

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