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Experimental Characterization of Commercial Scroll Expander for Micro-Scale Solar ORC Application: Part 1

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Abstract: In order to reduce greenhouse gas emissions and achieve global decarbonisation, it is essential to find sustainable and renewable alternatives for electricity production. In this context, the development of distributed generation systems, with the use of thermodynamic and photovoltaic solar energy, wind energy and smart grids, is fundamental. ORC power plants are the most appropriate systems for low-grade thermal energy recovery and power conversion, combining solar energy with electricity production. The application of a micro-scale ORC plant, coupled with Parabolic Trough Collectors as a thermal source, can satisfy domestic user demand in terms of electrical and thermal power. In order to develop a micro-scale ORC plant, a commercial hermetic scroll compressor was tested as an expander with HFC-245fa working fluid. The tests required the construction of an experimental bench with monitoring and control sensors. The aim of this study is the description of the scroll performances to evaluate the application and develop optimization strategies. The maximum isentropic effectiveness is reached for an expansion ratio close to the volumetric expansion ratio of the scroll, and machine isentropic effectiveness presents small variations in a wide range of working conditions. The filling factor is always higher than one, due to leakage in the mechanical seals of the scroll or other inefficiencies. This study demonstrates that using a commercial scroll compressor as an expander within an ORC system represents a valid option for such applications, but it is necessary to improve the mechanical seals of the machine and utilize a dedicated control strategy to obtain the maximum isentropic effectiveness.

Keywords: scroll expander; solar ORC; micro-scale; experimental characterization; isentropic effectiveness

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1. Introduction

The European Green Deal reflects the EU's commitment to addressing the urgent challenges posed by climate change, proposing climate neutrality by 2050 across various sectors, including buildings, biodiversity, energy, transport and food [1].

In the past few decades, the climate impact of traditional energy sources and the need for efficient uses of energy have increased the development of new strategies and technologies for clean energy generation. One of the studied and proposed technologies is waste heat recovery and renewable energies application combined with thermodynamic conversion in electric energy. ORC (Organic Rankine Cycle) system is the most efficient system for low-grade thermal energy conversion, with results that are particularly suitable for distributed generation and for a wide range of different heat sources at different temperature levels [2].

One of the major advantages of an Organic Rankine Cycle system is its use starting from different heat sources; it can refer to the simple heat waste deriving from an internal combustion engine, as in the review of Shi et al. [3], or from different industrial processes.

An ORC system is also very interesting if connected to renewable thermal energy production systems, such as concentrated solar energy with parabolic through collectors for medium-temperature application [4,5] and geothermal energy [6], or maybe in combination with a dry gasification oxy-combustion with CO_2 as Kirtania and Shilapuram studied [7].

Furthermore, a solar Organic Rankine Cycle results in one of the most reliable renewable energy-based technologies to satisfy major energy demands, with poly-generation systems that can generate various useful energy outputs [8].

There are many types of Organic Rankine Cycle architectures [9]. To define various types of plant, it is necessary take into account several boundary conditions for isentropic efficiencies and pinch point temperature differences, heat source temperatures between 20 °C and 450 °C and technical characteristics for each modified cycle. Matching the temperature profiles of the waste heat carrier and the working fluid, a variety of advanced cycle architectures are proposed [9]

Golonis et al. [10] focused their research on the combination of an ORC engine with a CPVT (Concentrating Photovoltaic Thermal) system to identify the feasibility of utilizing the heat of the cooling circuit, which is mostly used for heating purposes with DHW (domestic hot water). The results prove that the coupling of ORC with CPVT systems is an affordable alternative for converting the cooling heat to electricity and, thus, increasing the CPVT productivity.

In a range of 80–200 °C, Organic Rankine Cycles are considered one of the most suitable technologies for converting low-temperature heat into work [11].

Micro-ORCs powered by solar energy represent one of the most effective tools for distributed generation and PED (Positive Energy District). Over the years, many advances have been made, both in the field of solar collectors [12–14] and in thermal storage for residential applications [15]. All of this makes micro-ORC systems even more attractive.

Furthermore, the scroll-type expander is a promising candidate for micro-scale (<10 kW) trans-critical CO_2 waste heat recovery power systems. Du et al. [16] analyzed various flank clearance sizes of the expander, highlighting an improvement in exergy efficiency with greater flank clearance.

For a relevant overview of the working fluids used to recover low-temperature heat sources, Lakew and Bolland [11] carried out interesting research on different working fluids, analyzed on the basis of power production capability and component (heat exchanger and turbine) size requirements. They considered working fluids of R134a, R123, R227ea, R245fa, R290 and n-pentane in a Rankine simple cycle. For temperatures around 160 °C, R24fa produces a higher work output. To obtain maximum power, the optimal evaporator pressure is found, which depends on the temperature of the heat source and the working fluid used. Furthermore, as this temperature increases, fluids with high vapour pressure values have greater working capabilities at higher pressures, and all of this increases the power supplied.

The expander is a key component in an ORC system; depending on the operating conditions and the size of the system, the most suitable type of expander is selected, mainly dynamic (turbo) and displacement (volumetric). Displacement-type expanders are more relevant for ORC applications, particularly for their main technical characteristics, such as lower flow rates, higher pressure ratios and much lower rotational speeds, compared to turbo-machines [17]. Among the different volumetric machines, the scroll-type expander seems to be the most suitable in ORC systems. This is mainly due to the reduced number of moving parts, its reliability and its wide output power range [18].

There are several relevant theoretical and experimental studies on the use of the scroll expander within these systems.

Tsai et al. [19] focused on realizing a micro-scale ORC plant to demonstrate that the 0.3 kW micro-ORC can achieve similar working output as predicted in a large-scale ORC. The performance of an ORC system with R1233zd-E as the working fluid and a scroll expander was experimentally investigated under different operating conditions, such as changing the system pressure differential, expander rotational speed, expander inlet superheat and heat source flow rate. The results obtained show optimal efficiencies of 7.1% and 87.9% for the ORC system and the expander, respectively.

Lemort et al. [20] carried out a theoretical analysis of a scroll expander for the construction of a semi-empirical model to calculate the mass flow rate, the shaft power and exhaust temperature, with eight parameters.

Gao et al. [21] studied different displacements of scroll expanders with thermodynamic and heat transfer models. For a given heat source temperature of 105 °C, experiments and simulation on an ORC are carried out. To obtain the isentropic efficiency of the scroll expander modified from an automobile air condition compressor, experiments on a test rig driven by compressed air are conducted. Subsequently, thermodynamic and heat transfer models of ORC are set on the basis of the obtained isentropic efficiency. Based on the simulation, an ORC plant is constructed and investigated. Experiments show that the energy efficiency of the system ranges from 1.7% to 3.2%, and the exergy efficiency of the system is 8.6% and 16.9%.

Kosmadakis et al. developed [22] an open-drive scroll used as an expander in an ORC system powered by heat up to 100 °C, with a capacity of 6 kW. With several laboratory tests in off-design conditions, the expander showed low volumetric efficiency, about 35%, which can be explained by the very low speed with significant internal fluid leakages and the consequent reduction in power production. This is the main reason that the expansion efficiency was low (about 25–40%), coupled with the low electrical efficiency of the expander's induction generator. Despite these issues, the net power production was positive, and the ORC thermal efficiency was in a range of 0.6 up to 1.7%, acceptable values under the utilized test conditions.

More recently, studies have also been conducted on these types of expanders from a fluid dynamics point of view; for example, Wei et al. [23] examined a scroll expander for small-scale waste recovery systems. They concluded that the flow losses in the suction process of the scroll expander have great effects on the expander efficiency. For a given geometrical structure of the scroll expander, transient performances were obtained, and the flow mechanisms of the vortices existing in the suction and expansion chambers were revealed. A deep understanding of the suction process of a scroll expander provides essential guidance for the optimal design of the suction port and the scroll top profile.

Emhart et al. [24] focused on the CFD model of small-scale ORC scroll expanders, using variable wall thickness. They found, through validation of the models, an optimum performance point at a pressure ration of 3.5 and that the decrease in radial clearance had a significant effect on the isentropic efficiency and the specific power output, with the isentropic efficiency significantly increasing from 31.9% up to 53.9%.

Zhang et al. [25] presented a parametric analysis on the system behaviour with three scroll expanders (one hermetic and two open-drive) in an ORC system as a support in the selection of expanders in the commercial development. The analysis of working fluid mass, system pressure difference and output expanders is compared, and the system thermal efficiency is evaluated.

The approach of Campana et al. [26] was, instead, to convert a scroll compressor into an expander so as to set up a small-scale Organic Rankine Cycle test rig to experimentally assess the performance of the commercial scroll compressor SANDEN TRS090 used as the expander; the performance was evaluated using R245fa as a working fluid and obtained a peak isentropic efficiency of 45%, with an expansion ratio of 1.9–2.1.

This study presents the characterization of a scroll expander for a low-cost small-scale ORC power plant application. A test bench is realised to test the scroll in different working conditions. The modified Sanden SHS33 compressor is used as the expander, and various tests are carried out by varying the rotation frequency of the ORC's scroll and pump to analyse the operation of the system.

The aim of this work was to establish the mechanical and thermodynamic conventional parameters as the filling factor and isentropic effectiveness, in order to compare the scroll performances with different technologies and other types of machines.

2. Description of the Scroll

Creux patented the scroll machine in 1905 as an engine. Industrial applications of scroll machines began in the 1980s, with the introduction of refrigerant scroll compressors [27]. The machine has not been widely used because of the high processing accuracy, and, with the advancement of technology, the high-precision problem of scroll machinery has been resolved. The first application of scroll machines used as expanders was recorded in the scientific literature in 1994 [18].

ORC systems imply that the system could recover part of the waste heat, but the relatively low efficiency of the scroll is a key reason preventing the spread of the technology. With the development of CFD technology, interest in exploring the internal flow field of the expander [28] increased, and, consequently, its use in such systems also increased. Today, the scroll compressor is a dominant technology in the HVAC&R industry.

Accurate modelling of the scroll expander is essential for efficiency analysis and optimal control, and various studies testify to its importance by highlighting its behaviour.

For example, Xu et al. [29] focused on the effects of four key parameters (inlet temperature, inlet pressure, outlet pressure and rotation speed) to highlight the output power from the expander and the isentropic efficiency. Ma et al. [30] instead developed an accurate mechanism-incorporated adaptive-network-based fuzzy inference system (MI+ANFIS) to establish the scroll expander model.

The scroll is a simple device, with very few moving parts; it uses two interleaving scrolls, one fixed and one orbiting eccentrically without rotating to generate variable volumes, in which the fluid can be compressed, pumped or expanded. A scheme of the working principle is shown in Figure 1, in which the fluid evolution process is shown from the suction port to the discharge.

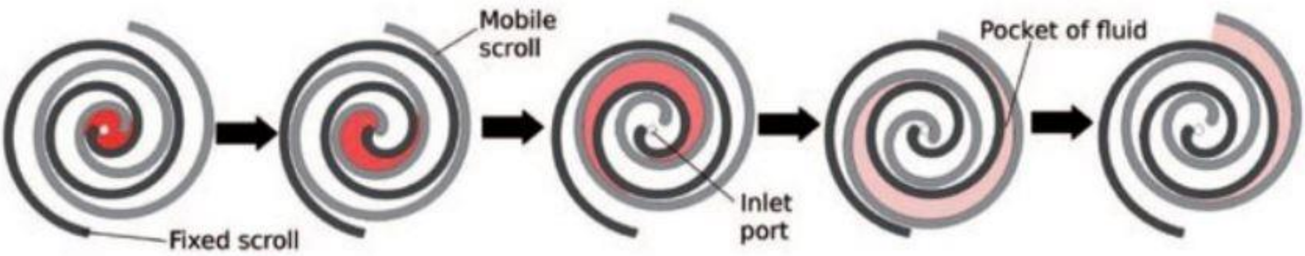


Figure 1. Scroll expander working principle scheme [31].

The machine selected for this application is a 33 cm^3 scroll compressor with the geometric ratio of the suction and discharge volume equal to 4; so, in expansion mode, the inlet volume of the expander is 8.25 cm^3. It consists of a hermetic scroll directly connected to a brushless three-phase motor that will be used as a generator. The model of the Sanden compressor used on the test bench is the SHS33 [32], Figure 2, with HFC-245fa working fluid.

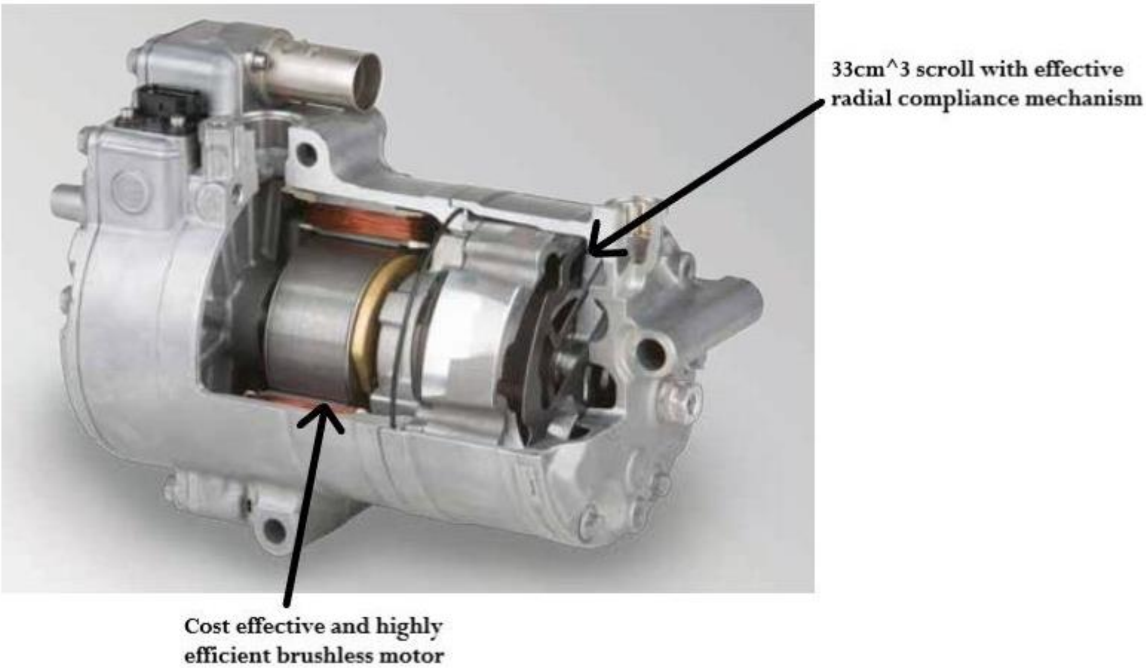


Figure 2. Scroll expander selected [33].

2.1. Isentropic Effectiveness

Isentropic effectiveness is an evaluation parameter useful for making comparisons between different technologies. It measures the actual output energy in the real transformation compared to the ideal one. Theoretically, from a thermodynamic point of view, the isentropic efficiency is the parameter used to describe the irreversibility of the system, and it is defined as the ratio of the measured enthalpy difference and isentropic enthalpy difference, calculated at suction and discharge of the expansion device (Equation (1)):

$$\eta_{is} = \frac{h_{in} - h_{out}}{h_{in} - h_{out\ is}} \tag{1}$$

Some authors [34–36] use this approach, but in experimental analysis, instead, it is more common to use the isentropic effectiveness, the ratio of generated power and isentropic enthalpy difference, in accordance with [18,37,38], as described in Equation (2):

$$\varepsilon_s = \frac{P_{el}}{\dot{m}_f * \left(h_{in} - h_{out\ is}\right)} \tag{2}$$

The use of isentropic effectiveness instead of isentropic efficiency is necessary to correctly interpret the experimental data acquired during the tests. In fact, as heat losses are not completely avoidable, the fluid expansion is non-adiabatic, and the entropy calculated at scroll discharge can be lower than the entropy at scroll suction. This could lead to finding an isentropic efficiency value higher than 100%.

2.2. Filling Factor

Another evaluation parameter is the filling factor: it is the ratio of measured mass flow rate and theoretical mass flow rate, calculated using machine geometry parameters at a specified rotational speed and fluid density. The definition is in Equation (3):

$$\Phi = \frac{\dot{m}_f}{\varrho_{f\ in} * cil * rpm} \tag{3}$$

The filling factor is a parameter that measures the variation in percentage between the theoretical mass flow rate and the real mass flow rate. This difference is due to leakage phenomena that could occur for two different causes inside the expander, both shown in Figure 3:

- Radial clearance on the scrolls tip;
- Flank clearance between the scrolls walls.

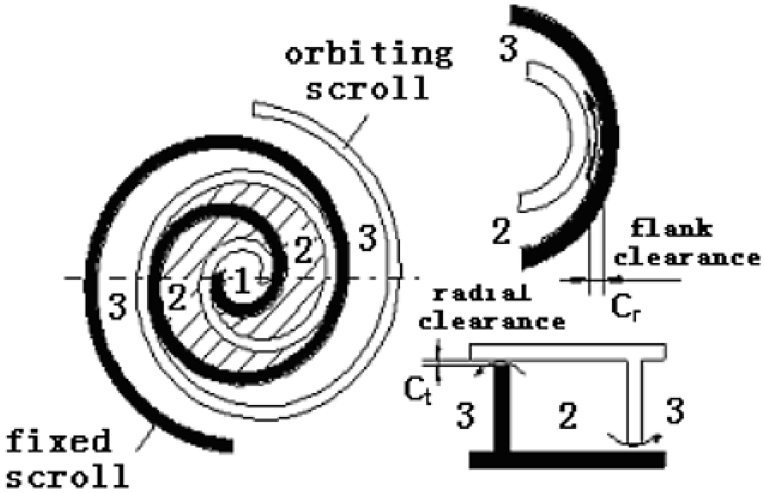


Figure 3. Graphical description of the two different types of leakage causes [39].

2.3. Expansion Ratio

Expansion ratio is the quotient of the pressures measured at the suction and discharge ports of the scroll. These pressure values depend on the plant working condition as phase-changing temperatures of the working fluid, amount of working fluid inside the plant and pump rotational speed.

$$\beta = \frac{P_s}{P_d} \tag{4}$$

A positive displacement machine has a fixed geometrical ratio that depends only on the architecture of the device, and the expansion ratio is the fraction of discharge and suction volumes. The use of a positive displacement machine is very flexible in a wide range of rpm, without strong variation in efficiency and performance, but it is possible to point out the differences that emerge during the operation out of the design point.

Different phenomena could occur inside the scroll expander in operation out of the design point. It is possible to distinguish two different behaviours, according to [31], described graphically in Figure 4a,b:

- Under-expansion, Figure 4a, happens when the expansion ratio in the circuit is higher than the nominal geometrical ratio of the inlet and outlet volumes of the expander. The effect of this condition is a loss for the non-exploitation of fluid expansion inside the machine from which reduced working extraction follows;
- Over-expansion, Figure 4b, happens when the expansion ratio in the circuit is lower than the nominal geometric ratio of the inlet and outlet volume of the expander. The effect of this condition is a discharge pressure higher than the exhaust pressure of the machine, which reduces the flow rate of the working fluid and the work extracted from the system.

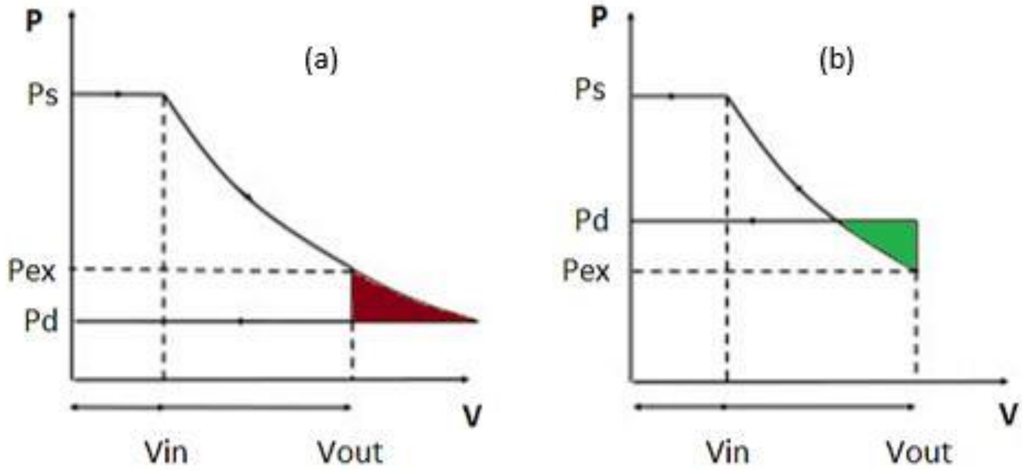


Figure 4. (**a**) Under-expansion and (**b**) over-expansion phenomena [26].

3. Experimental Investigation

In this section, we describe the experimental process carried out for testing and characterise the commercial scroll compressor used as an expander.

In the first section, the test rig is described considering all the mechanical devices and the instrumentation used. In the second part, special attention is dedicated to the data acquisition system developed. The result of these preliminary activities is the P&I scheme of the plant shown in Figure 5a.

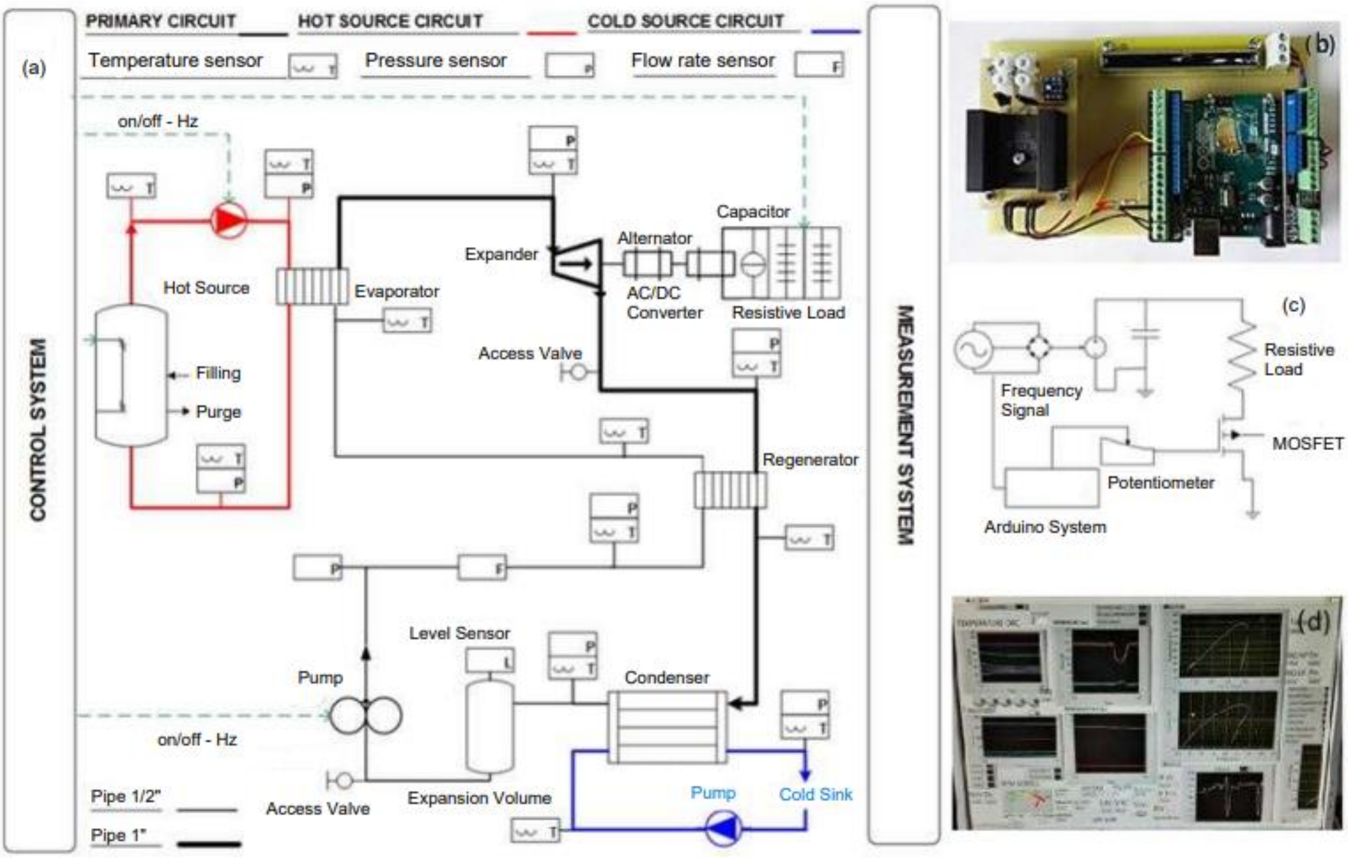


Figure 5. (**a**) P&I of the test rig, (**b**) frequency-driven rpm regulator, (**c**) scheme of the electric control system and (**d**) user interface of the monitoring software.

In the last part of this section, some consideration is given to the test execution and the data reduction techniques.

3.1. Description of the Test Rig

The experimental plant consists of three circuits separated by heat exchangers:

- The main circuit, in which the working fluid circulates and the thermodynamic cycle takes place;
- The hot source, represented by variable pressure and flow rate water circuit with a 60 kW electric boiler;
- The cold source is represented by a water loop with regulating valves that realise a variable condensation temperature with a thermocouple PID-regulated valve.

The main circuit is made of a variable-speed circulating pump, three plate heat exchangers, a working fluid storage tank and a scroll expander. The high-temperature source yields thermal energy to the ORC through a heat exchanger, in which the working fluid evaporates. Then, the working fluid expands in the scroll that extracts mechanical energy and converts it into electric energy through the brushless motor.

3.2. Data Acquisition System

The data acquisition system is monitoring software developed in Labview® 2014 SP1 (Figure 5d) that acquires the signal from the sensors and allows for a real-time interface with NIST REFPROP libraries for evaluating and displaying the thermodynamics characteristic of the fluid in the relevant points of the circuit during operation.

The main characteristics of the software are the graphical display of the thermodynamic cycle, the regulation of the pump speed and the control of the system working variables and output parameters as electric power and cycle efficiency.

3.3. Description of the Test and Data Reduction

The test conducted covered a wide range of different operational conditions in terms of pump speed, expansion ratio and rotational speed of the scroll. The scroll rpm control was realized with three-phase AC/DC converters and a variable-resistance DC electric circuit in which a logarithmic chopper was implemented, with feedback control through the frequency signal of the current generated by the scroll motor.

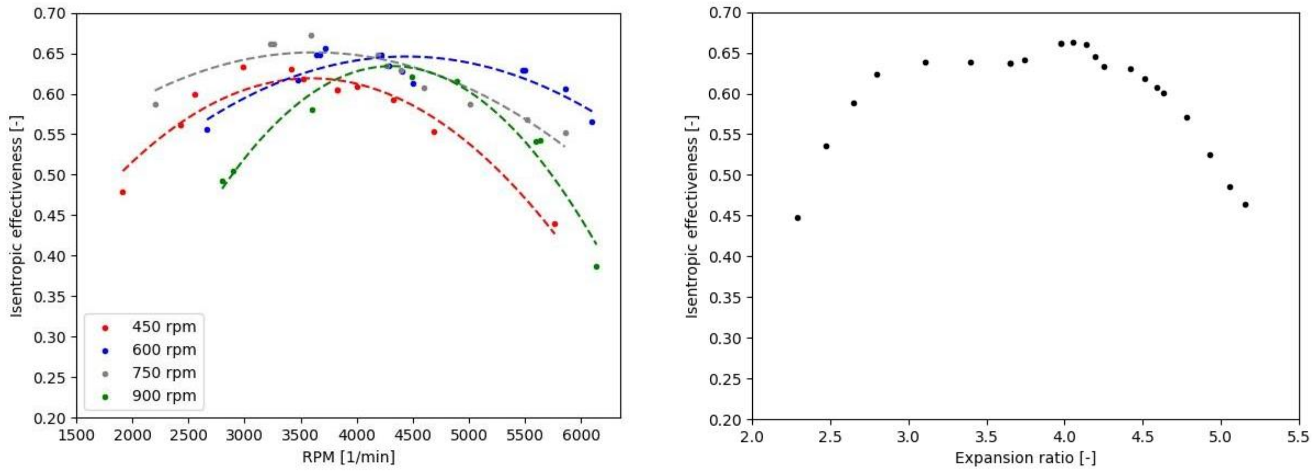
The acquired data from the experimental campaign were evaluated statistically. As a first step, different tests were conducted for about 10 minutes, setting the boundary conditions such as the thermal input and the bottom electrical loads. Within each step, the dataset was selected under steady-state requirements. In order to achieve that, the mean and standard deviation were evaluated for the measured and derived parameters in the periods of interest. Stability was assessed with a variation coefficient under 5%.

4. Results

The results presented in this paper are the mechanical and thermodynamic characteristics of the scroll. Isentropic effectiveness is reported as a function of the scroll rpm (Figure 6a) and, in non-dimensional terms, as a function of the expansion ratio realised in the circuit (Figure 6b).

In Figure 6a, different curves of isentropic effectiveness are shown for different circulating pump rotational speeds.

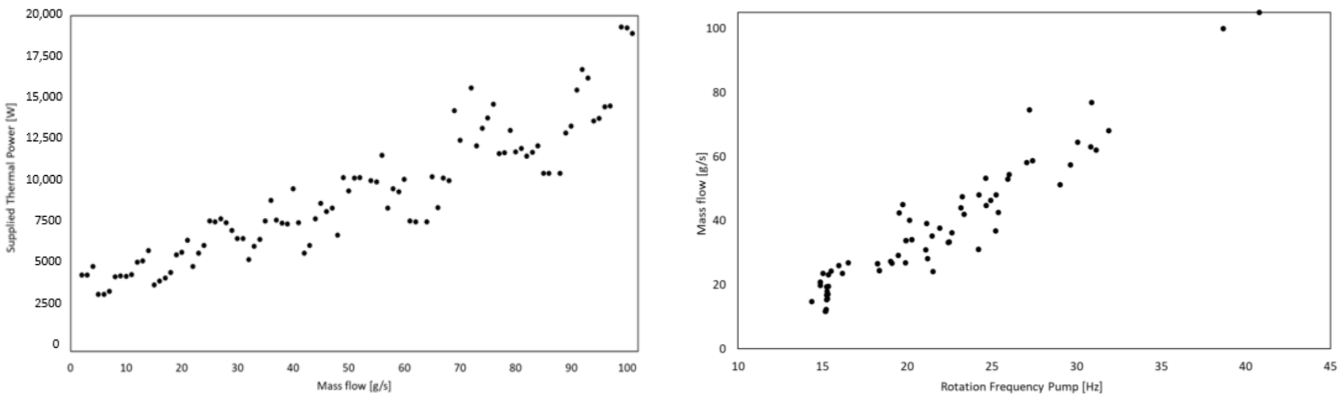
The isentropic effectiveness curves have different trends as the flow rate and head characteristics of the analysed system change. In fact, changing the rotating speed of the feed pump modifies the hydraulic characteristic of the circuit in terms of mass flow rate, pressure losses and evaporating and condensing temperature of the working fluid. The maximum isentropic effectiveness reached during the operation was 0.67, which is a very common value in the scientific literature for this type of machine [18,20,39]. For each rotational speed of the feed pump, a relative maximum is reached at different scroll rpm.



(**a**) (**b**)

Figure 6. Isentropic effectiveness (**a**) expressed as a function of scroll rpm for different pump speed and (**b**) expressed as a function of expansion ratio.

Qualitatively, the curves always identify a maximum isentropic efficiency with values of approximately 65%. The analysis of the trends gives indications about the possibility of operating the system in different configurations to obtain the maximum energy conversion efficiency based on the availability of thermal energy entering the thermodynamic cycle. In fact, the heat required by the cycle grows linearly with the flow rate (Figure 7a), and the latter, in the first approximation, depends linearly on the rotation speed of the ORC cycle feed pump (Figure 7b).



(**a**) (**b**)

Figure 7. (**a**) Thermal power necessary for the thermodynamic cycle as a function of the working fluid flow rate. (**b**) Relationship between pump rotation speed and measured flow rate.

In Figure 6b, the same points are presented as a function of the expansion ratio registered during tests at different pump rotational speeds. The expression of the isentropic effectiveness as a function of a non-dimensional parameter allows for a clearer view and offers an interesting perspective, as it is noted that the experimental points are distributed following a well-determined curve, and it is not necessary to differentiate different curves for different rotation speeds of the thermodynamic circuit feed pump.

The diagram underlines the behaviour of the expander: the maximum of isentropic effectiveness is reached for an expansion ratio close to the volumetric expansion ratio of the scroll (peaks are around $\beta = 4$). In this operating condition, the expander works at its maximum potential, and losses are kept to a minimum.

The machine's isentropic effectiveness has a stable trend between the system expansion ratio with $\beta = 3 \div 4.5$, and this fact confirms the ease of adapting the expander to different working conditions. This allows for the development of system management strategies, providing flexibility in operation during transient conditions or when sudden changes occur in the required electrical power or in the available thermal source. It is also possible to assert that the machine is more tolerant to under-expansion than to over-expansion.

It should be noted that the isentropic efficiency, calculated as defined in Equation (2), includes various phenomena that cause losses in energy efficiency, including the conversion of mechanical energy into electrical energy and the alternating/continuous conversion of electrical energy, since the power generated, in the system, is measured as the direct current electrical power to which the resistive load is subjected.

In Figure 8, we present the filling factor as a function of the scroll rpm. It can be observed that the mass flow rate of the working fluid is higher than the theoretical flow rate from the expander for each rotation speed tested. Leaks of the working fluid lead to flow rate excesses of approximately 20% in the machine's design working range identified between 4000 and 6000 RPM. These values increase up to 70%, in conditions in which the expander is slowed down too much by the electrical load, and, therefore, a high pressure drop occurs between the inlet and outlet of the machine. It can, therefore, be stated that the filling factor is always greater than one mainly due to the losses in the mechanical seals of the scroll. Another explanation for the trend of the filling factor is the pump over-sizing for the specific application, but this must be verified in future tests.

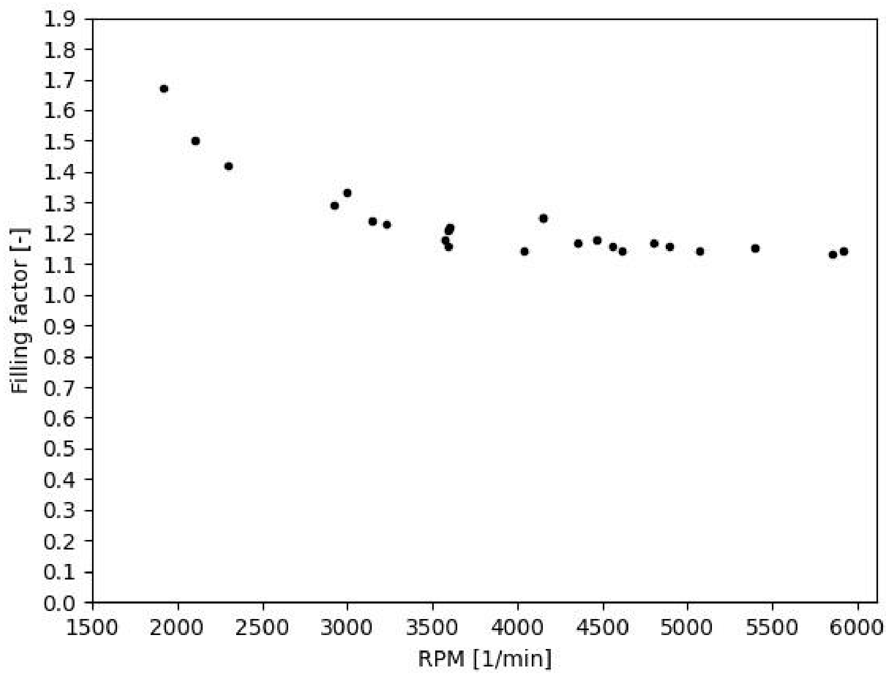


Figure 8. Filling factor expressed as a function of the scroll rpm.

Higher filling factor values, registered for low scroll rpm, are connected to inefficiencies: part of the working fluid, pumped and evaporated, spending electric energy and heat, passes through the expander without delivering mechanical work.

5. Conclusions

The aim of this work was the characterization of a commercial scroll compressor for use as an expander in a solar-driven micro-ORC plant, in order to compare the scroll performances with different technologies and other types of machines, with mechanical and thermodynamic analysis.

A test bench was realised for testing the scroll in different working conditions; the modified "commercial" Sanden SHS33 compressor was used as the expander of the system, and various tests were carried out. By varying the rotation frequency of both the scroll expander and the pump present in the system, the most influential parameters on the operation of the ORC were analysed.

The performance of the scroll was evaluated using R245fa as a working fluid in terms of isentropic effectiveness and filling factor under variable flow rates, expansion ratios and rotational speeds.

The maximum isentropic effectiveness was reached for an expansion ratio close to the volumetric expansion ratio of the scroll (peaks are around β = 4), and the machine isentropic effectiveness presents small variations in a wide range of working conditions ($\beta = 3 \div 4.5$). It is also possible to assert that the machine is more tolerant to under-expansion than to over-expansion. However, as regards the filling factor, values higher than one are always observed due to losses in the mechanical seals.

Analysis of the results leads to following considerations:

- Before proceeding to an application of the scroll tested in a solar ORC power plant, it will be necessary to improve the mechanical seals of the machine;
- A dedicated control strategy should be developed for exploiting the solar energy in the range of the expansion ratio, in which the scroll has the maximum isentropic effectiveness;
- The use of a commercial scroll compressor as an expander within an ORC system can represent a valid option for such applications, paying attention to the design of the system.

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Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

β	expansion ratio
cil	displacement
Cr	radial clearance
Ct	flank clearance
hin	enthalpy calculated at scroll suction
hout	enthalpy calculated at scroll discharge
hout is	enthalpy at scroll discharge calculated with an isentropic process
$\dot{m}$f	fluid mass flow rate
ηis	isentropic efficiency
ORC	Organic Rankine Cycle
Pel	electric power

Pd	discharge pressure
Pex	Pressure at scroll exhaust
Ps	suction pressure
rpm	rotation per minute
Vin	inlet volume of the scroll
Vout	outlet volume of the scroll
εis	isentropic effectiveness
ρf in	fluid density at scroll suction
Φ	filling factor

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