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A B S T R A C T

The aim of this paper is to facilitate the selection of the expander for a small-scale organic Rankine cycle based on an experimental comparison of a piston, a screw, a scroll and a roots expander. First, based on a literature review, a comparison between these four technologies of volumetric expansion machines is performed. Afterward, four displacement expanders [2–4 kW] are tested on two similar small-scale ORC unit with fluid R245fa. The maximum effective isentropic efficiencies measured are 53% for the piston expander and the screw expander, 76% for the variable-speed scroll and 48% for the roots machine. However, these performances do not reflect the highest efficiencies achievable by each expander: the test-rig presents experimental limitations in terms of mass flow rate and pressure drop (among others) that restricts the achievable operating conditions. The calibration of semi-empirical models based on the measurements allows to overcome this issue and to predict the isentropic efficiency in optimal conditions despite the limitations of the test-rigs. Based on experimental results, extrapolated prediction of the semi-empirical model and practical considerations, some guidelines are drawn to help the reader to select properly a volumetric expander.

1. Introduction

1.1. Content of the paper

After a brief comparison of the technologies based on a literature review (section 1: introduction), the experimental setup with the four expanders and the semi empirical model of volumetric expander are described in section 2 (Methodology). The experimental performances of the machines are then compared in terms of isentropic efficiency and filling factor (section 3: Results). Based on the experimental data, semi-empirical models are calibrated to identify the main losses of each expander and to identify their optimal operating conditions. Also, an operating map is built to help the selection of the optimal expander depending on the temperature levels of the ORC. In section 4 (Discussion), general recommendations for selecting the optimal expansion machine are proposed while accounting for the good off-design performance, the compactness, the efficiency, the power range and the ability to handle high pressure/high temperature for a given application. Finally, in the conclusion (section 5), a summary of the work is provided with perspectives to complete this work.

1.2. State of the art

Many theoretical investigations have demonstrated the considerable influence of the expander efficiency over the global performance of ORC systems ([1,2] among others). A single expander technology cannot be identified to be the optimal one for every situation, particularly for micro- and small-scale systems [2–6]. The best technology depends on a large number of parameters, including the cycle operating conditions, the system compactness, its costs and the components availability. It is therefore necessary to evaluate and to compare the performance of different expander technologies in order to help the selection of the best candidate for a given application. Very few references in the scientific literature compare the experimental performance of different expanders. In this paper, such a comparison is proposed. Volumetric machines are often chosen for small-scale applications because of their low rotational speeds, their low flow rate for a

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relatively high pressure ratio and their acceptance of two-phase flows (which may appear at the end of the expansion in some operating conditions [2]). In this work, four machines, namely a roots supercharger (Fig. 1a), a modified hermetic scroll compressor (Fig. 1b), a twin-screw expander (Fig. 1c) and a swash-plate piston expander [7] (Fig. 1d), are tested in a same micro-scale ORC system using R245fa as working fluid [8]. This refrigerant is chosen because it is one the most widespread fluids for small-scale ORC power system (<10kWe) for heat source temperatures ranging between 100 °C and 200 °C [9].

Table 1 summarizes the differences between these four technologies of volumetric expander used in ORC systems. Piston expanders are suited for low displacement (<75 l/s) and low power application (10 kW). They present the advantage to be able to work with high supply temperatures, supply pressures and pressure ratios. Those machines can present very high internal voume ratios, up to 14, which can be profitable in some applications. Their efficiency is, so far, always below 70%. It is known that piston expanders can handle a small fraction of liquid but no extensive literature can be found on this topic. Scroll expanders benefit from few rotating parts. They present a limited expansion pressure ratio since the maximum volume ratio is usually limited to 4.2. A means of increasing the volume ratio also consists in associating two expanders in series [11]. These expanders can handle very high mass fraction of liquid (>90%). Their maximal power is similar to the piston expanders (10 kW). Also, the maximum temperature and pressure are respectively 250 °C and 40 bar. Screw expanders present several advantages, such as high allowed shaft speeds (up to 20,000 RPM), compactness (see section 2.1.1), and wet expansion handling (>90%). It appears that screw expanders can work at relatively low power but are mainly used in a range of power higher than scroll or piston expanders (due to fabrication costs). Roots expanders are not frequently encountered. Technical and scientific literature about those machines is scarce. Their volume ratio is generally close to one, which leads to low pressure ratios applications. Roots expanders show power ranging approximately from 1 to 30 kW with a maximal rotational speed of roughly 20,000 rpm. These machines can handle a large fraction of liquid in expansion (appendix: Fig. 8).

2Methodology

2.1. Experimental facility

The test-rig used to characterize the expanders performance is depicted in Fig. 2. It is constructed using standard mass produced components from the HVAC industry. The working fluid is R245fa (with 5% oil mass fraction) and the test bench consists of a brazed plate evaporator, a shell and tube water-cooled condenser, a brazed plate recuperator, a gear pump and a liquid receiver. More data about the components can be found in Table 2 and in Ref. [25].

Pressure and temperature system are measured by different sensors at key locations of the test-rig. The refrigerant mass flow rate is recorded with a Coriolis flow meter at the pump exhaust. A complete description of the sensors can be found in Table 3.

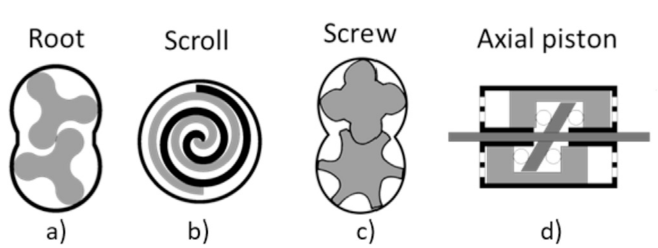

Fig. 1. Different expander technologies [10].

2.1.1. expander characteristics

2.1.1.1. Roots supercharger used as expander. A roots supercharger designed for air as working fluid is used as an expander. It presents a swept volume of 100 cm^3, which corresponds to a nominal power of 3.5 kW (Table 4). Because of its geometrical similarity to the screw expander (Fig. 1a), a slight change in the volume of the expansion chamber occurs, starting when the supply port is closed and ending at the opening of the exhaust port, leading to a volume ratio equals to 1.12. Detailed information about the component and the experimental campaign can be found in Ref. [26]. This expander is tested on a second very similar test-rig. This test rig is equipped with sensors showing the same accuracy as those used in the first test rig. The acquisition system, the fluid and the oil mass fraction are the same which guarantees the consistency of the analysis.

2.1.1.2. Piston expander. The piston expander tested is a swash-plate piston expander characterized by a total cylinder capacity of 195 cm^3 (Table 4). The lubrication is performed by an external circuit with oil injection at main friction points. The admission and exhaust processes are achieved by means of a valve-less system that induces symmetric opening and closing of the cylinder volumes. The expander is connected to an asynchronous electrical motor and a four-quadrant variable-frequency drive is used to control the shaft speed [7]. It has been sized for water/ethanol mixture applications and converted to run with R245fa. More recent (efficient) piston expanders exist and should be tested in future works [26].

2.1.1.3. Screw expander. A twin-screw expander is tested. The sizing methodology for this screw expander is presented in Ref. [27]. The design of the screw expander is oriented towards an unsynchronized machine with the rotors and bearings being lubricated by the working fluid. It presents a swept volume of 19.96 cm^3 and a built-in volume ratio of 2.5 (Table 4). The nominal supply pressure (12 bar) is relatively low compared to the two other expanders.

2.1.1.4. Variable speed scroll compressor. It is designed for vehicle air-conditioning system with fluid R134a. Its volume ratio is 2.19 and the swept volume is 12.74 cm^3 as presented in Table 4. The modification of this kind of compressor into an expander is described in Refs. [28,29]. The shaft power is evaluated using the efficiency curve of the generator.

Moreover, the compactness factor is defined as the ratio of the nominal shaft power (defined by the manufacturer) divided by the total volume of the expansion and mechanical parts (without the shaft, the generator and the casing). It shows that the best compactness is reached for the screw, followed by the scroll, the roots and the piston. The roots compactness factor is low compared to the screw mainly due to the important internal leakages of the roots prototype.

In this comparison, the four machines do not present the same nominal power. This is not a significant drawback since:

- The aim is not to compare directly the power or the efficiency of the machines but more to understand their behavior and limitations depending on the operating conditions.
- The semi-empirical model (see section 2.2) allows to evaluate their performance in optimal conditions (including the nominal power).

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Table 1
State of the art (scroll, piston, screw, roots).

Parameter	Scroll	Piston	Screw	Roots
Displacement [l/s]	0.76–32	[1.25–75]	[25–1100] [12]	–
Power [W]	[5–10,000] [10]	[1–10,000] [10]	[2000-2e5] [10]	[1000–30000]
Max. rotational speed [RPM]	10,000 [13]	3000 (swash plate: 12,000) [10]	21,000 [10]	20000 [10]
Built-in volume ratio	[1.5–4.2] [10]	[2–14] [10]	[n.a.-8] [14,15]	~1
Maximum pressure [bar]	~40	70 [15]	–	–
Max. temperature [°C]	250 [16]	560 [17]	–	–
Two-phase flow handling	Yes [18]	Low [19]	Yes [20]	Yes (Fig. 8)
Isentropic efficiency [%]	87 [21]	70 [22]	84 [23]	47 [24]

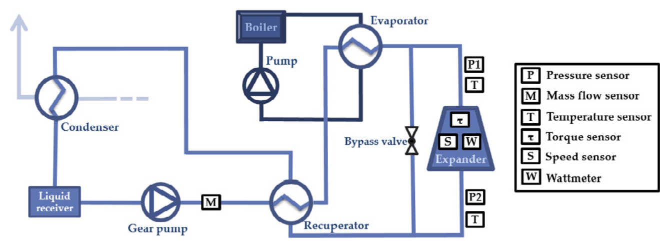

Fig. 2. Hydraulic scheme of the test-rig.

Table 2
Technical description of the components.

Component	Name	Comment
Working fluid	R245fa	–
Heat source fluid	Pirobloc HTF-basic	Thermal oil
Heat sink fluid	Ambient air	–
Diaphragm pump	Hydracell G03	Variable speed
Condenser	Alfa Laval Solar Junior 121	Fan with variable speed
Evaporator	Alfa Laval CB76-100E	Thermally insulated
Recuperator	Alfa Laval CB30-40H-F	Thermally insulated
Liquid receiver	Vertical tank	Volume is 5.7 l

2.2. Semi-empirical model

In this work, the semi-empirical model adopted to simulate volumetric expanders is an extended version of the model proposed by Lemort [30]. By accounting for the most influent physical phenomena in the expansion process with a limited number of parameters, this model demonstrates a good ability to extrapolate the expander performance out of the calibration dataset [25,31] while maintaining low computational times. One advantage of this approach is its common framework to simulate different types of technologies (scroll, screw, piston, vane, etc.) [32]. Besides of under- and over-expansion losses (due to the fixed built-in volume ratio of the machine), the model can account for pressure drops and heat transfers at the supply and exhaust ports of the machine, internal leakages, mechanical losses, heat losses to the environment and recompression phenomena. Depending on the case study, the level of details of the model may be adjusted by adding or removing some parameters. As depicted in Fig. 3, the expansion process of the fluid is divided into successive steps i.e. a supply pressure drop – Appendix:Eq. (4) ($su \rightarrow su_1$), a supply heat transfer – Appendix:Eq. (5), appendix - (su1 →su2), a two-stage expansion – Appendix:-Eqs. (6) and (7) ($su_2 \rightarrow ex_2$), an exhaust heat transfer – Appendix: Eq. (5) ($ex_2 \rightarrow ex_1$), an internal leakages flow ($su_2 \rightarrow ex_1$ in black) and a recompression flow ($ex_1 \rightarrow su_2$ in green). The supply and exhaust pressure drops are modeled as isentropic flows through converging nozzles, whose diameters d_{su} and d_{ex} respectively, are parameters to be identified. The three heat transfers (supply, exhaust and ambient) are characterized with nominal heat transfer coefficients i.e. $AU_{su,nom}$, $AU_{ex,nom}$ and AU_{amb} (Appendix:Eq. (8)). Leakages are lumped into one parameter as an isentropic flow through a simply convergent nozzle whose diameter, d_{leak}, is another parameter to identify. Mechanical losses are computed with loss proportional to the shaft speed – Appendix: Eq. (9) (by means of a losses torque τ_{loss}). Finally, in the case of piston expanders, the recompression losses due to the fluid trapped inside the clearance volume (V_0) is modeled by means of two-stage compression as proposed by Ref. [7]. For further information regarding the governing equations of this model, refer to [10].

Table 3
Sensors characteristics.

Sensor	Type	Range	Accuracy
Expander supply pressure - P1	Keller PA-21Y	[0–40] bar	0.4 bar
Expander exhaust pressure - P2	Keller PA-21Y	[0–20] bar	0.2 bar
Mass flow rate - M	Krohne optimass 7000	[10–140] $g.s^{-1}$	0.2%
Torque/rotational speed meter – τ	ETH messtechnic DRFL II	[0:20] $N.m^{-1}$	0.02 $N\,m^{-1}$
Wattmeter – W (scroll)	Gossen A2000	[0–2000] W	0.5%

Table 4
Technical data for the scroll, roots, piston and screw expanders.

Parameter	Roots	Scroll	Screw	Piston
Swept volume [cm^3]	100	12.74	19.96	195
Volume ratio [–]	1.12	2.19	2.5	4.74
Maximum supply temperature [°C]	150	130	140	250
Maximum supply pressure [bar]	12	25	16	40
Rotational speed range [RPM]	12,000	[800–8000]	20,000	[1000–4000]
Manufacturer nominal shaft power [W]	3500	2000	2000	4000
Compactness factor [W/cm^3]	1.31	3.39	21	1.2

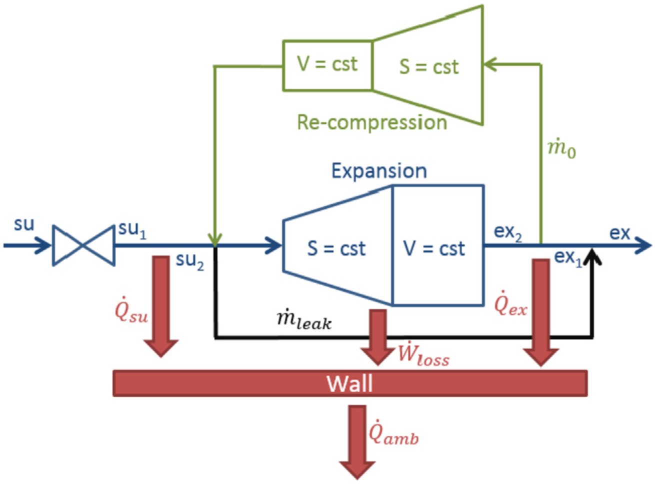

Fig. 3. Schematic representation of the overall expander model (the green part – re-compression- is only used in the case of the piston expander [33].

Ultimately, the modeling of most volumetric expanders can be summarized by the proper identification of 7 parameters (if neglecting exhaust pressure drop). Some authors use more advanced lumped-parameters models with more calibration parameters. Sometimes an additional exhaust pressure drop is modeled [7,24]. Giuffrida used a more advanced model for mechanical losses and considered additional radiative ambient losses for a single screw expander [34]. Also, Ziviani et al. [35] investigated a more complex model combining two classical semi-empirical model in series for a single screw expander. However, those models are not used here because:

- The optimization of the parameters to obtain a decent fitting of the outputs with the experimental results can lead to over-fitting problems due to the larger number of parameters.
- The simple model using only 7 parameters leads to decent prediction results (appendix - Table 8) and allows good extrapolation performance [31].
- Moreover, this simple model with 7 parameters is general and allows to model all the technologies with the same formalism.

Based on these parameters and five independent inputs (the machine rotational speed, the fluid supply and exhaust pressures, the fluid supply enthalpy and the ambient temperature), the model computes the fluid exhaust enthalpy, the shaft power and the fluid mass flow rate. It should be noted that sometimes the volume ratio is not known and therefore also becomes a calibration parameter. The parameters calibration can be performed using experimental measurements or manufacturer data, as well as simulation results from deterministic models. To this end, an optimization algorithm is used to calibrate the parameters x (detailed in section 3.3) so as to minimize a global error residual f as given in Eq. (1).

3. Results

3.1. Experimental investigation

3.1.1. Important note about the comparison of performance

Generally, a perfectly objective comparison between different types of expander is not possible for different reasons:

- It is not possible to test expanders with exactly the same level of maturity. In this case, the scroll compressors are produced in large series for many years and have reached a commercial maturity. On the other hand, the piston, roots and screw expanders are still at a prototype stage and they might see their performance and commercial maturity increasing in the future.
- Not a single expander in this experimental investigation has been sized for the test-rigs. This means that limitations in the test-rigs in terms of mass flow rate, pressure and temperature affect the performance of the expanders (not necessarily in the same way for each one).
- The fluid for which each expander is designed are not the one used in this ORC system (i.e. R245fa).
- Nominal working conditions in terms of pressure and temperature are different for each technology (higher pressure and temperature for the piston, for example).

Nevertheless, the proper choice of an expander technology is not yet straightforward and such an experimental comparison between different expander technologies does not exist in the literature. Some precautions are used to tackle the aforementioned arguments. In this study, the same test-rig with the same organic fluid is used for all expanders except for the roots one. However, the latter is tested on a very similar test rig also operating with R245fa. Only the expander is replaced which means that temperature, pressure and flow limitations are the same for each machine. Based on the experimental data, semi-empirical models are calibrated to evaluate the performance in optimal conditions not reachable with the test-rigs (section 3.3). Such methodology allows to better compare the expander's behavior despite the limitations of the test-rigs and their different nominal powers.

3.1.2. Range of operating conditions

Table 5 presents the range of operating conditions reached on the test-rigs. Only a small range of pressure ratios is covered because of limitations on the test rigs (pressure drops in the piping at the supply and exhaust of the expander and pump limitations). The piston expander is able to work on a large range and with pressure ratios up to 10.6. On the contrary, the roots expander only allows pressure ratios up to 4.47. The quality of the experimental database has been checked and validated through the Gaussian process and reconciliation method [32,33].

3.2. Performance in terms of isentropic efficiency and filling factor

The performance comparison is performed in terms of filling

$$\min_{\mathbf{x}} f = \sqrt{\sum_i \left(\left(\frac{\dot{m}_{meas,i} - \dot{m}_{pred,i}}{\dot{m}_{meas,i}} \right)^2 + \left(\frac{T_{meas,i} - T_{pred,i}}{\max(T_{meas}) - \min(T_{meas})} \right)^2 + \left(\frac{\dot{W}_{sh,meas,i} - \dot{W}_{sh,pred,i}}{\dot{W}_{sh,meas,i}} \right)^2 \right)} \quad (1)$$

Table 5
Range of operating conditions and performance.

Parameter	Roots	Scroll	Piston	Screw
Supply pressure [bar]	2.7–10	5.7–14.7	17.7–30.2	6.4–11.0
Exhaust pressure [bar]	1.3–4.1	4.3–11.1	1.75–4.01	1.6–6.1
Pressure ratio [–]	1.14–4.47	1.4–7.4	6.2–10.6	1.9–4.17
Flow [kg.s^{-1}]	0.08–0.394	0.014–0.07	0.0273–0.104	0.0290–0.137
Supply temperature [°C]	70–124.4	122–133	118–153	75–130
Exhaust temperature [°C]	58–108	47–97	60–89	50–120
Highest shaft power [W]	3049	1544	2700	1292
Shaft speed [RPM]	1000–11000	1137–7920	1000–4000	500–12450
Maximum torque [N.m]	10.4	4.6	16.4	3.31

factor (FF) and isentropic efficiency(η_{is}) evaluated over a wide range of working conditions (Eq. (2)) and (Eq. (3)).

$$FF = \frac{\dot{V}_{meas}}{\dot{V}_{th}} \quad (2)$$

$$\eta_s = \frac{\dot{W}_{sh}}{\dot{m}_r\left(h_{exp,su} - h_{exp,ex,s}\right)} \quad (3)$$

$\dot{V}_{meas}$ is the volumetric flow rate, $\dot{V}_{th}$ is the theoretical volumetric flow rate, $\dot{W}_{sh}$ is the mechanical power generated at the shaft, $\dot{m}_r$ is the refrigerant mass flow rate, $h_{exp,su}$ is the expander supply enthalpy and $h_{exp,ex,s}$ is the expander exhaust isentropic enthalpy. $\dot{V}_{meas}$ and $\dot{V}_{th}$ are defined at the supply of the expander.

The experimental isentropic efficiency calculated for the four machines in function of the pressure ratio and the shaft speed is depicted in Fig. 4. For more clarity, Figs. 9–12 are added in the appendix for the reader who would like to focus on a given expander with an adapted scale for each machine. An uncertainty propagation is performed based on Table 3 and leads to a maximal error of 3% for more than 97% of the measurements points. The errors bars are not plotted for the clarity of the figure. The trend is the same for each machine: at low pressure ratio, the efficiency is rather low mainly because of over-expansion losses and at high pressure, a decrease is observed because of under-expansion phenomenon, pressure drop and mechanical losses. Because of experimental limitations of the test rigs (low maximum refrigerant flow rate and high pressure drops in the pipelines) the optimum working conditions are not reached for each expander. Section 3.3 presents calibrated semi-empirical models to evaluate this optimum efficiency versus the pressure ratio to get a comparison unbiased by the test-rigs experimental constraints.

The filling factors measured for each expander are plotted

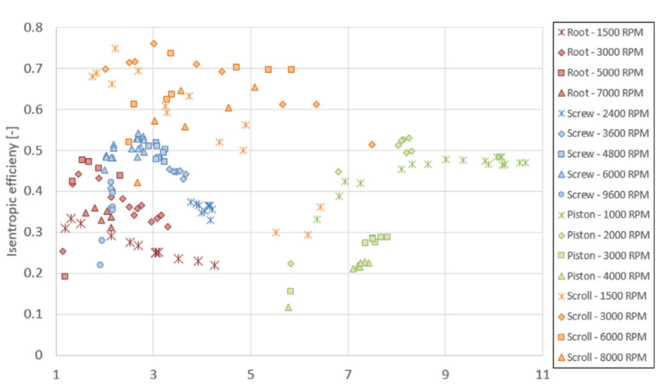

Fig. 4. Isentropic efficiency of the roots, screw, scroll and piston expander as a function of the pressure ratio.

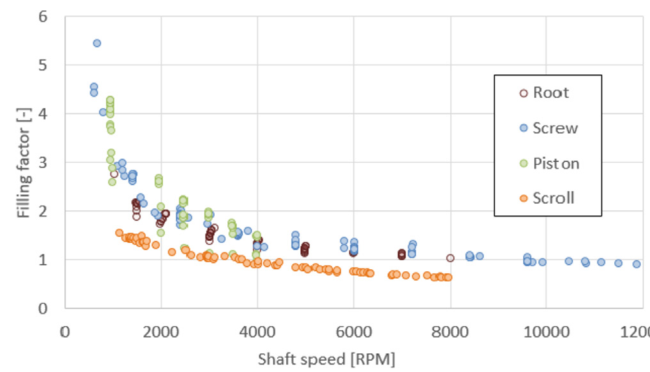

Fig. 5. Filling factor for the scroll, screw, piston and roots expander as a function of the shaft speed.

versus the shaft speed in Fig. 5. The four expanders present the same trend: the filling factor decreases monotonically when increasing the shaft speed. The filling factors of the screw and the scroll expanders become lower than one for high values of shaft speed. This is a classical trend at high shaft speed (and therefore high mass flow rate) explained by a decrease in the fluid density due to pressure drops occurring before the expansion process (in the supply line and port of the expander). The scroll expander presents the lowest filling factor (i.e. the lowest relative leakages). For the piston and roots expanders, the filling factor presents vertical lines that underline its significant dependency to the pressure ratio for the two expanders. This influence of the pressure ratio on the filling factor is depicted in (Appendix: Fig. 13).

3.3. Extrapolation trough the semi-empirical model

Following the approach described in section 2.2, the experimental results of the four machines are used to calibrate the parameters of the semi-empirical model (Table 6). The variable $\dot{m}_{nom}$is a parameter that is commonly chosen equal to the maximal flow [29].

Table 6
Calibrated parameters of the semi empirical model calibrated based on measurements.

Parameter	Roots	Scroll	Piston	Screw
d_{su} [mm]	14.3	3.2	3	10
AU_{su} [W/K]	9.7	35.1	20	36
AU_{ex} [W/K]	4.9	23.8	18	34
AU_{amb}[W/K]	5	1	5.5	4
$\dot{m}_{nom}$ [kg.s^{-1}]	0.4	0.068	0.107	0.127
A_{leak} [m^2]	3.5e-6	1.6e-6	1.1e-6	2.6e-6
τ_{loss} [N.m]	0.16	0.01	(see Oudkerk, 2016)	0.8
V_0 [cm^3]	–	–	6.5	–

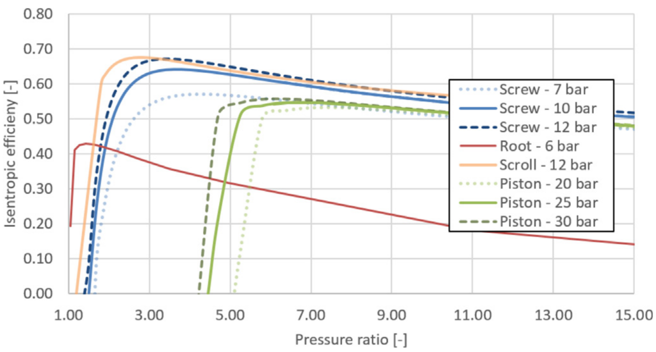

Fig. 6. Optimal efficiency curve for the scroll, the roots, the screw and the piston expander as a function of the pressure ratio.

The validity of the calibration process is ensured trough the calculation of the coefficient of determination (Appendix: Fig. 8) and the maximum difference between measurements and prediction (Appendix: Table 9). The observed values show a good agreement between model and measurements.

The semi-empirical models are used to extrapolate the performance of each expander in optimum conditions (Fig. 6). For each simulation, a constant 5 K overheating and an ambient temperature of 25 °C are imposed. For each curve, the shaft speed is adjusted to get the maximum of isentropic efficiency for each supply pressure. A wide range of pressure ratios is computed by adjusting the exhaust pressure. Isentropic efficiencies are evaluated for typical operating supply pressures in Fig. 6 (for the scroll and roots machines, only one pressure is plotted since the supply pressure does not influence the efficiency significantly - this low influence of the supply pressure comes from the shaft speed optimization which minimizes the losses for each operating point). For the two other technologies, the efficiency observed at high supply pressure is almost always the highest. Indeed, such high supply pressures induce larger mass flows, leading to higher power and so to a lower influence of quasi-constant losses. Compared to section 3.2, conclusions are essentially the same except that the limitations of the test-rigs do not influence the performance curves anymore. The screw expander efficiency simulated in optimal conditions (high mass flow rates and shaft speeds allowed) is significantly higher than the one measured on the test-rig. During experimentation, the isentropic efficiency is dropping sharply when increasing the pressure ratio. This is due to the limitation of the expander shaft speeds at high pressure induced by the test-rig limitations (mass flow rate delivered by the pump). Higher mass flow rate would

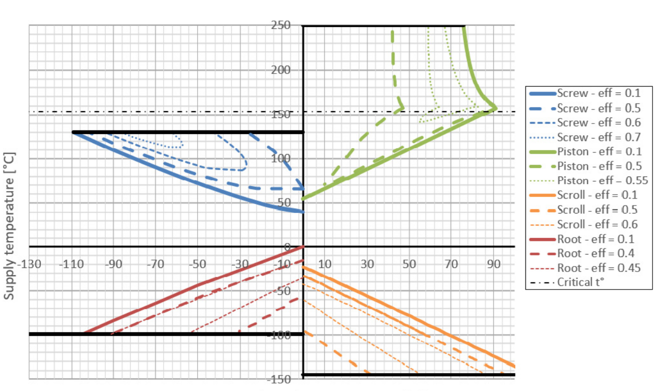

Fig. 7. Optimal performance map for the different prototypes tested with fluid R245fa evaluated for several isentropic efficiencies.

allow to reach higher shaft speeds for the expander at high pressure ratios, which would lead to higher isentropic efficiencies. Furthermore, the model predicts a decrease of the efficiency of the piston expander with high pressure ratios (mainly because of mechanical losses), which was not observable during the tests because of pressure drops in the pipes of the test-rig (at the supply and the exhaust of the expander). The roots expander presents the best efficiency for very low pressure ratios (below 1.5).

Ultimately, a mapping of the optimal performance in sub-critical conditions is performed for each expander in function of the condensing pressure and the expander supply temperature. The results are depicted in a four-quadrant graph (see Fig. 7) for which the different axes are all positive and symmetric to the origin (each quadrant is referring to an expander). The mapping assumes a constant fluid superheating of 5 K at the expander supply and is limited to subcritical operations.

Four curves are displayed to inform the variations of the optimal isentropic efficiency in function of the operating conditions. For a given supply temperature, two condensing pressure leads to an identical efficiency because of under- and over-expansion losses. The top horizontal line (i.e. the highest supply temperature) in each quadrant is referring to the supply temperature limitation of each expander (see Table 4). The black dotted horizontal line represents the critical temperature of the refrigerant (R245fa). Only for the piston machine, the maximum pressure is reached before the maximum temperature of the machine. In this case, the supply pressure is imposed at its maximum (critical pressure of the fluid) and the supply temperature is increased to cover wider ranges of power.

The scroll and screw expander maps are rather close. Whatever the isentropic efficiency considered, the screw expander map is slightly narrower because of its supply temperature limitation and, more generally, its lower isentropic efficiency. The piston shows less possibility to work at low supply temperature as expected but shows the widest running area because of the high allowed temperature and its high volume ratio. Globally, the roots expander presents rather low efficiency except at low pressure ratios (supply temperature close to the condensation temperature) where it does not suffer from over-expansion in contrast to the other technologies.

4. Discussion

Based on the scientific literature, one of the main criteria to account for when selecting a volumetric machine is the system power range. For a power larger than tens of kW, screw expanders are usually recommended. For an expander power lower than 2.5 kW, scroll and piston machines could be chosen. However, this paper shows that a screw expander can present decent efficiency at powers below 10 kW. Besides the power range, other technical limitations must be taken into consideration such as the highest allowable operating pressure and temperature, the ability to operate without lubrication oil, the highest achievable built-in volume ratio, the machine cost, and its compactness. For instance, the piston expander may be used for application with much higher supply pressure and supply temperature. Therefore, it allows to achieve higher shaft power production if those conditions cannot be reached by the other technologies. However, piston expanders only handle limited wet expansions. In terms of compactness, the best choice is the screw expander followed by the scroll, the piston and the roots (see Table 4). The flexibility (the ability to work efficiently outside of the nominal point) is important for the screw expander through its wide range of shaft speed.

In conclusion, the selection of a volumetric expander depends on the requirements of the dedicated application: is the efficiency,

Table 7
Comparison of expander advantages and disadvantages.

	High Pressure and temperature	Wet expansion	Compactness	Flexibility	Efficiency
Piston	+	−	+	+	+
Screw	−	+++	+++	+++	+
Scroll	−	+++	++	++	++
Roots	−	+++	+	−	−

the working conditions, the flexibility or the compactness the most important criteria? A comparison in terms of compactness, efficiency, achievable working conditions and flexibility (i.e. the adaptability of the expander speed to varying working conditions) is proposed in Table 7. An economic comparison is not performed since it essentially depends on the maturity of the machine. A large-scale production could decrease the price of a prototype to a level comparable with the cheapest technologies.

5. Conclusion

Four different technologies of volumetric expanders (namely scroll, screw, roots and piston) are tested experimentally in two small-scale ORC test rigs using R245fa as working fluid. Experimental measurements over a wide range of operating conditions are used to assess their performance in terms of filling factors and isentropic efficiencies. The experimental measurements are then used to calibrate semi-empirical models in order to extrapolate the machines' performance to define optimal performance maps for each technology. The scroll expander shows the highest isentropic efficiency (76%) while the piston and the screw present a 53% efficiency and the roots a 47% efficiency. It is important to note that these results are gathered in the case of small-capacity expanders (<5 kW) with different maturity of development. Furthermore, the analysis is performed using only one working fluid (R245fa). A discussion to help the selection of the most appropriate expander for a small-scale ORC is also proposed. Based on the state of the art and on the proposed analysis, the choice of the expander technology has to be conducted in parallel with the selection of the ORC architecture, range of power, operating conditions and working fluid for the selected application.

In the future, more experimental investigations should be performed with different technologies (Wankel, vane, etc.), different fluids, and different sizes of machine.

Nomenclature

A	Area (m^2)
C	Speed (m/s)
$\dot{m}$	Mass flow rate (kg/s)
max	Maximum
N	Shaft speed (rps)
oh	overheating (K)
p	Pressure (Pa)
$\dot{Q}$	Heat flow rate (W)
FF	Filling factor (−)
h	Enthalpy (kJ/kg.K)
T	Temperature (°C)
U	Heat transfer coefficient (W/(m^2.K))
v	Specific volume (m^3/kg)
$\dot{V}$	Volumetric flow (m^3/s)
$\dot{W}$	Shaft power (W)
x	Calibration parametrs

Greek

Δ	Difference (−)
η	Efficiency (−)
ρ	Density (kg/m^3)
τ	Torque (Nm)

Subscripts

amb	ambient
ex	exhaust
exp	expander
in	internal
log	mean logarithmic difference
loss	mechanical losses
meas	measured
nom	nominal
pred	predicted
r	refrigerant
s	isentropic
sh	shaft
su	supply
th	theroretical
thr	throttle
wall	wall

Appendix

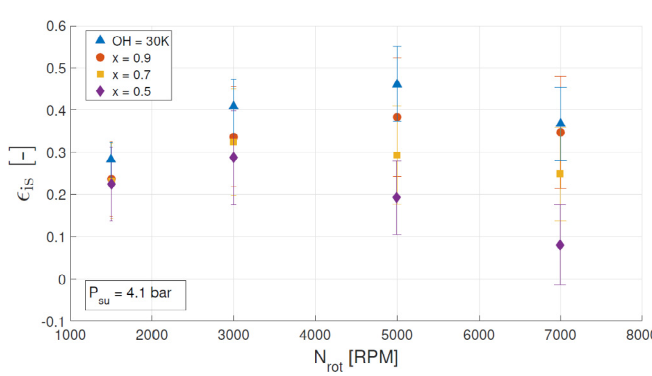

Fig. 8. isentropic efficiency of a root expander as a function of the quality of the fluid at the supply.

7.1Modeling equation for the expander semi-empirical model

The pressure drop is modeled by an incompressible flow through a fictitious nozzle presenting a diameter to calibrate (d_{su}). $C_{thr,su}$ is the speed of the fluid at the supply of the expander.

$$\Delta P_{su} = \frac{C_{thr,su}^2}{2v_{su}} \tag{4}$$

The suction heat transfer is described by a semi-isothermal process whose uniform.

Temperature is the envelope temperature (Eq. (5)). AU_{su} is also a calibration parameter.

$$\dot{Q}_{su} = AU_{su}\left(\frac{\dot{m}}{\dot{m}_{nom}}\right)^{0.8}\Delta T_{log} \tag{5}$$

The isentropic expansion to the internal pressure imposed by the built-in volume ratio of the expander (su, 2 → in) is described by Eq. (6).

$$\dot{W}_1 = \dot{m}_{in}(h_{su,2} - h_{in}) \tag{6}$$

The isentropic expansion is followed by an expansion at fixed volume to the exhaust pressure (in → ex, 2) (Eq. (7))

$$\dot{W}_2 = \dot{V}_{in}(p_{in} - p_{ex}) \tag{7}$$

The mechanical losses are assumed to be proportional to the shaft speed (Eq. (8)). τ_{loss} is a calibration parameter.

$$\dot{W}_{loss} = \frac{2\pi}{60}N\tau_{loss} \tag{8}$$

The ambient loss is computed by introducing a global heat transfer calibration coefficient AU_{amb} between the expander wall (representing the mass of the expander, assumed to be isothermal) and the ambient (Eq. (9)). The wall temperature is evaluated through an energy balance involving mechanical losses, heat transfers with the fluid and with the ambient. The latter one is computed by Eq. (9).

$$\dot{Q}_{amb} = AU_{amb}(T_{wall} - T_{amb}) \tag{9}$$

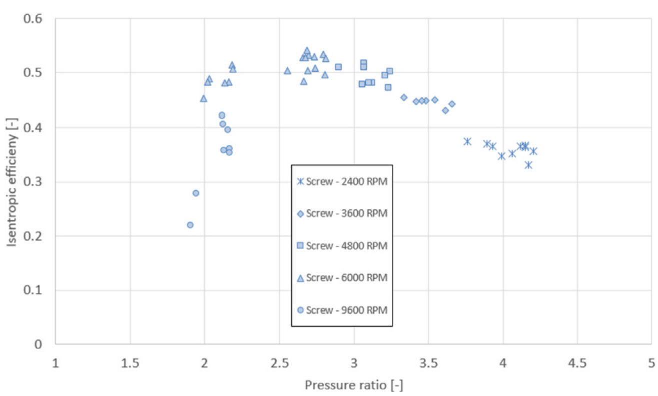

Fig. 9. Isentropic efficiency of the screw expander as a function of the pressure ratio.

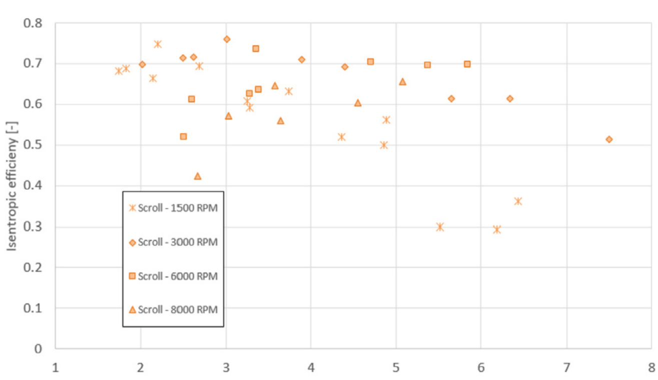

Fig. 10. Isentropic efficiency of the scroll expander as a function of the pressure ratio.

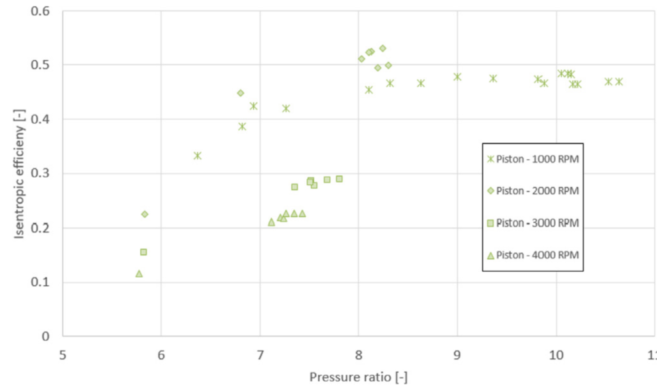

Fig. 11. Isentropic efficiency of the piston expander as a function of the pressure ratio.

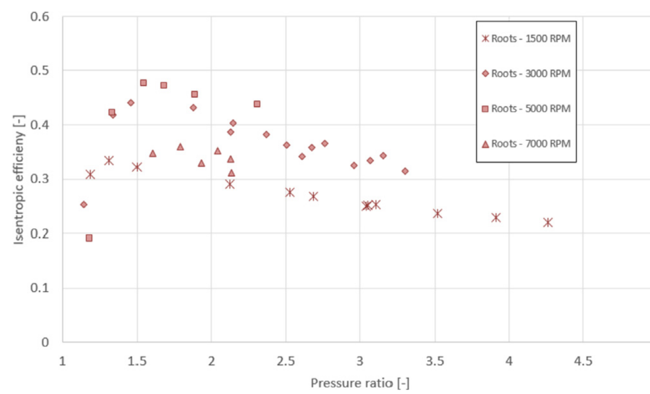

Fig. 12. Isentropic efficiency of the roots expander as a function of the pressure ratio.

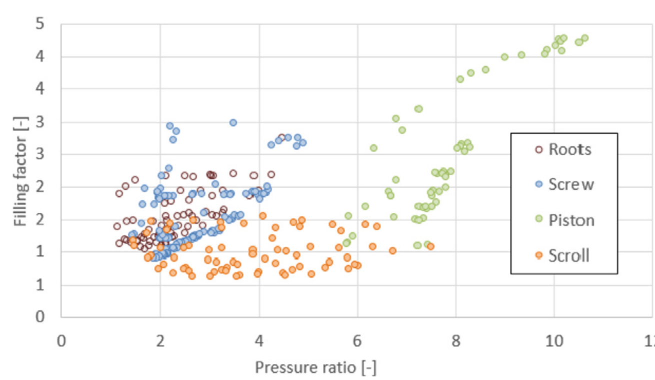

Fig. 13. Filling factor of the four machines as a function of the pressure ratio.

Table 8
Determination coefficient for the output prediction of each semi-empirical model

R^2 [%]	Scroll	Roots	Piston	Screw
Shaft power [W]	98	96.9	90	95
Mass flow [$g.s^{-1}$]	98	99	96	97
Exhaust temperature [°C]	87	99	94	98

Table 9
Maximum error for the output prediction of each semi-empirical model (Mean Absolute Percentage Errors in bracket).

Maximum error	Scroll	Roots	Piston	Screw
Shaft power [W]	89 (13%)	287 (32%)	292 (11%)	333 (40%)
Mass flow [$g.s^{-1}$]	7 (6%)	1.5 (9.5%)	3 (5.5%)	18 (14%)
Exhaust temperature [°C]	5.8	2.7	4.6	7.0

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