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Converting a commercial scroll compressor into an expander: experimental and analytical performance evaluation

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Abstract

The development of low cost small scale Organic Rankine Cycle (ORC) has a very interesting potential in generating electricity using low temperature waste heat sources. Moreover, HVAC companies could significantly extend their market if a commercial scroll compressor can be converted into an expander using similar units. Therefore, this work reports experimental test funded by an Italian HVAC company on a scroll compressor modified to work as scroll expander in a non-regenerative cycle and a subcritical fluid regime, aimed at reducing system cost and complexity. The scroll expander has been tested with its fluid R410A in a ORC cycle in order to obtain the isentropic efficiency of the scroll expander (0.5) and the pump (0.4).
On the basis of the experimental tests, a model accomplished by means of MATLAB/CoolProp has been set up to evaluate the performance of the ORC group to achieve 10 kWe as target power output. Four operative fluids have been simulated, i.e. R245fa, R134a, R1234yf, R1234ze, fixing 100°C as evaporating temperature and considering the condenser temperature in the range 20-50°C. The results have showed that R245fa is the most promising working fluid since there is a higher expansion ratio within lower pressure values. As a consequence, not only a lower mass flow rate is necessary, but overall a lower pump consumption is needed, reaching greater overall conversion efficiency (about 6.5% with condensing temperature of 20°C) and power. Thus, a commercial heat pump scroll compressor can be effectively converted into an expander. The fluid selection shows that the most common ORC fluid can be used with relative low performance but at low cost and easy management.

Keywords: small scale ORC; ORC model; MATLAB simulation; low temperature operation; working fluid selection

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1. Introduction

Nowadays, one of the major concerns of our society is the increasing energy demand worldwide. Fossil fuels indeed are limited and the related environmental impact has serious effects on human health, ecosystems and climate. In the last decades, renewable energy technologies such as photovoltaics (PVs) and wind turbines have been widely adopted and energy production from renewables has accounted for about 19.2% of the global final energy consumptions [1] in 2014. Besides PVs and wind turbines one of the most promising technology for power generation are small scale Organic Rankine Cycle (ORC) plants. Such systems can be efficiently fuelled both by renewables or using low temperature waste heat sources thus having a very interesting potential [2].

An Organic Rankine Cycle plant works similarly to a Rankine steam power plant, but it makes use of organic working fluids which are able to condense and evaporate at acceptable temperatures [2]. At present, several manufacturing companies for ORC exist but their products are mainly limited to medium and large scale [3, 4, 5, 6]. Few companies indeed offer products on a small scale i.e. in the order of <50 kWe while <10 kWe ORC systems are limited to prototype units or have not completely entered into the market. On the contrary, industrial and academic research is paying plenty of attention on small scale ORC plants [7, 8]. For example, Li et al. [9] evaluated the influence of heat source temperature and ORC pump speed on the performance of a small-scale ORC system using R245fa as working fluid. Although, the ORC system is able to recover the low grade thermal output from a 80 kWe Combined Heating and Power (CHP) unit, it reached very low performance. On the contrary, Al Jubori et al. [10] focused on the influence of several turbine design features on turbine performance in ORC systems. In particular, this study has shown that a radial-inflow turbine is able to reach higher total-to-total efficiency and power output with respect to axial configuration although it has a higher maximum overall size.

Pie, Li et al. [11] experimentally investigated the performance of a specially designed radial-axial turbine using R123 as working fluid. The test has shown that a turbine isentropic efficiency of 65% and an ORC efficiency of 6.8% can be obtained with a temperature difference of about 70°C between the hot and the cold sides. The same authors [12] evaluated the energetic and exergetic performance of the updated ORC system and the related thermal efficiency at different heat source temperatures. Muhammad et al [13] designed and tested a 1 kWe ORC system using low pressure steam in the range 1-3 bar as heat source. In particular, the authors investigated the effect of superheating on thermal efficiency recording a maximum thermal efficiency of 5.75% and an isentropic efficiency of the expander of about 58.3% at maximum power output.

Although of interest, the limited availability of commercial expanders for small scale ORC systems has hindered their adoption so far. Qiu et al. [14] conducted an expanders market research for ORC-based m-CHP systems concluding that scroll and vane expanders are good choices within the capacity of the 1–10 kW power output. According to this study, Galloni et al. [15] realized and tested a small scale ORC plant with a scroll expander and using R245fa as working fluid in order to exploit low grade heat source in the temperature range 75-95°C. Results showed interesting potential and the best obtained performance were 1.2 kW electric power, 20 kJ/kg specific work and >9% cycle efficiency. Declaye et al. [16] and Oralli et al.[17] showed that a commercially available scroll compressor can be modified to operate in expander mode reaching, e.g., cycle efficiency of 8.5% for evaporating and condensing temperatures of 97.5°C and 26.6°C. These studies remark the importance of the potential application of this conversion using a low grade heat source.

In this work, the operation of a 52 cm^3 commercial scroll compressor converted into an expander has been experimentally investigated using the native working fluid R410A to assess its feasibility to be used in small-scale ORC plants. After that, a numerical model has been developed in order to evaluate the best working fluid and operating conditions to achieve 10 kWe of target power. The paper shows a simplified procedure in order to choose the right organic fluid and estimate the related cycle performance.

2. Methodology

A complete understanding of the performance of a plant can certainly be achieved through an extended field test phase of the real system. Nevertheless, comprehensive experimental investigations to elucidate the influence of the main operating conditions are time and cost consuming. As an alternative, a numerical tool offers the opportunity to expand on the limits of the experimental analysis supporting the design process through a more cost-effective manner.

For these reasons, the commercial scroll compressor converted into an expander has been initially tested using the native working fluid R410A in a non-regenerative cycle. In this way, it has been possible to experimentally investigated the performance of the expander taking advantage of the test bench facility already set up for the compressor operation with the general aim of assessing its feasibility to be used in small-scale ORC plants.

After the test campaign and the analysis of the results, an analytical model has been developed in order to evaluate the best working fluid and operating conditions of the ORC group. Indeed, one of the objectives of the work was to evaluate the limits and the potential of this configuration in case of a subsequent industrialization phase.

3. Experience report

The experimental test bench has been set up in the facilities of the Italian HVAC company HiRef [18] interested in evaluating the performance of a scroll compressor converted into an expander. More precisely, the scroll expander under investigation was the Mitsubishi Siam ANB52FKEMT. In order to readapt it as an expander, the suction valve inside the compressor has been removed and the inverter disconnected. For the sake of simplicity and to reduce the cost of the system, the expander operated according to a non-regenerative cycle in subcritical fluid conditions. In particular, the working fluid used in this initial test phase was the R410A taking advantage of the compressor facility already set up. However, R410A has a very low critical temperature (71.35°C) and it requires higher work of the pump compared to other refrigerants operating in the same temperature differences [19, 20].

The evaporator and the condenser consist of 90 plates heat exchangers ACH 70X made by Alfa Laval [21] while water is used as heating and cooling medium. According to the pressure drop across the expander a vane pump changes its rotational speed by means of a 0.55 kWe electric motor driven by an inverter thus guaranteeing continuous system operation with time. Therefore, the total electric consumption including the inverter losses can exceed 0.55 kWe.

On the contrary, the electric motor embedded into the expander vessel operates as generator. With respect to the generator, in order to assure its safe operation, the ratio between the output voltage and the rotation speed has been kept constant during its operation. Finally, several lamps connected in parallel constituted the electrical load. Figure 1 and Figure 2 report the process and instrumentation diagram and the experimental test bench.

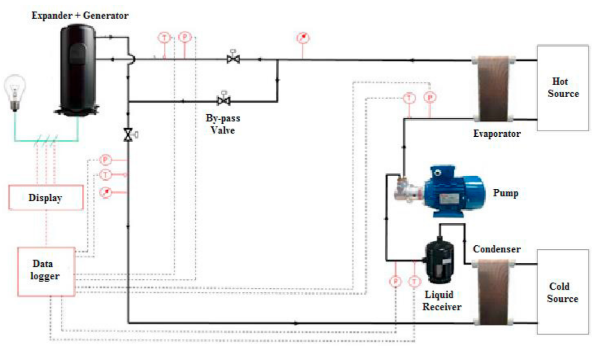

Figure 1 P&ID of the ORC systems

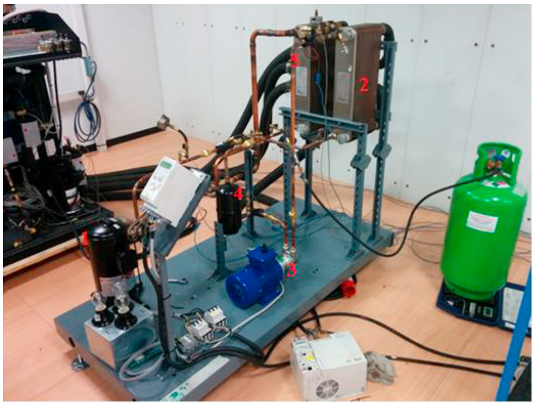

Figure 2 The experimental test bench: 1)Expander +Generator; 2)Condenser; 3)Pump; 4)Liquid Receiver; 5) Evaporator

The Agilent acquisition system [22] monitors all the relevant parameters of the plant: expander inlet temperature and pressure, expander outlet temperature and pressure. Table 1 summarizes the main characteristics of the considered ORC prototype system.

Table 1 main characteristics of the ORC system

Working fluid	R410A; 5.5 kg
Type of the expander	Mitsubishi Siam ANB52FKEMT
Vane pump	0.55 kWe impeller Greenpumps
Heat exchangers	2x Alfa Laval ACH 70X with 90 plates (both evaporator and condenser)
Data logger	Agilent
Digital power meter	WT1600 Yokogawa (voltage, current, cos φ accuracy ±0.5 %, power accuracy ±1 %)
Pressure sensors	Piezoresistive Pressure Sensor (accuracy ±1%) 2x 0-45 bar 4-20mA (high pressure side) 2x 0-30 bar 4-20mA (low pressure side)
Temperature sensors	2x PT100 DIN 1/10 (water, cold side) (accuracy ±0.15 K) 6x Thermocouples “T” type (oil, hot side and refrigerant) (accuracy ±0.5 K)

During the start-up phase, in order to prevent liquid at the inlet of the expander the scroll expander is by-passed. As soon as the pressure drop through the valve reaches approximately 10 bar the by-pass valve closes. Since it is not connected to the grid, during its operation the expander is able to rotate at frequencies different than 50 Hz. In particular, it reaches the maximum expander power output with a rotational speed of 5830 rpm.

Several tests at different flow rates of the refrigerant have been carried out to assess the influence of varying operating conditions on the expander performance. Table 2 reports the thermodynamic conditions of the main components at different time steps.

Table 2 performance of the ORC prototype system using R410A as working fluid

Time [s]	Expander Inlet Pressure [bar]	Expander Outlet Pressure [bar]	Expander Inlet Temperature [°C]	Expander Outlet Temperature [°C]	Pump Inlet Pressure [bar]	Pump Outlet Pressure [bar]	Pump Inlet Temperature [°C]	Pump Outlet Temperature [°C]
0	17.63	11.2	48.2	39.9	11.2	18.0	10.2	11.9
200	20.14	10.4	51.3	35.0	10.7	20.8	7.5	8.9
400	22.33	11.7	50.3	33.3	11.5	23.5	7.6	9.1
600	23.81	11.5	48.3	30.4	11.6	24.8	8.3	9.8
800	23.57	11.0	47.6	28.4	11.8	25.2	8.8	10.3
1000	22.2	10.9	48.9	30.1	11.2	22.7	7.8	9.3
1200	17.39	9.8	50.0	33.8	9.7	17.9	7.5	9.0
1400	15.18	9.4	50.0	36.3	9.4	15.1	7.4	9.0

In particular, an average mechanical power output of about 1298 ± 93 W has been obtained. With respect to the isentropic efficiency, a peaked value of about 50% has been achieved at 297Hz of the generator frequency (the generator connected to the expander is a 6-pole motor).

Hence, the experimental results were used to validate the numerical model of the ORC system developed by the authors in Matlab [23] while CoolProp [24] was used to obtain the properties of the fluid at specific thermodynamic states. More precisely, with an inlet pressure and temperature at the expander equal to 18 bar and 49.8°C and a pump inlet temperature of 9°C, the mechanical power output is about 1.36 kW, which is < 5% higher than the experimental result since pressure drops. Figure 3 depicts the T-s diagram of the R410A relating to the experimental conditions mentioned above while the green points are the inlet and outlet temperatures of the two heat exchangers.

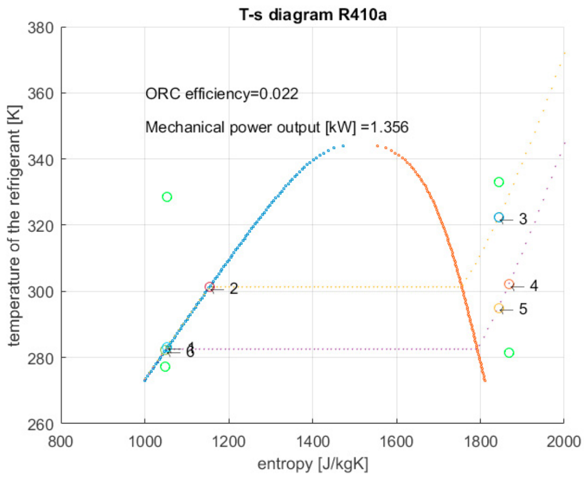

Figure 3 T-s diagram of the R410A at nominal operating conditions

4. Numerical analysis

Later on, the validated numerical model has been used to evaluate the performance of an ORC group using a similar expander technology. The numerical analysis was able to address also a common problem concerning the adoption of ORC systems, i.e. the working fluid and its impact on the plant performances with respect to the effective operating conditions. Indeed, many properties of the fluid affect the performance of the system such as its critical pressure and temperature, the vapour saturation curve, the density and the latent heat of vaporization [25]. Besides the thermodynamic properties also other characteristics of the fluid have to be taken into account to meet the severe legislative requests in terms of reduced environmental impact.

Taking into account the interesting market potential of small scale ORC plants, a target power output of the ORC equal to 10 kWe has been considered. Therefore, the thermal power input changes accordingly with the fluid while the cycle is forced to operate at fixed temperatures of the heat and cold sources. Then, temperature of the condenser has been varied thus assessing the performance of the thermodynamic cycle of each fluid with varying operating conditions of the cold source. Steady state conditions have been considered in order to evaluate the whole operating cycle for each fluid under analysis.

With regard to the model, thermal inertia, pressure drops and subcooling of the fluid at the condenser have been neglected. Referring to the main components, the following assumptions have been considered according to the results of the previous experimental tests campaign:

- generator electric efficiency equal to 0.9
- global pump efficiency equal to 0.4
- constant heat transfer coefficient at the evaporator

More precisely, the evaporator has been modeled according to the ε-NTU method considering it as a whole since the exact characterization of the heat transfer coefficients in the different regions of the vapor generator was out of the scope of this work. As regards the condenser, the heat transfer coefficient changes with the thermodynamic condition at the outlet of the expander.

With respect to the operating temperatures, an evaporation temperature equal to 90°C and an overheating temperature difference of the working fluid equal to 10°C to prevent liquid phase at the expander inlet have been assumed. On the contrary, temperature at the condenser has been varied in the range 20-50°C.

From the theoretical point of view, the isentropic efficiency of the expansion depends both on the expansion ratio and the fluid properties through the specific heat. Therefore, in order to take into account the influence of the properties of the different fluids the polytropic efficiency has been used in the analysis. The polytropic efficiency indeed depends on the specific expansion ratio as reported in equation 1:

$$\eta_{t_{poli}} = \frac{\log_{\beta}\left(\frac{1}{1-\left(1-\frac{1}{\beta^{\varepsilon}}\right)\cdot\eta_{t_{iso}}}\right)}{\varepsilon} \tag{1}$$

where $\varepsilon = (\gamma-1)/\gamma$ and $\gamma = c_p/c_v$.

For each specific operating condition the isentropic efficiency of the expansion can be calculated from the obtained polytropic efficiency thus evaluating the performance of the overall system. In general, the simulation code operates as follows: firstly the refrigerant mass flow rate m_r is supposed and after that the thermodynamic conditions of the superheated fluid at the evaporator outlet calculated. Then, since the evaporator specifications have been fixed to obtain the correct overheating temperature difference of 10°C, the refrigerant mass flow rate m_r is iteratively adjusted until the target power output of 10 kWe is achieved.

5. Results and discussion

Considering the operating conditions in the low temperature ranges, the performance of the system have been evaluated using the following working fluids: R245fa, R134a, R1234yf, R1234ze. Table 3 reports the predicted plant performance at nominal operating conditions (evaporation temperature equals to 90°C and condensation temperature equals to 20°C). In terms of results, the highest overall conversion efficiency of about 6.5% can be achieved using R245fa as working fluid. This is due to the best expansion ratio at lower overall pressure of the R245fa that requires less mass flow rate while generating high useful power.

Table 3 predicted performance of the ORC system with different fluids

Working Fluid	Expander Inlet Pressure [bar]	Expander Outlet Pressure [bar]	Expansion Ratio	Mass flow rate [kg/s]	Expander Isentropic Efficiency [%]	Overall Conversion Efficiency [%]
R134a	32.34	5.71	5.65	0.90	53.3	5.37
R245fa	10.02	1.23	8.14	0.79	49.43	6.48
R1234yf	30.71	5.92	5.19	1.03	52.92	5.53
R1234ze	24.67	4.27	5.77	0.91	51.64	5.48

Moreover, independently from the condensation temperature R245fa has proved to be the most promising working fluid as shown in Figure 4.

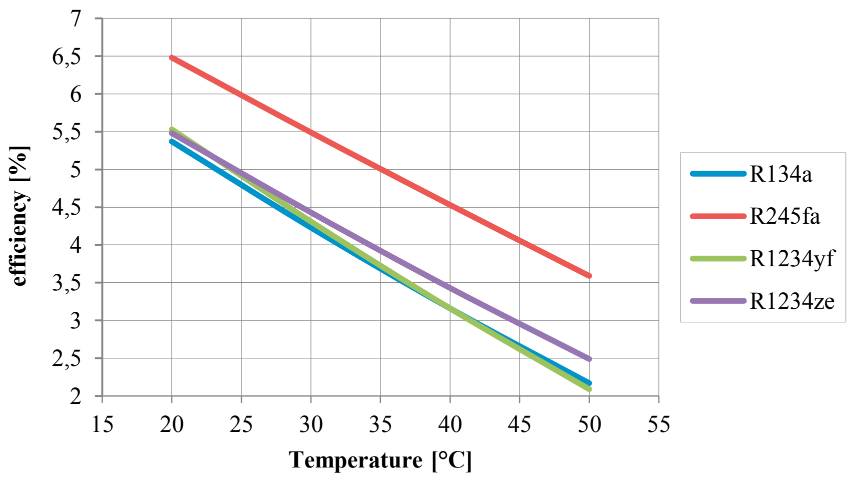

Figure 4 Overall conversion efficiency of the system with temperature at the condenser

6. Conclusions

The present work has allowed to evaluate the feasibility of converting a commercial scroll compressor into an expander for small scale ORC plants. In general, the potential performance of the system are low but complexity and cost of the system are kept down too. Within the range of temperatures and fluids considered R245fa is the most promising fluid since it allows to achieve the highest expansion ratio within lower pressure ranges. As a consequence not only a lower mass flow rate is necessary to meet the target power output of 10 kWe but also a lower pump consumption is required thus reaching the greater overall conversion efficiencies. In particular, an efficiency of about 6.48% is obtained at nominal operating conditions which is similar to those of other works in literature.

This means that the simplified methodology here adopted is able to provide a quick response about the performance prediction for a marketable product.

Therefore, the present work has allowed to prove that a commercial heat pump scroll compressor can be effectively converted into an expander having low performance but also low cost of the expander.

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