EXPERIMENTAL INVESTIGATION OF A COMMERCIAL SCREW COMPRESSOR AS AN EXPANDER WITH COMPRESSED AIR

Marco Francesconi[1]*, Gianluca Pasini, Luca Sani, Marco Antonelli[1]

University of Pisa, Department of Energy, Systems, Territory and Constructions Engineering (D.E.S.T.eC.), Italy,

marco.francesconi@ing.unipi.it

* Corresponding Author

ABSTRACT

Small scale power plants require component cost reduction to remain competitive in the energy market. Due to this scenario, the development of expander devices is usually derived from the technology of volumetric compressors, achieving the experimental analysis as a mandatory step to assess the techno-economic feasibility of a prototype.
This work investigates the experimental tests of a commercial 3 kWe twin screw oil-injected machine which has been used as compressor and expander using air as working fluid. The tests were performed for three values of pressure ratios (5:1, 7:1 and 9:1) and rotating speed ranging from 2000 to 5000 rpm. Main performance indicators, such as efficiency and operating maps, were measured and discussed, showing their potential for micro generation systems.

1. INTRODUCTION

The need to revise the energy production from fossils fuel aims to develop alternative ways for the amount of energy demand. In the last decades, the ORC technology has been receiving attention because of its flexibility in the conversion of renewable sources. In detail, this relies in the possibility of selecting the working fluid, the operating conditions and the expander, for the sake of optimizing a specified objective function for the target application. As an example, Quoilin in [1] pointed out that combined heat and power plants or solar energy applications require the plant efficiency maximization, while other applications, e.g. waste heat recovery, aim at maximizing the power output.
Among the different fields involved in the ORC analysis, the selection of the expander is a key point because it affects the design of the whole plant in terms of operating conditions and costs.
Generally speaking, expander devices suitable for ORC applications can be divided in dynamic and volumetric machines, as reported [1][2]. Dynamic machines are a mature technology on the market for output power over 50 kW, even if they suffer a dramatic amount of the rotating speeds for reduced size. Conversely, volumetric devices appear suitable for low sizes for a series of reasons such as a reduced rotating speed (namely between 1500-3000 rpm), a better compatibility with the humidity at the end of the expansion process and an acceptable value of the isentropic efficiency [1] [2]. Nonetheless, most of these machines still are prototypes [1], even if some devices start to be commercialized as proposed by Imran et al. in [3].
The choice of the expander type usually relies on the delivered power. Scroll machines are mostly used below 10 kWe, while screw expanders are used in the other cases. [3]. Nonetheless, other devices such as piston, multivane and Wankel expanders may be employed in this power range. More in detail, published studies [3]-[4] carried out on piston devices reported a power between 2-20 kW with an isentropic efficiency of 50%. Conversely, a series of papers reported in [3] evaluated the multivane machines as suitable for output power up to 2kW showing an isentropic efficiency of 70%. Other studies [5]-[6] proposed a Wankel expander for ORC applications in the range 10-50 kW, because with a similar or even higher efficiency.

In literature the most investigated device for an output power below 10 kW is the scroll expander, that has received attention for two decades because of its possible employment in ORC applications. In detail, the isentropic efficiency of this expander was found to be between 45 and 77% for output power of about 1 kW as described in [7],[8],[9],[10], even if Emhartd et al. in [11] reported a maximum value of 10 kW. In detail, Emhartd et al. proposed an analysis of the geometry on the performances, highlighting that use of geometry with a variable thickness may improve the built volume ratio thus increasing the shaft power [11]. Finally, a general summary of the performances of commercial scroll expander was summarized in [12] by means of Balje maps. On the other hand, twin screw expander earned a lot of attention for high power output, that is to say about 100 kW [1],[3],[13][14]. In detail, Stosic, Kovacevic and Smith in ([15],[16]) provided an analysis of screw machines with regard to their design and application both as compressors and expanders, while Papes et al. in [17][18] described the effects due to the pressure disturbs on the noise. Considering experimental researches, Tang et al. stated that only a few experimental data on the twin screw expanders are available in literature, because of their confidential industrial development [19]. However, a series of experimental analysis may be found in [13],[14],[19],[20].

In this work the commercial 3 kWe twin screw compressor shown in Figure 1 was tested as an expander by means of compressed air. Despite the fact that twin screw machines are reported for use above 10 kWe, a much smaller machine was used in this work to assess its suitability for very low size generation systems. The aim of this work was the assessment of the performance of a commercial compressor when used as an expander in a low-cost generation system.

2. EXPERIMENTAL TESTS

Experimental activities were performed in a facility of the University of Pisa, employing a test system designed to feed the machine by air at different operating pressure. The test rig allowed the operation of the device both as a compressor and as an expander (see Figure 2). The main characteristics of the tested machine (Figure 1) are reported in Table 1. An electric DC machine was connected to the device with a ratio 1:1 to work as a motor and as a generator. The rotational speed was regulated by a dedicated driver (ABB type DCS550) while a torque sensor (Kistler type 4503A) was installed between the electrical machine and the volumetric device to measure both speed and torque. A 1 m^3 plenum was placed upstream to the machine for compressed air storage during expander configuration tests. During the operation as a compressor, the test rig included the air intake filter and the oil separator. In effects, the use of the oil was mandatory to obtain performance worth of interests, because, apart from the rotor gear lubrication, it works as a seal between a vane and the others and, in its absence, the volumetric efficiency is practically zero. Especially, during the operation as a compressor, the pressure gradient is adverse to the flow direction and, without the oil injection, the leakages are so large that they equal the flow conveyed by the vanes. The original configuration allowed the self-injection of lubricating oil into the intake port by means of the pressure differential between discharge and suction. Conversely, during the operation as an expander, only the oil flow sufficient for the lubrication was employed. Even if a relevant oil injection might lead to better results, the device was tested under operating conditions similar to an oil-less machine to evaluate the technical feasibility of the concept. In both cases, the inlet air flow temperature was the same of the environment, namely 24-25 °C.

Figure 1: The investigated twin screw compressor

The measured quantities were the inlet and outlet pressures and temperatures, the volumetric air and oil flow rates, the mechanical torque and rotating speed. In detail, pressures were measured by piezoresistive sensors working in a range between 1 and 10 bar, while the temperatures were measured by means of T thermocouples. All the signals were acquired every 50 ms by a NI system 9201 cDAQ and processed in a LABview code, in which the data were averaged during a step of 3s. As operating conditions, the rotating speed was between 2000-5000 rpm while the higher pressure was set up to 5, 7 and 9 barA.

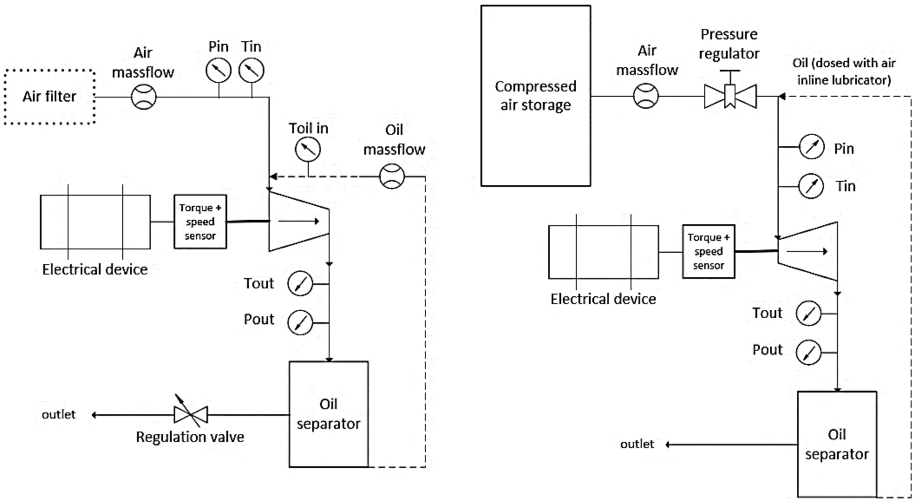


Figure 2: Experimental apparatus employed during the operation as a compressor (left) and as an expander (right)

The global efficiency η_{global} was introduced as a preliminary parameter to evaluate the behavior of the device as a compressor and expander according the Equations (1) and (2), in which the measured power at shaft $P_{meas,shaft}$ was compared with the isentropic power P_{iso} in the cases of compression and expansion indicated by the subscripts ***compr*** and ***exp***.

$$\eta_{global,compr} = \frac{P_{iso,compr}}{P_{meas,shaft}} \quad (1)$$

$$\eta_{global,exp} = \frac{P_{meas,shaft}}{P_{iso,exp}} \quad (2)$$

In detail, the Equations (1-2) considered the global effect due to mechanical, isentropic and volumetric aspects, whose single evaluation was not allowed by the employed instrumentation. Especially, these Equations did not strictly consider the presence of the oil flow during the operation, because they were formulated considering the device as a black box in which an air flow was compressed or expanded under a defined pressure ratio. Nevertheless, their crude formulation was acceptable to ascertain the behavior of the device under different operating conditions.

Table 1: Main parameters of the investigated twin screw compressor

	Male Rotor	**Female Rotor**
Number of teeth	5	6
External diameter [mm]	54	45
Inner diameter [mm]	34	25
Wrap angle [deg]	300	250
Rotor center distance [mm]	39.7	
Length [mm]	83	

3. EXPERIMENTAL RESULTS AND DISCUSSION

The experimental results will be presented with their mean values and standard deviations as uncertainties. In effects, these last ones were so small that a proper magnification has to be used for their appreciation (Figure 3). Nevertheless, their relative values were within 10% for the compressor and 6% for the expander during all the tests.

3.1 Analysis of the machine used as a compressor

During the operation as a compressor, the oil flow rate consumption was measured as a function of the rotating speed and compression ratio (see Figure 3). For a fixed rotating speed, the ratio of the oil over the air mass flow rate increased with the compression ratio, because the oil flow rate from the oil separator to the compressor was driven by the pressure difference. Conversely, the oil to air mass flow ratio decreased with the rotating speed at constant compression ratio because the oil flow rate remained practically the same (being controlled by the pressure difference) while the air flow rate was nearly linear with the rotating speed.

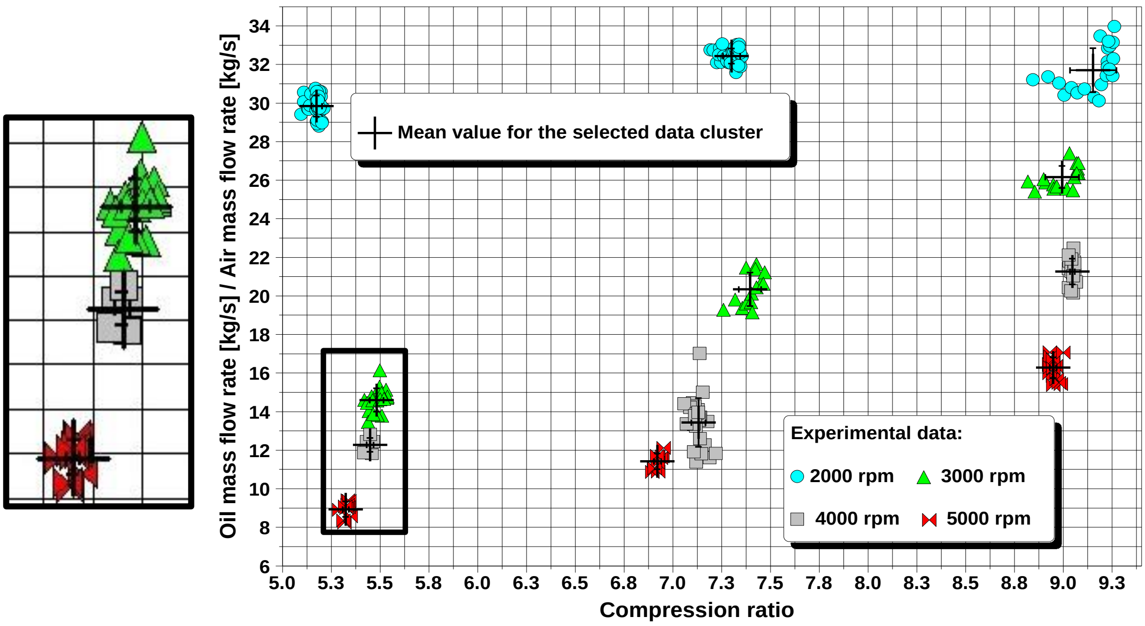


Figure 3: Ratio of the oil mass flow rate to the air mass flow rate during the operation as a compressor. A magnification of a cluster of selected data is reported aside the image for the appreciation of the uncertainties.

Similarly, to all the other volumetric machines, the compression ratio had a limited influence on the air mass flow rate and the operating maps appeared in the form of nearly vertical lines (Figure 4). The compression efficiency evidenced a reduction of the efficiency with the rotating speed (Figure 5). This may be due to the increase of the fluid velocity resulting in larger pressure losses. In detail, the device exhibited practically the same value of the efficiency at 2000-3000 rpm for a compression ratio of 7, while the absolute maximum of 0.84 appeared at 2000 rpm with a pressure ratio of 5. As for the temperatures of air and oil, it was generally observed that the outlet air temperature increased with the rotating speed (Figure 6). The only one exception was at 2000 and 3000 rpm, where the inlet oil was not cooled enough. The analysis of the temperatures gave much more emphasis to the presence of the injected oil. In effects, the outlet temperature had a minimum for a pressure ratio of 9 and this might appear as unexpected, because the outlet temperature should increase with the compression ratio. The reason was explained considering that the oil removed heat from the compressed air and this effect was relevant at the highest compression ratio in which the oil to air mass flow ratio was maximum (see Figure 1).

3.2 Analysis of the machine used as an expander

During the operation as an expander some differences were found with respect to the operation as a compressor. In detail, the major differences were:

- An increase of the mass flow rate, especially with the expansion ratio (Figure 7), while a weakly opposed behavior was found in the compressor operation;

- The relation occurring between the expansion ratio and the mass flow rate was not described by nearly vertical lines as in the previous case (Figure 7);
- Significantly lower efficiencies (Figure 8) were found (39% versus 84%).

The mass flow rate increase was explained considering the positive pressure difference existing across the inlet and outlet of the expander that limited the inlet mass backflow. Conversely, the large efficiency reduction might be due to the following reasons:

- A relevant mismatching between the geometric volume ratio and the fluid specific volume ratio, that involves under or over expansion losses;
- A cut-off grade deriving from the shape of the ports which was optimized for the operation as a compressor and not for expander in the test conditions;
- Fluid flows inside the device that were not trapped by the rotors in the passing from the suction to the discharge ports;
- Pressure losses due to a not optimal design of the ports.

Another aspect to be cited is the strong variation of the properties of the injected oil with the operating temperature of the device that might have affected the gear lubrication (Figure 9). As expected, for a fixed rotating speed, the outlet temperatures were reduced with the expansion ratio, while a rotating speed reduction caused their increase because of the greater contact time with the device surface (see Figure 9).

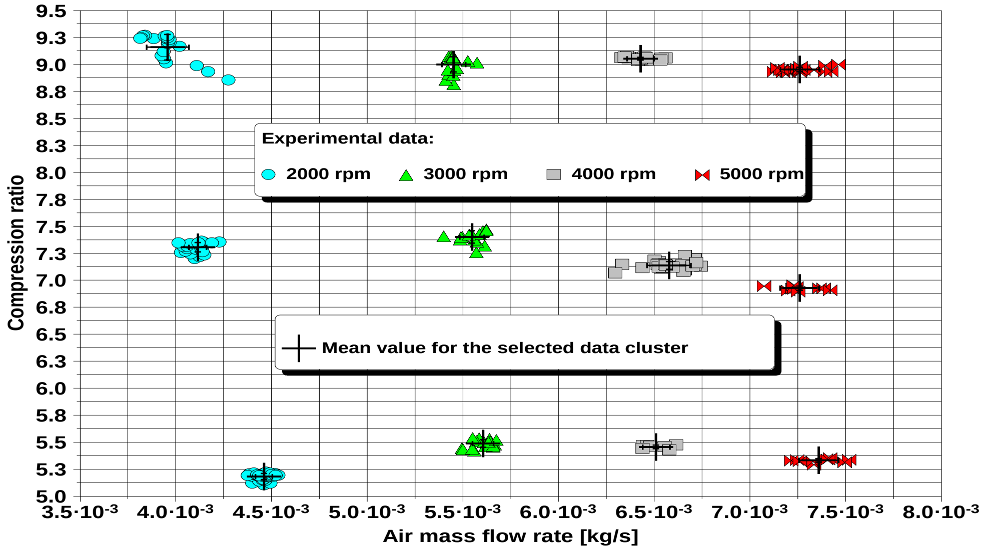


Figure 4: Compressor pressure ratio versus mass flow rate

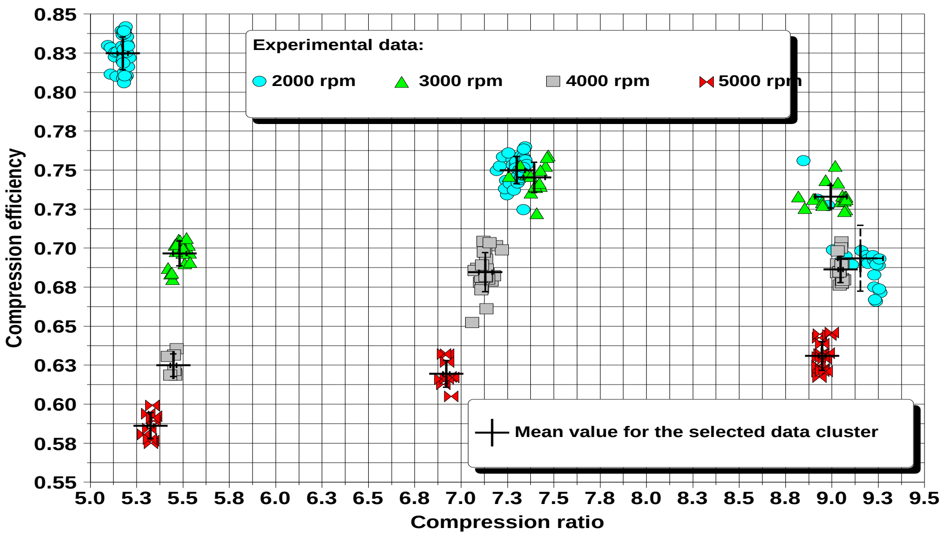


Figure 5: Compressor efficiency versus pressure ratio

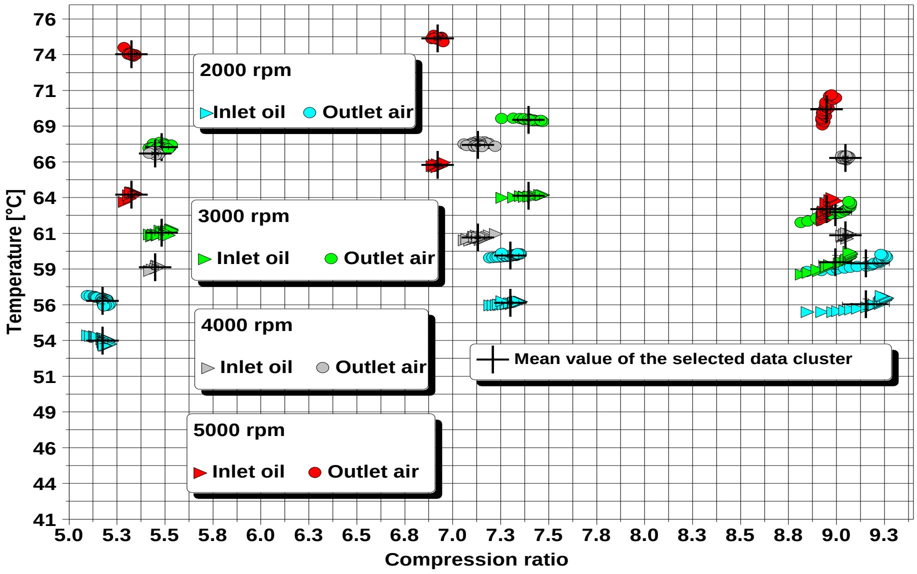


Figure 6: Air and oil temperature versus pressure ratio

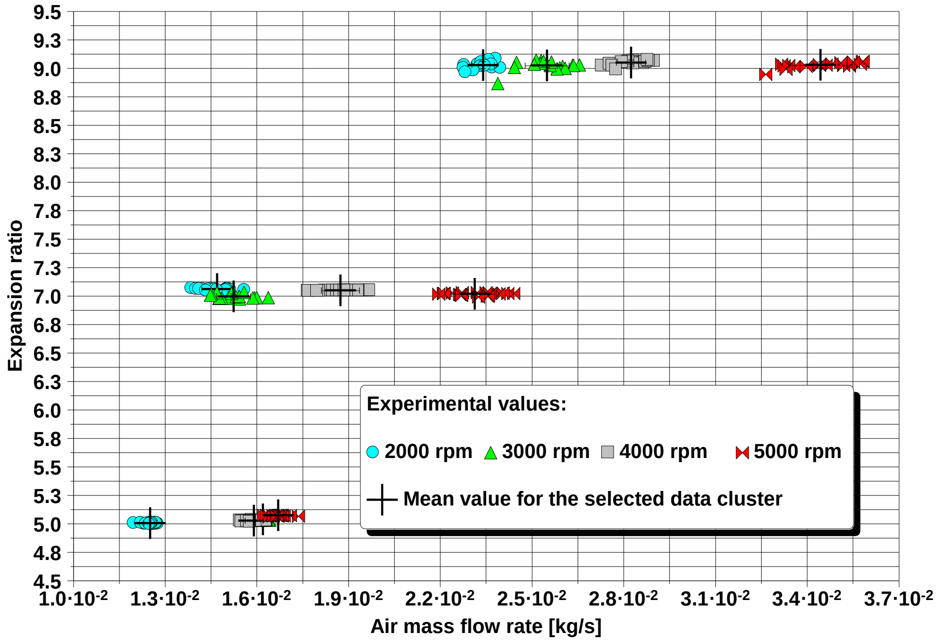


Figure 7: Expander pressure ratio versus mass flow rate

3.3 Performance comparison between compressor and expander

As last, a comparative study was shown to point out the variations of the physical quantities obtained by turning the compressor into an air expander without any modifications.

Under the same operating conditions, the measured mechanical power resulted linear with the pressure ratio in both cases (Figure 10). The power obtained when operating the machine as an expander was lower than the power requested by the same machine when operated as a compressor, mainly because of the much lower efficiency. As a matter of facts the mass flow rate requested during the operation as an expander was from 2.3 up to 5.8 times larger than the mass flow rate delivered during the compression (Figure 11). The compressor mass flow rate decreased with the pressure ratio, while the expander mass flow rate increased (Figure 4 and 7). In effects, the leakages in the compressor have to be subtracted from the ideal mass flow rate, while in the expander they have to be summed. Regarding the efficiency, the lower values appeared in the expander operation, even if they are in accordance to other studies about machines of similar size [14,21-22]. As a general guideline, the analysis suggested the device as a potential solution for expansion processes, even if it should be undergone by a series of modification to reduce leakages, as well as to obtain a proper design of the ports because the actual geometry was optimized for the compressor operation.

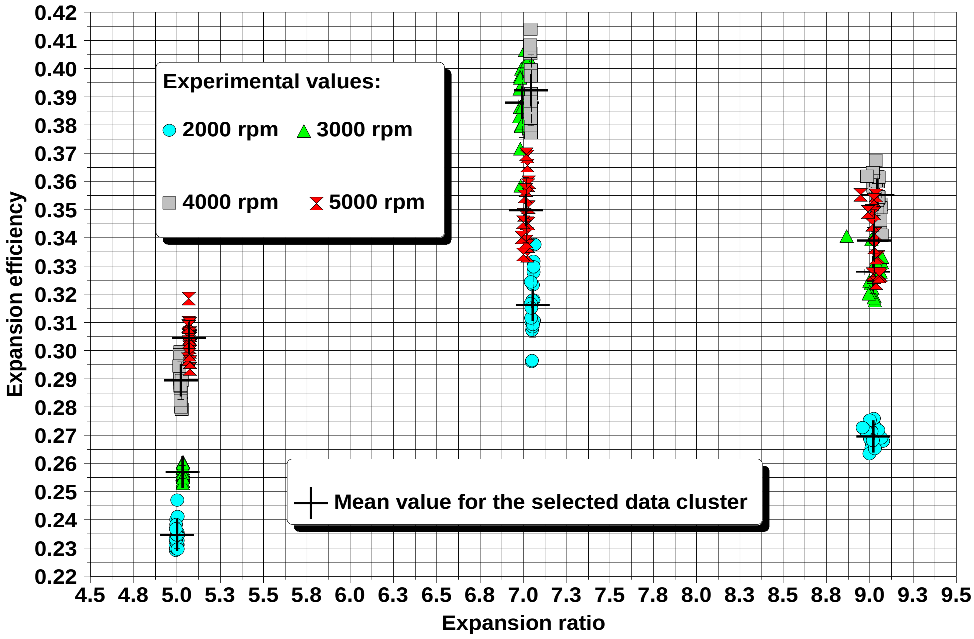


Figure 8: Expander efficiency versus pressure ratio

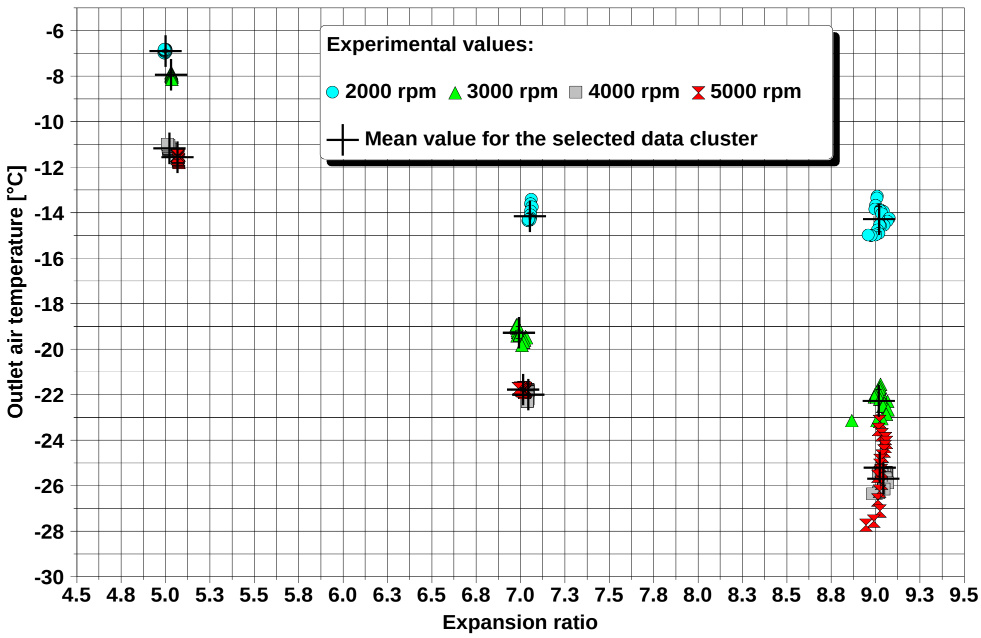
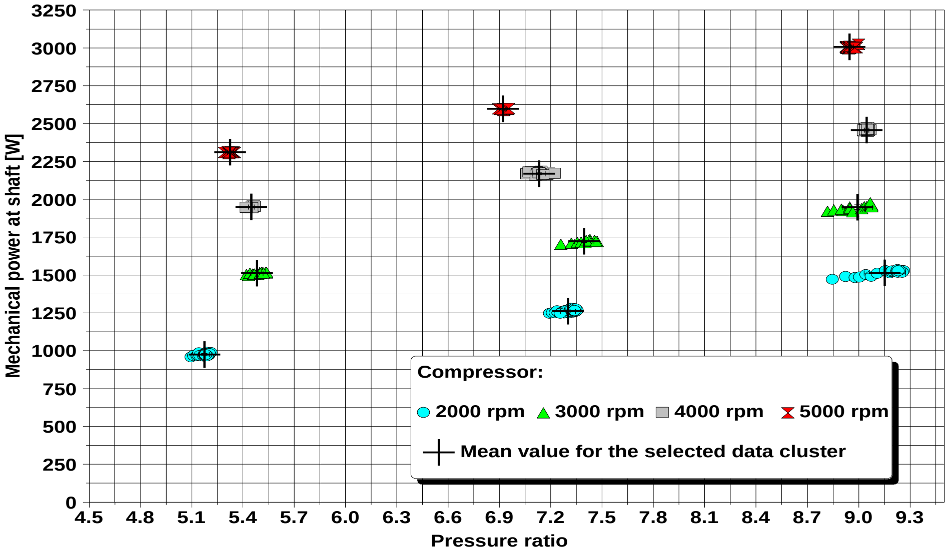
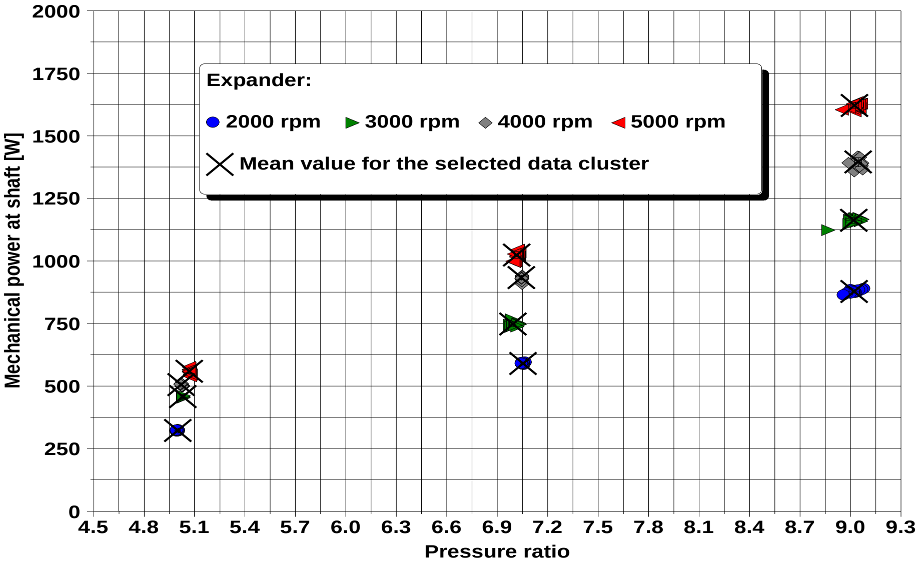


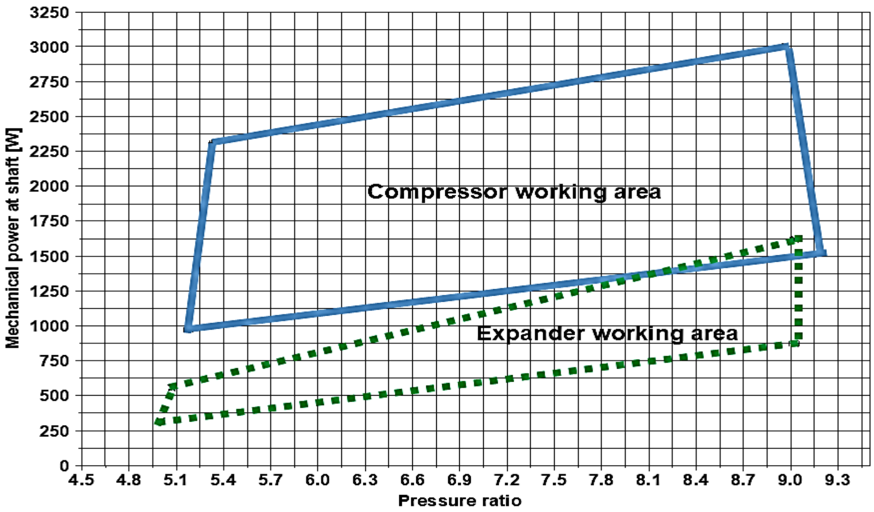
Figure 9: Outlet air temperature versus pressure ratio



(a)



(b)



(c)

Figure 10: Experimental power measured at the dynamic test bench during the operation as compressor (a) and as expander (b) with their comparison (c)

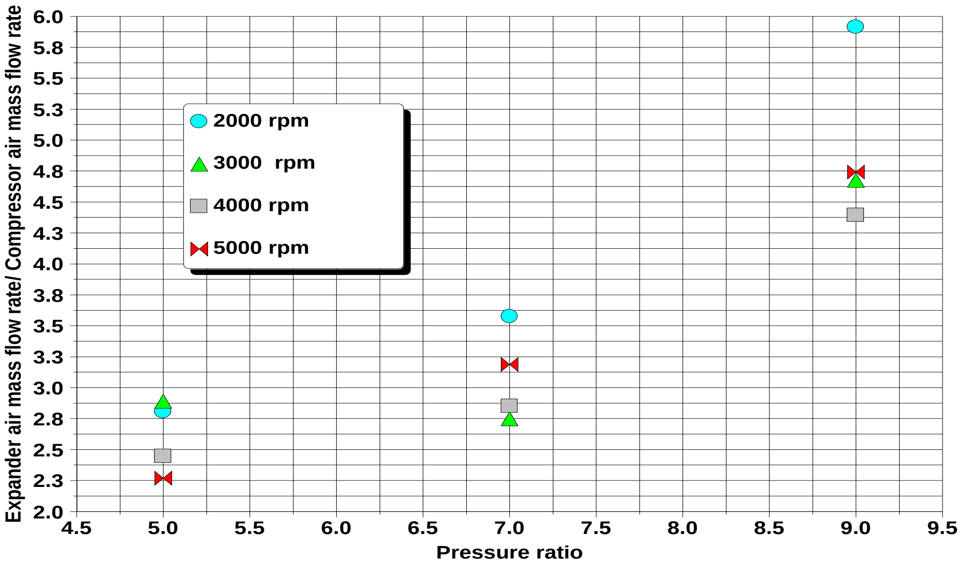


Figure 11: Expander to compressor air mass flow ratio versus the nominal pressure ratio

6. CONCLUSIONS

In this work a 3 kWe commercial twin screw compressor was experimentally tested both as a compressor and as an expander by using air as working fluid. The aim of the test was to point out the differences when the same machine is operated as a compressor or as an expander. The tests were performed at inlet pressure between 5 and 9 barA and rotating speed from 2000 to 5000 rpm.

A large reduction in both power and efficiency was observed while using the machine as an expander with the inlet compressed air at ambient temperature. Conversely, mass and volumetric flow rates were greatly increased. Despite this, the machine was able to reach an efficiency between 26 and 39% when used as an expander, similarly to other published works. The most relevant fact was that, without any modification, the machine could be operated as an expander with a satisfactory efficiency at least for a small CHP plant, in which the electricity produced may be considered in some cases no more than a by-product. A deeper analysis, for instance about the inlet and outlet ports geometry (shape, angular width), is therefore required to further improve the efficiency.

REFERENCES

[1] Quoilin S, Broek M Van Den, Declaye S, Pierre Dewallef, Lemort V. Techno-economic survey of Organic Rankine Cycle (ORC) systems. Renew Sustain Energy Rev 2013;22:168–86. doi:10.1016/j.rser.2013.01.028.

[2] Quoilin S, Declaye S, Legros A, Guillaume L, Lemort V. Working Fluid Selection and Operating Maps for Organic Rankine Cycle Expansion Machines. Int Compress Eng Conf 2012:1–10. doi:10.1126/science.aam9868.

[3] Imran M, Usman M, Park BS, Lee DH. Volumetric expanders for low grade heat and waste heat recovery applications. Renew Sustain Energy Rev 2016;57:1090–109. doi:10.1016/j.rser.2015.12.139.

[4] Bouvier JL, Lemort V, Michaux G, Salagnac P, Kientz T. Experimental study of an oil-free steam piston expander for micro-combined heat and power systems. Appl Energy 2016;169:788–98. doi:10.1016/j.apenergy.2016.01.122.

[5] Antonelli M, Baccioli A, Francesconi M, Martorano L. Experimental and numerical analysis of the valve timing effects on the performances of a small volumetric rotary expansion device. Energy Procedia 2014;45:1077–86. doi:10.1016/j.egypro.2014.01.113.

[6] Francesconi M, Dori E, Antonelli M. Analysis of Balje diagrams for a Wankel expander prototype. Appl Energy 2019;238:775–85. doi:10.1016/j.apenergy.2019.01.096.

[7] Lemort V, Quoilin S, Cuevas C, Lebrun J. Testing and modeling a scroll expander integrated into an Organic Rankine Cycle. Appl Therm Eng 2009;29:3094–102. doi:10.1016/j.applthermaleng.2009.04.013.

[8] Clemente S, Micheli D, Reini M, Taccani R. Energy efficiency analysis of Organic Rankine Cycles with scroll expanders for cogenerative applications. Appl Energy 2012;97:792–801. doi:10.1016/j.apenergy.2012.01.029.

[9] Guangbin L, Yuanyang Z, Liansheng L, Pengcheng S. Simulation and experiment research on wide ranging working process of scroll expander driven by compressed air. Appl Therm Eng 2010;30:2073–9. doi:10.1016/j.applthermaleng.2010.05.015.

[10] Gao P, Jiang L, Wang LW, Wang RZ, Song FP. Simulation and experiments on an ORC system with different scroll expanders based on energy and exergy analysis. Appl Therm Eng 2015;75:880–8. doi:10.1016/j.applthermaleng.2014.10.044.

[11] Emhardt S, Tian G, Chew J. A review of scroll expander geometries and their performance. Appl Therm Eng 2018;141:1020–34. doi:10.1016/j.applthermaleng.2018.06.045.

[12] Olmedo LE, Mounier V, Mendoza LC, Schiffmann J. Dimensionless correlations and performance maps of scroll expanders for micro-scale Organic Rankine Cycles. Energy 2018;156:520–33. doi:10.1016/j.energy.2018.05.001.

[13] Hsu S, Chiang HD, Yen C. Experimental Investigation of the Performance of a Hermetic Screw-Expander Organic Rankine Cycle 2014:6172–85. doi:10.3390/en7096172.

[14] Dumont O, Dickes R, Lemort V. Experimental investigation of four volumetric expanders Experimental investigation of four volumetric expanders. Energy Procedia 2017;129:859–66. doi:10.1016/j.egypro.2017.09.206.

[15] Smith IK, Stosic N, Kovacevic A. Power Recovery from Low Grade Heat by Means of Screw Expanders. 2014. doi:10.1533/9781782421900.

[16] Kovacevic. screw compressors. n.d.

[17] Papes I, Degroote J, Vierendeels J. Development of a thermodynamic low order model for a twin screw expander with emphasis on pulsations in the inlet pipe. Appl Therm Eng 2016;103:909–19. doi:10.1016/j.applthermaleng.2016.04.159.

[18] Papes I, Degroote J, Vierendeels J. New insights in twin screw expander performance for small scale ORC systems from 3D CFD analysis. Appl Therm Eng 2015;91:535–46. doi:10.1016/j.applthermaleng.2015.08.034.

[19] Tang H, Wu H, Wang X, Xing Z. Performance study of a twin-screw expander used in a geothermal organic Rankine cycle power generator. Energy 2015;90:631–42. doi:10.1016/j.energy.2015.07.093.

[20] Hu F, Zhang Z, Chen W, He Z, Wang X, Xing Z. Experimental Investigation on the Performance of a Twin-screw Expander Used in an ORC System. Energy Procedia 2017;110:210–5. doi:10.1016/j.egypro.2017.03.129.

[21] Nikolov A., Brümmer A., Investigating a Small Oil-Flooded Twin-Screw Expander for Waste-Heat Utilisation in Organic Rankine Cycle Systems, Energies 2017, 10, 869; doi:10.3390/en10070869.

[22] Qi Y., Yu Y., Thermodynamic Simulation on the Performance of Twin Screw Expander Applied in Geothermal Power Generation, Energies 2016, 9, 694; doi:10.3390/en9090694.