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Mechanical energy recovery from low grade thermal energy sources

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Abstract

An ORC based power plant for waste heat recovery in stationary applications has been developed and experimentally characterized. The aim of the study was to investigate the performance of a sliding vane rotary expander as the device to convert the enthalpy of the working fluid, namely R236fa, into mechanical and electric energy. A theoretical model of the expander supported the design and allowed to assess the thermodynamic transformations that take place in it. Furthermore, a deep experimental campaign explored the behavior of the expander and the one of the recovery system also at off design conditions. The experimental activity on the expander included the reconstruction of the indicated diagram using a set of high frequency piezoelectric pressure transducers that provided an accurate prediction of the pressure evolution inside the cell. The overall cycle efficiency achieved was close to 8% and further improvements concerned to the expander design have been addressed. The temperature of the upper thermal source at around 120 °C and the mechanical output power close to 2 kW make the expander and the whole system suitable for plenty of potential recovery applications.

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1. Introduction

Mechanical energy production from low and medium grade thermal energy recovery has been acquiring a growing awareness with respect to the past in many engineering applications. This is mostly due to the opportunity of participating to the new energy and environmental markets that allow to reach international commitments and additional revenues. Among the wasteful sectors, transportation is one of the most interesting, being characterized by a mean mechanical efficiency less than 30 % and a sensible thermal energy waste at medium temperature (< 250°C). Therefore, a potential recovery with an efficiency close to 10 %, would represent a smart way to reduce the fuel consumption and to respect the CO₂ emissions limitations imposed for 2015 [1]. Waste Heat Recovery (WHR) technologies are also adopted in micro-cogeneration power plants [2] or renewable energy applications as the solar and geothermal ones [3]. For these reasons, waste energy recovery into mechanical and electrical forms is a challenge both from a

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scientific and economic point of view [4]: it could add consistency to the green economy sector that currently has many difficulties to grow and become significant.

The Organic Rankine Cycle (ORC) is one of the most universally referenced solution for WHR into mechanical form. Organic fluids, thanks to a lower heat of vaporization are indeed preferable when the target power is limited and the temperature of the upper thermal source is low [5]. Although the ORC might belong to the basic thermodynamics, the availability of the technology in terms of reliability, efficiency, low cost and safety is a complex problem that prevented an industrial development of the concept [1,6]. Among the others, the main concerns are the pump (PT) and expander technologies (ET) and a proper selection of the Working Fluid (WF). Although PT could rely on the know-how of industrial sectors like the automotive one, the novelty of the application calls for the development of new components.

The great interest in lots of potential applications of the WHR through ORCs is demonstrated by the large number of studies available in literature [7–11]. Most of them provide deep theoretical studies that investigate the recovery capabilities of several organic fluids by means of energy and exergy analyses. The role of the cycle parameters is also investigated even though the ranges explored are often strictly influenced by the plant layout and application. Together with the energetic criteria, some authors take also into account the environmental performance of the organic fluids in terms of global warming potential (GWP), ozone depletion potential (ODP), atmospheric lifetime [5]. These indicators become significant for an industrial development of the application. Control strategies have been eventually set up in the automotive field to adapt the recovery system to the transient nature of the engine [12].

Expander technology recently acquired a growing interest in the scientific community [13,14] even though many concerns take place when the mechanical power is few kW or even less: dynamic machines are unsuitable because of design problems (e.g. small aspect ratios), positive displacement machines (screws, scrolls) have many operative issues related to reliability, performance and maintenance that limit the industrial interest. Volumetric reciprocating machines are suitable but limited by dimensions, reliability and operating problems such as noise, vibration, condensation control, etc [15]. Among the rotary volumetric machines, scroll [16,17], gerotor [18], rotary vanes [19,20] and screw machines [21] have been historically tested; a dominance for the scroll type must be recognized. Isentropic efficiencies of these expanders are in the range of: 0.45-0.85 for gerotors, 0.3-0.8 for scrolls, 0.5-0.85 for rotary vanes and 0.5-0.8 for screw expanders. The differences depend on the machine size. A higher power up to 50 kW has been reached for screw expanders. Scroll and rotary vanes machines have been recently more referenced, being the screw type characterized by an evident greater complexity [22]. Hence, rotary volumetric machines seem to be the most suitable ones in the range of few kW.

In the current work, an experimental test bench of a small scale ORC plant has been set up to recover the waste heat coming from the lubricant of a Sliding Vane Rotary Compressor (SVRC). A Sliding Vane Rotary Expander (SVRE) has been adopted to convert the enthalpy of the organic fluid R236fa into electric energy. The development of the prototype has been supported by a theoretical model of the expander that describes the fundamental thermodynamics that takes place in it. Off-design conditions of the recovery system have been reproduced, measured and simulated acting on the upper thermal source and the mass flow rate of the circuit. The results here discussed concern the expander performance improvement and provide a basic knowledge for a model based control of the heat recovery system.

2. Recovery System Layout

As it happens in most of volumetric displacement machines during the compression phase, in a SVRC some oil is properly injected to fulfill sealing, lubrication and air cooling purposes [23]. At the discharge, the oil is separated from the air and recycled in the lubrication circuit. Because of the friction and the compression process, at the compressor outlet the oil temperature may reach up to 100-120 °C. To prevent the variation of the lubricant properties, before being re-injected, the oil is usually cooled in a radiator.

In the current application, the waste heat usually dissipated in the radiator of a SVRC has been used as the upper thermal source for the organic Rankine cycle, as Figure 1 shows. The heat recovery takes place in a plate type heat exchanger that acts as a heat recovery steam generator (HRSG) for the working fluid. As represented in Figure 2.a, at the cold side of the HRSG, the organic fluid reaches a slightly superheated vapor state (4) to maximize the work extraction and, at the same time, to prevent a wet expansion minimizing the condensation due to the unavoidable

contact between the vapor and the colder surfaces of the vane. Therefore, at the end of the expansion (5) the WF is still in vapor phase.

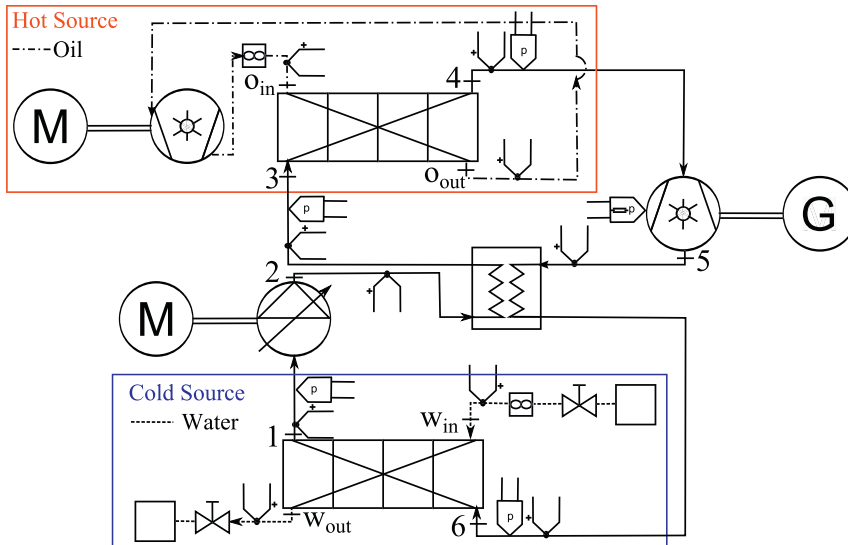


Fig. 1: Recovery system layout with sensors positioning

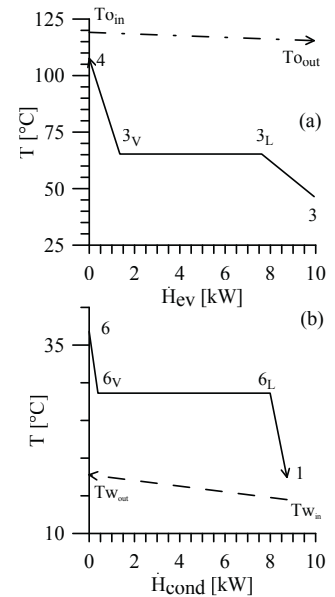


Fig. 2: Thermal power exchanged at the HRSG (a) and at the condenser (b) - test #1 of Table 3

At the expander outlet, the organic working fluid is cooled in an auxiliary heat exchanger that acts as regenerator and economizer. The degree of superheating decreases from point 5 to 6, so pre-heating the liquid downstream of the pump (2-3). Regeneration is well-known technique to increase cycle efficiency and, in a recovery application, leads to a reduction of the waste heat received (and recovered) from the upper thermal source. Therefore, if the goal is to maximize the recovery, the introduction of an intermediate regenerative heat exchanger is not advisable. On the other hand, the efficiency increase due to the regeneration allows to reduce the overall plant dimension for a given mechanical power. This is often a goal in many industrial applications such as the transportation ones. In the current work, an additional advantage of the regeneration is due to a slightly reduction of the heat extracted from the oil because of the need to keep its temperature within acceptable ranges to preserve the lubricant properties thus the proper working of the compressor.

At the regenerator outlet (6) the WF is slightly superheated and further cooled in a plate type heat exchanger that uses tap water as lower thermal source (Figure 2.b). This device leads to the condensation and sub-cooling of the WF before being re-pressurized through a rotary vane pump (1-2) connected to a brushless motor that allowed to vary the rotational speed of the cells, thus the flow rate in the circuit.

The expander is connected to a synchronous electric motor that allows to start up the plant accelerating the expander up to the correct speed and phase alignment before connecting it to the grid. Afterwards, the motor acts as a generator and keeps the rotational speed of the expander fixed by the grid frequency. Several tests at different rotational speeds have been also performed replacing the electric motor with an hydraulic circuit previously characterized to perform a preliminary direct measurement of the mechanical power [24].

The organic fluid adopted in the current study is the R236fa (Hexafluoropropane) with a GWP_{100} of 6300, an atmospheric lifetime of 209 years and a nil ODP [25]. Comparing this choice with a WF typically adopted in automotive applications, like the R245fa, the WF selection has been motivated by a lower critical temperature and a better availability and cheapness. A few oil (5 %) has been added to the R236fa to fulfill sealing and lubrication needs.

3. Expander Modeling

A comprehensive theoretical model of the expander has been developed by the authors relying on the know-how acquired when dealing with SVRCs and air as working fluid [26,27]. The model, thanks to the predictions of the expander behavior, has been used for the prototype development as a design and optimization tool. The modular structure of the model is composed of a core unit that describes what happens inside the expander vanes and interacts with auxiliary sections.

Among them, the vane filling and emptying modules provide the mass exchanges between the cell and the intake and discharge ports respectively. While the vane filling has been modeled as an isobaric process, at the expander outlet the 1D continuity, momentum and energy equations have been solved in a quasi-propagatory way that easily reproduces the unsteady phenomena due to mass exchanges between cells and the discharge plenum [28].

Under the assumptions of no leakage between adjacent cells and thermodynamic properties spatially uniform, the explicit solution of the energy equation provided the pressure evolution inside the cell during the closed volume transformation. Since experimental measurements state that R236fa keeps superheated during the whole expansion, an averaged perfect gas behavior between the expansion beginning and end states has been assumed for the WF.

The geometrical model provides the volume variation of the expander cells to achieve the expansion ratio imposed by the angular displacements of the intake and exhaust ports. The model eventually considers the blades dynamics to predict their positioning inside the rotor slots and to estimate the friction losses. Forward and backward blade tilting configurations have been also implemented to investigate how these geometries affect the mechanical efficiency of the machine acting on the friction loss distribution [29]. The friction forces are calculated from the momentum equations that describe the blade rotation and its sliding along the rotor slot. Together with the inertia, centrifugal and Coriolis forces, the pressure difference between adjacent cells may tilt the blade with respect to the slot. This required a simultaneous solution of the blade dynamics and the thermodynamic transformations inside the cell. In order to simplify the treatment, the interactions between the blade and the metallic surfaces of the machine have been assumed to happen in three points, namely between the blade tip and the stator and at the side walls of the rotor slot. Depending on the signs and on the magnitude of the forces in these locations, four configurations have been proposed: the blade can be tilted or pushed on the slot walls, either forward or backwards. Since the materials in contact are the same in all three points, a unique value for the friction coefficient has been assumed: this allows to identify a value for that parameter once the indicated power is known from p-V data and the shaft power from direct measurements of torque and angular speed.

4. Experimental Setup

The experimental campaign aimed at characterizing the behavior of the recovery system and particularly of the expander at different steady working points. The test bench reported in Figure 4 displays the most relevant components. To maximize the heat exchangers efficiency and to prevent energy losses in the piping, these components have been carefully insulated. The geometric features of the expander are listed in Tables 1 while Figure 3 provides a section view of the prototype.

The thermodynamic properties such as pressure and temperature (T-type thermocouples) have been measured upstream and downstream of all components, as shown in Figure 1. Flow rate measurements along the oil and water circuits have been performed across the evaporator and the condenser respectively. Energy balances were verified and all the powers exchanged suitably calculated.

The mechanical power at the expander shaft resulted from a calibration of the electric motor working as a generator, at different speed regimes and loads. Therefore, the mechanical power has been derived measuring the current of the electric generator with a Fluke 345 clamp multimeter. Since the rotational speed of the expander was constrained by the grid frequency, to vary the mechanical power at the shaft the WF mass flow rate was changed acting on the rotational speed of the rotary vane pump driven by a brushless motor.

The expander performance has been characterized through the reconstruction of the indicated diagram. Four Kistler 601A piezoelectric pressure transducers have been installed at different angular positions in a circumferential direction on the front cover of the expander, as reported in Figure 3. This arrangement allowed to track the pressure evolution during the expansion and the unsteady effects that take place during the vane filling and emptying. After being

ensemble averaged from a suitable number of acquisitions, the fluctuating pressure signals have been referenced thanks to the low frequency pressure measurements across the expander [30]. Piezoelectric pressure transducers, in fact, measure differential pressures on which an offset must be applied to make them matching between each other and with the reference pressures across the expander.

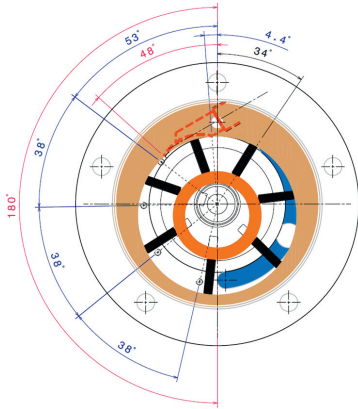


Fig. 3: Ports and piezoelectric transducers positioning

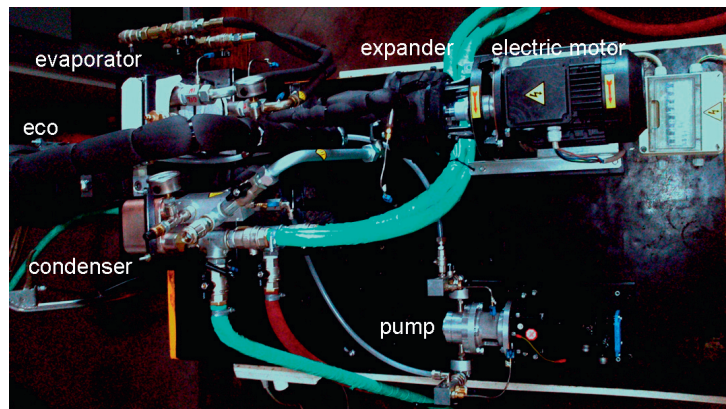


Fig. 4: Experimental test bench

Stator diameter	D_{st}	76	mm
Rotor diameter	D_{rot}	65	mm
Eccentricity	e	5.5	mm
Stator length	L_{st}	60	mm
Number of cells	N_{cell}	7	

Table 1: Expander prototype dimensions

4.1. Uncertainty Analysis

Table 2 synthesizes the absolute uncertainty of all the quantities measured in the experimental campaign. The values reported include the contributions due to the transducers and their measurement chains. The ones concerned with the thermal powers exchanged at the evaporator and the condenser derive from the uncertainty propagation of all the quantities measured, namely temperatures and flow rate. An uncertainty source lies on the reconstruction of the indicated diagram from the differential pressure signals of the piezoelectric transducers. The uncertainty of these data is directly linked to the indicated power, being the cell volume known with high accuracy. Although the accuracy of the sensors is high and the procedure to match each signal to the others enough referenced, a continuous p-V trace (in the values and in the first derivatives) is very difficult to achieve. The lack of a reference common pressure for all the sensors and the variability of the quantities from one cycle to another induce to inaccuracies in the indicated power estimated in 8-10 %. Improvements on this quantity are expected upgrading the measurement chain by means of absolute piezo-resistive pressure transducers: this feature is ongoing and will be used in future tests.

5. Results

Measurements have been performed at different steady working conditions of the recovery system. The lower saturation pressure was imposed by the cooling capabilities of tap water whose temperature was almost constant at ambient conditions. In addition, the rotational speed of the expander was constrained by the synchronism with the grid frequency. However, the brushless motor that drove the rotary vane pump and a by-pass on the hot oil circuit allowed to explore several recovery regimes. Off-design conditions were indeed produced acting on the WF mass flow rate and the upper thermal source in terms of mean oil temperature.

Quantity		Uncertainty	Units
Temperature	T	0.1	$^{\circ}\text{C}$
Low Frequency pressure transducers	p	0.08	bar
High Frequency pressure transducers	p_{pz}	0.01	bar
Oil flow rate	Q_o	0.6	l/min
Water flow rate	Q_w	5	l/min
Expander angular velocity	ω_{exp}	1	RPM
WF mass flow rate	\dot{m}_r	3.57%	
Thermal power at the evaporator	\dot{H}_{ev}	2.90%	
Thermal power at the condenser	\dot{H}_{cond}	3.58%	

Table 2: Direct and derived uncertainties

	test #1 Low Recovery	test #2 Medium Recovery	test #3 High Recovery	
p_{min}	3.60	3.52	3.66	bar_a
T_{min}	28.6	31.1	36.1	$^{\circ}\text{C}$
p_{max}	8.76	8.70	12.70	bar_a
T_{max}	107.6	78.2	98.3	$^{\circ}\text{C}$
ρ_4	52.56	47.09	78.82	kg/m^3
ω_{exp}	1533	1540	1554	RPM
\dot{m}_{oil}	1.39	1.34	1.34	kg/s
ΔT_{oil}	3.7	5.0	7.2	$^{\circ}\text{C}$
\dot{m}_w	0.63	0.95	0.75	kg/s
ΔT_w	3.3	3.3	5.2	$^{\circ}\text{C}$
\dot{m}_r	0.0580	0.0700	0.1086	kg/s
$\dot{m}_{r,max}$	0.0584	0.0526	0.0889	kg/s
\dot{H}_{ev}	9.82	13.13	19.08	kW
$P_{mech,exp}$	0.761	1.097	1.473	kW
$P_{el,pump}$	0.179	0.100	0.314	kW
$\eta_{plant,ene}$	5.93	7.59	6.45	%
$\eta_{plant,exe}$	30.12	43.93	31.78	%

Table 3: Test cases at different recovery regimes

The set of three test cases presented in this study aims at showing the recovery features of the ORC system: a particular focus is done on the expander in order to understand its behavior and the way to improve it. The measured and calculated quantities are listed in Table 3. Furthermore, Figures at page 8 report the absolute entropy diagrams of the cycle and the indicated diagrams resulting from the p-V measurements in the expander. On the left hand side of the page, the absolute entropy variations for the oil, the WF R236fa and the cooling water are shown: the three entropy fluxes have been referenced in a way to overlay the curves, without affecting entropy differences. The right hand side reports the p-V diagrams together with predictions from the expander model.

The tests progressively correspond to a low, medium and high recovery. The thermal power exchanged at the evaporator increases from 9.8 kW to 19 kW due to a WF flow rate variation in the range of 0.058-0.109 kg/s respectively. The rotational speed of the expander remains almost fixed and it is slightly pushed higher than the value that corresponds to the grid frequency as the mechanical power produced increases. The effects of the different WF flow rates is evident in the T-S diagrams, observing the shape of the lower and upper saturation states of the WF.

The entropy diagrams also offer a tool to perform a quick qualitative exergy analysis of the recovery system. The irreversibility associated with the heat exchanges at the HRSG and at the condenser are proportional to the area between the curves of the different fluids involved. An energy recovery at multiple pressure levels would certainly reduce the irreversibility, increasing the exergetic efficiency of the plant that can be calculated according to Eqn. 1 and whose values for test cases examined are listed in Table 3. However, in this kind of applications, dimensions,

weight, plant simplicity and cost saving are of great importance, especially in automotive contexts. The plant energy efficiency for the three tests is equal to 5.9%, 7.6% and 6.5% respectively and its trend does not match with the one of the mechanical power recovered.

$$\eta_{plant,exe} = \frac{P_{mech,exp}}{\dot{E}_{oil} + P_{el,pump}} \quad (1)$$

The pressure levels of 8.76 bar in test #1, 8.70 bar in test #2 and 12.71 bar in test #3, depend on the equilibrium between the WF flow rate delivered by the pump and the pressure losses produced by circuit layout (seen by the pump). The saturation temperature corresponding to this pressure must match with the temperature of the source, for a given evaporator that is already designed and chosen. This happens always in all the conditions tested, having fixed at the design condition a heat exchanger with a high pinch point (40 °C-45 °C). Hence, when the WF flow rate decreases, the overheating increases as it is clearly demonstrated by Figure 5 and 7. The effect of a low flow rate on the operating pressure can be noticed comparing Figures 7 and 9. In Figure 9 the temperature of the oil entering the evaporator was increased in order to fulfill the thermal requirements of a greater WF flow rate and a greater degree of overheating: in this situation almost 19 kW were exchanged.

The operating conditions of the cycle affect the expander performance at design and off-design points. The most powerful tool to investigate the flow evolution inside the expander cells is the indicated diagram in Figures 6, 8 and 10. All the tests show an improper vane filling process that, in agreement with the theoretical model, should be isobaric or close to it. In reality, pressure during the filling process starts to decrease preventing a full and regular machine feeding. Reference flow rates are represented by the values that could be calculated according to:

$$\dot{m}_{r,max} = \rho_4 \dot{V} = 60 \rho_4 V_{inl} \omega_{exp} \quad (2)$$

that expresses the maximum flow rate achievable when the inlet thermodynamic inlet conditions are known (state 4) and machine geometry (V_{inl}) and rotational speed (ω_{exp} [RPM]) are fixed. Equation 2 gives for the three tests 0.0584 kg/s, 0.0526 kg/s, 0.088 kg/s respectively, while real values of the flow rates circulating are 0.0580 kg/s, 0.0700 kg/s, 0.1086 kg/s respectively: in all cases, circulating flow rates are greater than the maximum allowable flow rates that the machine can induct. From a design point of view of the SVRE, the following conclusions apply:

- A partial condensation takes place when the WF enters the vane as the result of unavoidable contacts with the cold metallic surfaces. This phenomenon is reduced when the degree of overheating decreases, as it happens in Figure 6 where a better vane filling than Figure 8 takes place. This condensation can be reduced keeping the expander warm through a better insulation or, more efficiently, heating it with the same hot source of the WF before the recovery;
- Leakage takes place among adjacent vanes and between vanes and lateral covers. This aspect could be improved mixing a given quantity of oil in the WF: the oil would be entrained with the WF across all the plant sections but kept in the liquid phase: this would favor an oil film formation on the stator surfaces of the expander thanks to the centrifugal forces, so decreasing the friction losses too. Different WF and oil mixtures are under testing in order to set up the best mass flow ratio: oil presence would also avoid leakage across the clearances between blades and slots inside the rotor as it usually happens inside compressors.

Another noteworthy effect in the indicated diagram of the test cases concerns the expansion trend. Although the expander was properly insulated, the expansion never matches with an adiabatic law. The real transformation is rather polytropic with a positive thermal energy exchange towards the cells. One of the major sources of this process is expected to be the heat produced by friction that reverses almost entirely in the vane (being the machine insulated). A secondary contribution could be given by the latent heat of condensation released when the superheated vapor impinges onto the colder metallic surfaces of the expander: it must be reminded that this contribution has a negative effect related to the reduction of the expansion work of the condensed liquid. The positive contribution due to the heat produced by the friction losses is quite evident if one compares the shaft work with the theoretical work that could be calculated considering the ideal mass entering the machine (Eqn. 2) and an adiabatic isentropic expansion assumed as reference without considering the power absorbed by friction (that in reality decreases significantly the net output work in positive displacement machines).

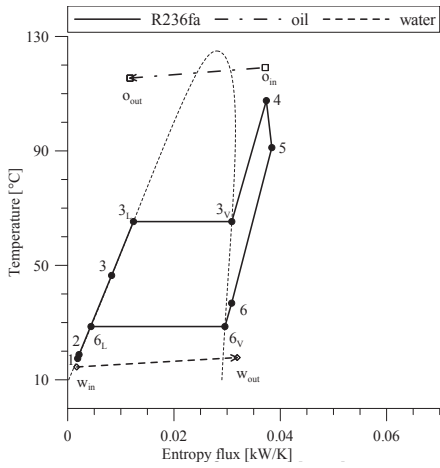


Fig. 5: Test #1 (Low Recovery) - Entropy diagram

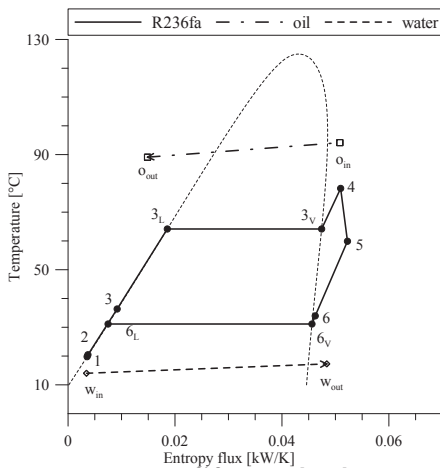


Fig. 7: Test #2 (Medium Recovery) - Entropy diagram

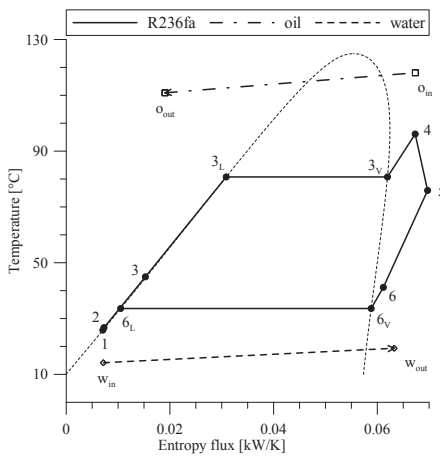


Fig. 9: Test #3 (High Recovery) - Entropy diagram

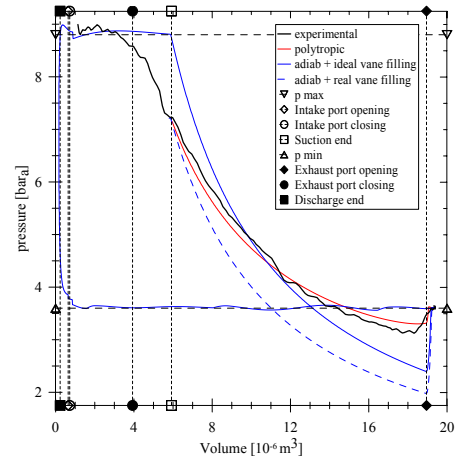


Fig. 6: Test #1 (Low Recovery) - Indicated diagram

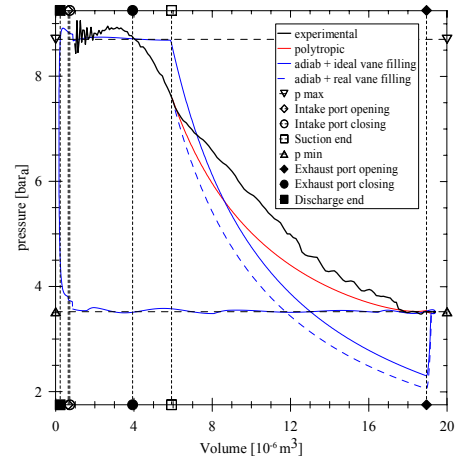


Fig. 8: Test #2 (Medium Recovery) - Indicated diagram

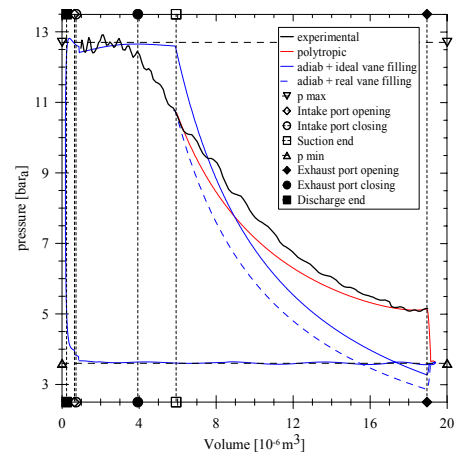


Fig. 10: Test #3 (High Recovery) - Indicated diagram

Shaft work for the three cases is respectively 0.76 kW, 1.01 kW and 1.47 kW while ideal work is 1.09 kW, 0.87 kW and 1.47 kW. The most evident situation happens in case 2 (design situation): ideal work (0.87 kW) is lower than real work (1.01 kW) in spite of:

- a real mass flow rate inside the vane that is lower than the ideal one (the pressure during filling decreases with respect to an ideal isobaric process);
- the presence of friction losses that certainly worsen the real situation with respect to the ideal one.

Figure 8 shows how the real expansion recovers the losses mentioned before. Starting from the pressure level at the suction end, the real expansion significantly differs from the adiabatic law that would have led the fluid at a pressure even below the one at the condenser. This fact would have implied a negative effect associated to a recompression at the expander exhaust. From this point of view, real expansion in test #2 matches very closely the condenser pressure, without any loss. This does not happen neither in Figure 6 where real expansion falls below the condenser pressure nor in the case reported in Figure 10 where the higher initial pressure at the suction end leads the WF to a pressure at the beginning of the discharge process higher than the condenser one. A sudden expansion occurs producing useful work but losing the part that would have been produced if the vane had opened at a pressure equal to the one at the condenser. Therefore, a substantial improvement is expected from a new expander design that is currently under development.

6. Conclusions

Waste heat recovery into mechanical energy from low grade temperature sources has been extensively treated in literature. A useful, universally recognized solution, is to use the low grade thermal energy in a small plant in which a fluid (usually having an organic nature) is vaporized at high pressure, expanded producing work, then condensed and re-pressurized, according to a Rankine cycle (slightly superheated too). The issues related to this technique are mainly due to the pump and the expander technologies. Indeed, the target mechanical power recovered limits the use of the available expanders and pumps so calling for new improved solutions. Positive displacement machines proved to be the most suitable technology. Among them, the rotary vane types fulfill other important properties like simplicity, reliability, reduced cost, reduced maintenance, degrees of freedom concerned to the machine geometry.

The paper presents the development of a new rotary vane expander that has been designed thanks to a comprehensive mathematical model derived from previous studies performed by the authors on sliding vane rotary compressors. Its performance has been experimentally assessed together with the one of an ORC plant to recover the waste heat from the lubricant of a sliding vane rotary compressor. At design point the efficiency of the recovery system achieved reached the 7.6%. Off-design conditions have been also reproduced varying the WF mass flow rate and the thermal power recovered. Even in these situations, the overall cycle efficiency was at least 5.9%.

In order to understand its behavior, a set of piezoelectric pressure transducers was mounted on the expander and allowed a complete reconstruction of the indicated diagram especially during the vane filling. The output mechanical power that the prototype was able to generate reached up to 1.5 kW. However, this result depends on the inlet absolute enthalpy of the superheated steam. Hence, the maximum energy recovery does not coincides with the maximum cycle efficiency.

The experimental campaign eventually allowed to address some performance improvements needed to reduce leakages between adjacent cells and across the expander covers. A proper mixing of the oil entrained with the WF along all the plant sections would certainly reduce leakages.

The p-V data during the closed volume expansion allowed to appreciate the benefits of a real expansion in which heat is reversed inside the cell because of friction losses. In fact, a real expansion differs significantly from a reference adiabatic isentropic transformation that allows a closer matching at the exhaust with the condenser pressure. During the real vane filling, pressure decreases instead of staying constant: this was due to leakages but also to a partial condensation of the vapor that inevitably impinged on the colder metallic surfaces of the vane. A better insulation will be certainly useful as well as some technological arrangements to keep the metallic surfaces hotter than the fluid entering into the vane. All these improvements are under development.

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