# EXTENDED ABSTRACT: AN EXPERIMENTAL ANALYSIS OF A LOW-LOSS RECIPROCATING PISTON EXPANDER FOR USE IN SMALL-SCALE ORGANIC RANKINE CYCLES

**Ilaria Guarracino, Richard Mathie, Aly Taleb, Christos N. Markides***

Department of Chemical Engineering, Imperial College London
South Kensington Campus, London SW7 2AZ, UK
*e-mail: c.markides@imperial.ac.uk

## INTRODUCTION

Small-scale turbo-expanders, which are conventionally used in organic Rankine cycles (ORCs), are associated with lower efficiencies and higher costs. This is problematic since ~30% of the total exergetic loss in a typical ORC arises in the expander (Mago et al., 2008). Instead, positive-displacement expanders may exhibit improved efficiencies when used in small-scale systems (Balje, 1962). Amongst the different types of positive-displacement expanders, reciprocating piston expanders are expected to achieve efficiencies as high as 70% for power outputs up to 3 kW (Glavatskaya et al., 2012). Although technically mature turbomachines (incl. microturbines) are available for ORC systems, positive displacement expanders in general only exist as prototypes, and even these are not piston but rotary expanders (Quoilin et al., 2009). Additionally, there are very few recently published studies on reciprocating piston expanders for ORCs. This study is aimed at investigating the performance and efficiency of this expander type.

The following occurrences have adverse effects on the efficiency of piston expanders: (1) pressure losses and fluid-dynamic dissipation, (2) heat transfer during ideally adiabatic processes and (3) working fluid leakages (Badami et al., 2009). This is especially significant for the expander's valve system (Hunter, 1946). Expansion valves are often more complicated than valves for compression machines as they need to be actively held open against a flow that induces closure forces. Cam and poppet valves are conventionally utilized for reciprocating machines, however they require large actuation forces when the pressure difference across the valve is significant. This results in a heavy system and large mechanical losses. The poppet valve must also be arranged so that the pressure forces act to seal the valve when closed. This can result in hard to achieve mechanical constraints. Conventionally, slide valves are also used in such expanders however both these and poppet valves have limited valve open area leading to high flow velocities and high pressure losses (Howes et al., 2012). In conclusion, the efficiency of a reciprocating expander is strongly affected by the operation of its valve system, and thus, improved valve designs and control strategies for the valve timing are crucial in allowing the expander, and by extension the ORC engine, to operate with optimal performance (Wronski et al., 2012).

This paper focuses on the testing of a reciprocating expander equipped with a novel rotary valve porting system for use with an ORC system with R245fa. Performance indicators are measured from which performance maps can be compiled. Such maps are readily available for turbo-machines but do not exist for piston expanders in low power ranges. A commercial air-compressor has been adapted for use as an expander, with a valve assembly specifically designed for this configuration. The system's scale has been designed to generate up to 3 kW of power, for small-scale applications.

## EXPERIMENTAL METHODS

### Experimental setup

The main components of the ORC test rig are illustrated in Fig. 3(a). The components were designed for a maximum working fluid temperature of < 150 °C, a pressure limit of 10 bar and water at ambient temperature as cooling medium. These are the expected operating conditions using a domestic solar collector. Expander parameters are listed in the table in Fig. 2. The working fluid selection was based on the need for a high thermal efficiency, a low vapour expansion ratio and a high specific power output as well as being non flammable and non toxic, which is desirable for domestic applications (Liu et al., 2012). The specific work and thermal efficiency were calculated with the Engineering Equation Solver software package for a Rankine cycle with superheating to 130 °C at 10 bar, no subcooling, 100% isentropic efficiency for the pump and expander, and no pressure losses. The pressure at the expander outlet was set equal to the saturation pressure of the working fluid at 17 °C. R245fa resulted in the highest thermal efficiency and a condensation pressure of about 1 bar at 17 °C (Figs. 3(b) and 4(a)). Additionally, its volume ratio was lower compared to non-refrigerants, which allows for a smaller expander. The heat required for R245fa to evaporate (equal to the vaporization enthalpy) is similar to that required for non-refrigerants at higher specific power outputs.

**Rotary valve design for a two-stage two-piston reciprocating expander**
Compared to the conventional valve types mentioned earlier, rotary valves offer several advantages, as noted by Hunter (1946), including: minimal high pressure loads on the actuating mechanisms; higher pressure ratios; simple variable cut-off mechanisms; low operation noise and low friction losses due to absence of reciprocating motions; reduced dead volume in the cylinder since valves do not need to protrude in the chamber.

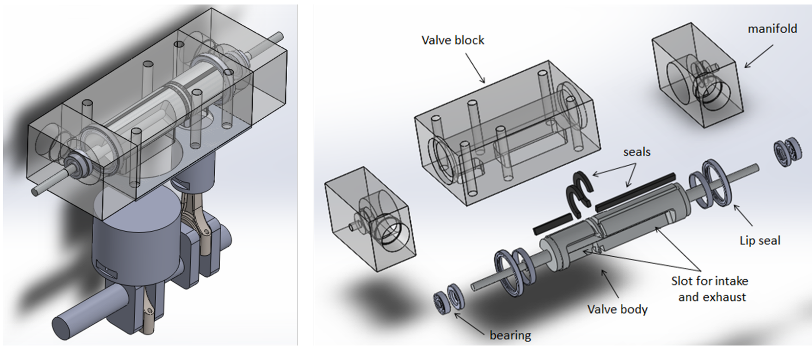
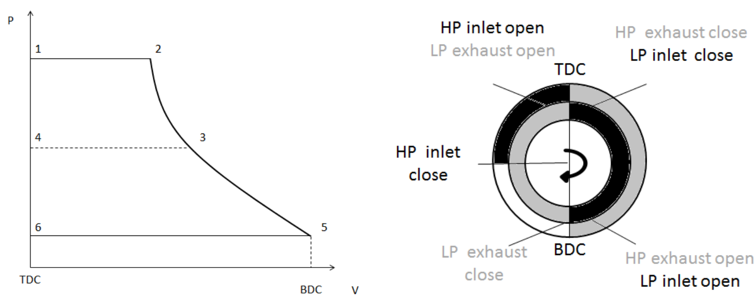



**Figure 1:** Left: valve block assembly on cylinder head. Right: exploded view of the valve.

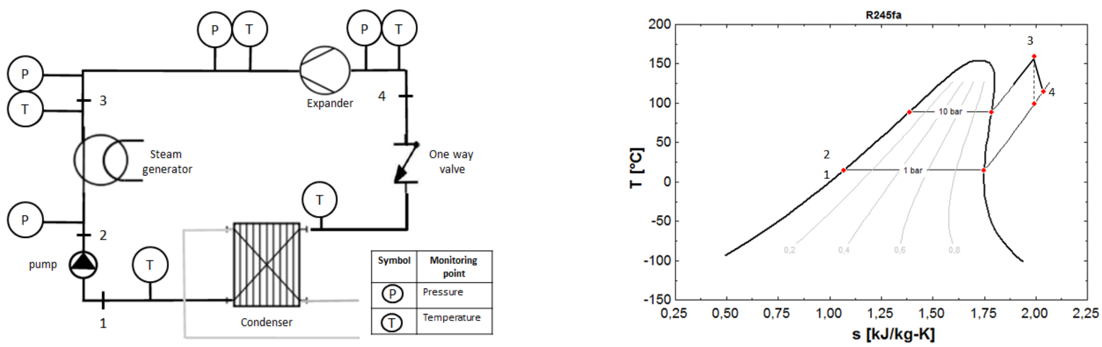
The valve timing is defined as the angle of rotation done by the crankshaft while the valve is opening and is determined by the operational parameters of the expander. It can be derived from the piston position represented in the *P*-*V* diagram (Fig. 2; left).




| Design parameters | Design values |
|---|---|
| Pressure ratio | 10 |
| $V_2$ (cc) | 83 |
| $V_3$ (cc) | 157 |
| $V_5$ (cc) | 692 |

**Figure 2:** Two-stage piston expander (from left): P-V diagram; valve timing; parameters.

At State 1, the high pressure (HP) piston is located at the top dead center (TDC) and starts moving down towards the bottom dead center (BDC). Simultaneously, the HP inlet valve aperture opens allowing gas to flow into the HP piston while moving from State 1 to 2 in the $P$-$V$ diagram. That is, the slot for the gas intake to the HP cylinder starts to open at 1 and will be fully obstructed by the rotating valve body after State 2. Once the HP piston has completed its expansion stroke to the BDC, at State 3, the HP exhaust and LP inlet aperture open to allow gas exchange between the HP and LP cylinders. The same process (States 1 to 3) is repeated with the gas in the LP cylinder, where the gas expands further and is finally expelled from the engine (State 3 to 6). During the next phase of the stroke the HP piston travels back up to the TDC whilst the LP piston returns to the BDC. As the pistons return to their initial positions the gas volume is expanding from States 3 to 4. Finally when the LP piston continues from the BDC to the TDC the LP exhaust aperture opens and expels the expanded vapour (States 5 to 6). The timing for the HP inlet is determined by the cylinder volume ratio of $V_2/V_3$ for the compressor block and the available pressure of the vapour in the ORC. In our configuration, this timing was equal to $90\,°$. The exhaust of the HP cylinder while the HP piston moves from the BDC to the TDC lasts for $180\,°$ of the crankshaft rotation. Similarly, the intake and exhaust of the LP piston are also $180\,°$.




(a) (b)

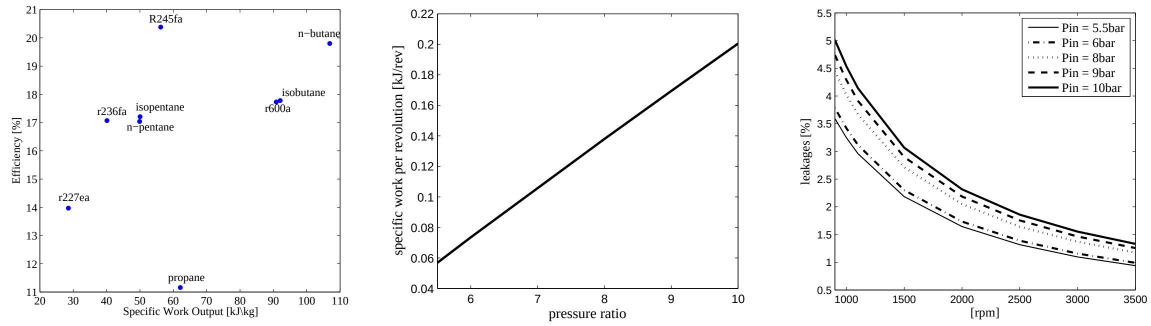
**Figure 3:** (a) Schematic diagram of the ORC system. (b) $T$-$s$ cycle diagram.

To maximize the valve aperture open area, the slot on the valve body must bisect half of the angle for the valve timing. Due to the different valve timing on the HP inlet and exhaust strokes, the HP exhaust valve slot was made $135\,°$ to account for the $45\,°$ valve body aperture required for the HP intake. The LP valve aperture angles were all $90\,°$.

**Measurements**
Pressure and temperature sensors have been installed on the valve casing to measure under- and over-expansion losses and the fluid conditions at the inlet and outlet of the expander (Fig. 3(a)), from which the isentropic efficiency can be evaluated. A heat flux sensor has also been included on the piston head to measure heat transfer in the expander. Experiments are currently under way. The performance of the valve system will be evaluated by measuring the efficiency, the leakage rate and the pressure at the ports. The measured efficiency of the expander will be related to the power output and to operational parameters such as frequency, pressure ratio and mass flow. The results will be presented at the conference.

## PRELIMINARY RESULTS AND CONCLUSIONS

For controllability and experimental flexibility, the ORC is arranged to drive an induction motor that is fed by an inverter and can be operated as a motor or generator. This system enables finer control over the operating rotation speed and load. It is expected that the power output will increase linearly with speed up to the point where losses in the valve




(a) (b) (c)

**Figure 4:** (a) Efficiency and work for different working fluids. (b) Specific work vs. pressure ratio. (c) Predicted worst-case leakage rate for different inlet pressures vs. rpm.

system and the inertia in the vapour stream prove to be detrimental to performance. The leakages for the designed rotary valve were predicted by estimating the friction factor for flow between concentric cylinders where the inner one is rotating (Nakashima et al., 2008). The dominant flow regimes are expected to be a combination of Couette, Poiseulle and Taylor flow. For the valve configuration here, the leakage gap size is assumed to be 50 $\mu$m. This is the smallest possible gap considering the achievable tolerance in our workshop facilities and the thermal expansion of the valve components and is similar to previous works such as Huff and Radermacher (2003). The flow regime is expected to be laminar ($Re < 1000$) with a Taylor number lower than 10, so rotation will not influence axial friction losses. The leakage flow-rate is then determined by the pressure difference between the inlet and outlet channel and the friction factor. The predicted worst case leakage rate for the designed rotary valve system is max. 5% (Fig. 4(c)). This is a conservative estimation and the contact seals should reduce this to an acceptable level.

Final results of the isentropic efficiency and power output as well as a performance map for different operating conditions will be delivered in the conference.

## REFERENCES


Badami, M. and Mura, M., Preliminary design and controlling strategies of a small-scale wood waste Rankine Cycle (RC) with a reciprocating steam engine (SE), *Energy*, 34:1315–1324, 2009.

Baljé, O., A study on design criteria and matching of turbomachines: part A—similarity relations and design criteria of turbines, *J. Eng. Gas Turbines Power*, 84(1):83–102, 1962.

Glavatskaya, Y., Podevin, P., Lemort, V., Shonda, O. and Descombes, G., Reciprocating expander for an exhaust heat recovery Rankine cycle for a passenger car application, *Energies*, 5:1751–1765, 2012.

Howes, J. H. and Macnaghten, J., *Valve*, European Patent 2450602A2, 2012.

Huff, H.-J. and Radermacher, R., 2003. CO2 compressor-expander analysis, s.l.: s.n.

Hunter, M. C., *Rotary Valve Engines*, s.l.: Hutchinson Scientific and Technical Publications, 1946.

Liu, G. L., Xu, J. L. and Zhang, B., Selection of working fluids for different temperature heat source organic Rankine cycles, *Advanced Materials Research*, 614–615:195–199, 2012.

Mago, P. J., Srinivasan, K. K., Chamra, L. M. and Somayaji, C., An examination of exergy destruction in organic Rankine cycles, *International Journal of Energy Research* 32(10):926–938, 2008.

Nakashima, C. Y., Oliviera, S., Caetano, E. F., Calculation of pressure drop in narrow rotating annular clearances, *Thermal Engineering*, 7:27–34, 2008.

Quoilin, S. and Lemort, V., Technological and economic survey of organic Rankine cycle systems, In: *Proceedings of the 5th European Conference on Economics and Management of Energy in Industry*, Algarve, Portugal, 2009.

Wronski, J., Skovrup, M. J., Elmegaard B., Rislå, H. N. and Haglind, F., Design and modelling of a novel compact power cycle for low temperature heat sources, In: *5th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems*, Perugia, Italy, 2012.