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A small power recovery expander for heat pump COP improvement

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Abstract

Heat pumps are becoming more and more applied for heating, due to their possibility of working as cooling systems in the summer period. However, up to now, recovery of expansion work in small system has not been considered as a viable solution, because of the limited amount of recoverable energy and of difficulties in designing and operating a two-phase flow expander.

The idea here presented is to investigate the application of a radial piston machine, adapted from oleodynamic motor designs, as an expander that will be coupled with the compressor motor shaft; the consequent power reduction goes directly to COP improvement.

First, heat pumps working conditions (pressures, mass and volume flows) are defined, and the possible power recovery is calculated. Then, a model for the performance calculation of the radial piston motors used as expander is presented; the model considers the kinematics of the mechanical system and uses real fluid properties.

The results indicate that such a machine could be developed from existing units with limited modifications, encourage to develop a test rig, and to run preliminary experimental work in order to measure the real performance.

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Keywords: heat pumps, expander, throttling valve, radial piston motors, power recovery.

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Nomenclature

Q_{cool}	specific heat removed
Q_{heat}	specific heat supplied
COP	coefficient of performance
T_{HOT}	indoor or heat transfer fluid temperature
Q_{HOT}	heat pump power output
W_c	compressor specific work
ε_{Ccond}	condenser effectiveness-capacitance rate product
ε_{Cevap}	evaporator effectiveness-capacitance rate product
η_{COMP}	compressor isentropic efficiency
η_{EXP}	expander isentropic efficiency
Θ	cranck angle

1. Introduction

Commercial heat pump units operate in base of reversed Rankine thermodynamic cycle, also known as vapour-compression cycles (VCC), where work is used to transfer heat from a cold environment to a hotter one [1]. This work is usually mechanical work, provided by an electricity-powered compressor, sometimes operated under variable rotational speed using a frequency inverter. Nowadays, heat pumps are used worldwide in several applications, both commercial and residential buildings, or in industrial applications; in addition, on the market can be found units from small size (3-5 kW) up to large size (>200 kW) [2]. The basic operational principle of the VCC, along with its pressure/enthalpy chart, is shown Figure 1. In comparison to the basic VCC, many modifications on the cycle have been proposed to increase VCC efficiency, as for example cascade configurations or multiple-compression stages configuration [3-5]. The idea presented in this paper is the development of a volumetric expander able to exploit the pressure drop of VCC that currently takes place in a dissipative throttling valve (point 3-4, Figure 1). The expander will produce power that will be connected to the compressor of the heat pump to reduce its power consumption, and thus to improve the coefficient of performance (COP) of the entire cycle. The idea is schematically represented in Figure 2, showing the basic layout of the heat pump cycle.

The topic of recovering the high pressure of the liquid refrigerant has been investigated by different researchers. Many researchers focused their activity on transcritical configurations using CO2, because of the high condensation pressure (100 bar or more) and the great margin of energy recovery due to the significant required compression work [6-8]. In the current state-of-the-art heat pumps a turbocharger is usually suggested, where the turbine/expander and compressor are attached on the same shaft; however, even at full load, the expansion efficiency is less than 50% [8-9]. Hewitt et al. [10-11] studied and tested a turbine coupled on the same shaft with a rotary compressor. They had some problems in their tests, such as leakage of refrigerant from the turbine to the compressor via the shaft and intense heat transfer between the expander and the compressor sides. Stosic et al. [12-13], on the other hand, developed a screw expander, operating with two-phase fluid, capable of producing work in an organic trilateral cycle used to recover energy from waste heat. In another work [14], they developed a device that replaces the throttling process and recovers power from the two-phase expansion process and directly recompresses a portion of the vapour formed during the expansion. Both the expansion and recompression processes are carried out in a twin screw machine with only one pair of rotors, showing overall expansion-compression efficiency in the order of 55%. These works show that most of the research has been carried on several different technologies (screw, vane or rotary expanders), but none has been carried out on radial piston expanders, which seem instead the most feasible technology for this application, at least for small-scale applications. Also, in most of the studies a CO_2 VCC unit was considered, whereas very few works exist with HFC refrigerants, which are the most common ones.

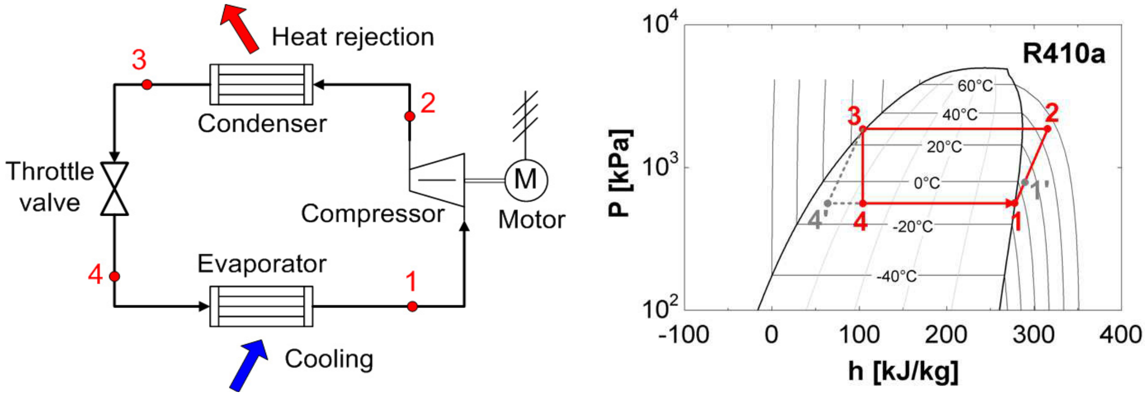


Figure 1 - Simple vapour-compression cycle (left) and pressure/enthalpy chart (right).

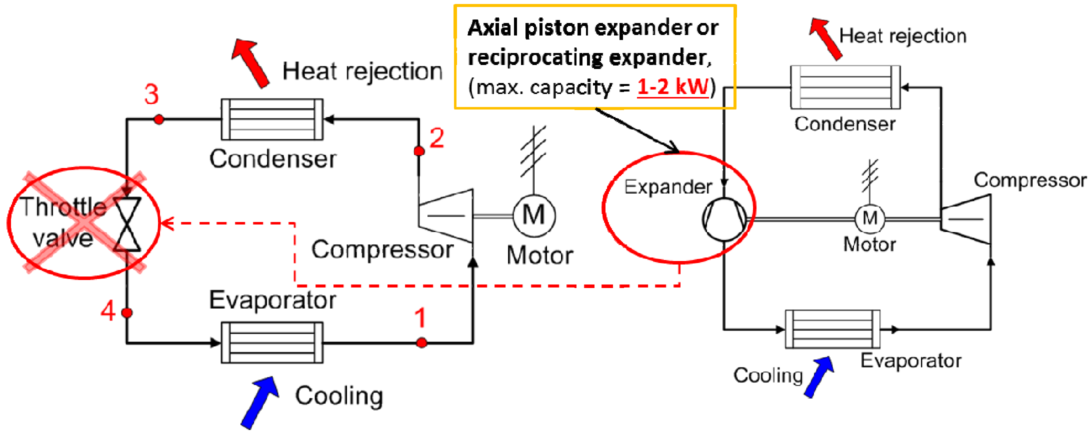


Figure 2 - Basic idea of the throttle valve replacement with the volumetric expander

The COP is normally used to assess the performance of a heat pump unit at cooling mode. The COP value is calculated using the following equation:

$$COP_{cool} = \frac{Q_{cool}}{W_c} \quad (1)$$

where Q_{cool} is the specific heat removed (h_1-h_4), which is actually the effective cooling, and W_c the specific compression work supplied to the vapour-compression cycle (h_2-h_1). COP in the heating mode has a similar formula:

$$COP_{heat} = \frac{Q_{heat}}{W_c} \quad (2)$$

where Q_{heat} is the effective heating (h_2-h_3).

With the proposed modification to the power cycle, the theoretical ideal potential performance improvement of COP should be up to 15 – 20%, depending on various cycle characteristics, in particular the working conditions of the heat pump and the working fluid, as shown in Figure 3.

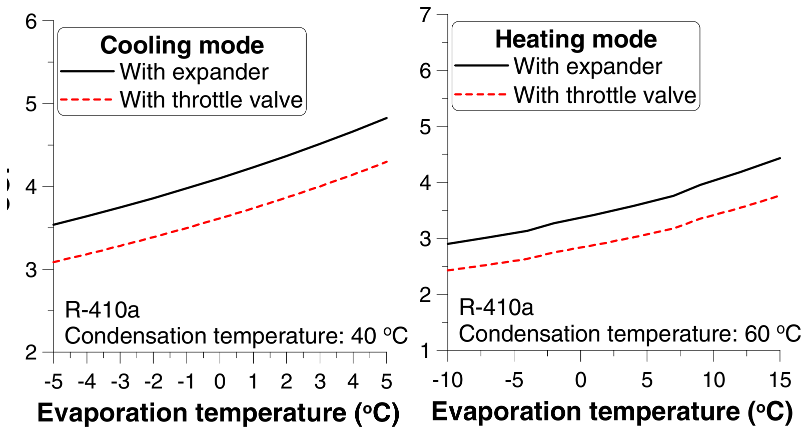


Figure 3 - Theoretical performance improvement of a heat pump by replacing the throttle valve with an expander.

2. Thermodynamic model of retrofitted heat pump

The basic working conditions of the heat pump cycle and the related design data of the expander replacing the throttling valve are calculated by a dedicated thermodynamic code developed into F-Chart® Engineering Equation Solver® software, which is especially suitable to solve these problems, given the very large number of fluid properties libraries. The code, basically developed for solving the standard heat pump cycle with throttling valve, is modified replacing the latter with an adiabatic expansion. The performance parameters are then recalculated considering the contribution of the expansion power output used to reduce the power consumption of the compressor. The condenser and evaporator temperatures are calculated by fixing the temperatures of the hot and the cold heat sinks (indoor or heat transfer fluid of the heaters, and outdoor respectively) and the heat exchangers effectiveness-capacitance rate product at 0.75 for the evaporator (ε_{Cevap}) and 1.75 for the condenser (ε_{Ccond}). In this way, the model determines the thermodynamic states of the actual heat pump cycle in function of the environmental and the room heater temperatures. In all calculations for cooling and heating modes, the efficiency of the expander η_{EXP} is assumed equal to 65%, while compressor/motor efficiency η_{COMP} is fixed at 70%. The isentropic efficiency of the compressor is assumed at the average values for these kinds of machines (usually scroll compressors). The isentropic efficiency of the expander is taken at the same level, which is, as the simulation will show, an overestimation. The reference heat pump power output is fixed at 20 kW_{th}, considering a medium – high domestic size, while the outdoor ambient temperature T_{amb} is fixed at 0 °C, and the room heat transfer fluid temperature T_{HOT} is fixed at 53 °C.

The main thermodynamic data of the heat pump cycle equipped with the expander for four different working fluids obtained for specific outdoor conditions are summarized on Table 1. Figure 4 shows the expander power output and the related improvement in COP (Δ_{COPR}) in function of the ambient temperature, which influences the evaporation (winter) or condensation (summer) temperature, in winter period with a room heater temperature fixed at 53 °C. R1234yf has the highest sensitivity to environmental temperature, while R410a has the lowest.

As shown in Table 1, the maximum potential power output of the expander is 2,1 kW (which means a COP increase of 30%); however, when the real expansion is considered, the potential improvement of COP is reduced to the 12 – 15%. The potential is higher for R1234yf and R410a, mainly due to the higher expander/compressor specific work ratio. Finally, high expansion volumetric ratio, which is an issue for the effective adoption of volumetric expanders, is obtained with all fluids.

Table 1 - Main thermodynamic data of the heat pump cycle equipped with the expander for different working fluids

	Heat pump Working fluid		
Parameter	R134a	R410a	R1234yf
Condenser temperature [°C]	64,4	64,4	64,4
Evaporator temperature [°C]	-9,9	-8,8	-9,0
Condenser pressure [kPa]	1866	4230	1812
Evaporator pressure [kPa]	201	598	230
Compressor output temperature [°C]	98,1	123	78,7
Ideal expander power output W_{expID} [kW]	1,659	2,142	2,169
Real expander power output W_{expR} [kW]	0,995	1,285	1,301
Compressor power demand Wc[kW]	8,828	9,718	9,528
Specific Real expander power output [kJ/kg]	8,73	12,2	8,18
Specific compressor power consumption [kJ/kg]	77,4	92,5	59,9
Expander/compressor specific work ratio [%]	11,3	13,2	13,7
Ideal expander COP increase $\Delta_{COP\ ID}$ [%]	23,1	28,3	29,5
Real expander COP increase $\Delta_{COP\ R}$ [%]	12,7	15,2	15,8
Real volumetric expansion ratio	49,57	17,6	39,95
Condenser/Evaporator pressure ratio	9,33	7,06	7,87
Expander mass flowrate [kg/s]	0,114	0,105	0,159
Evaporator input heat power [kW]	11,2	10,3	10,5
Quality at expander exhaust	0,482	0,526	0,564
COP of retrofitted heat pump	2,553	2,372	2,431
Fluid viscosity at expander inlet τ_4 [CSt]	0,113	0,0839	0,104
Fluid viscosity at expander outlet τ_5 [CSt]	1,043	0,474	0,814

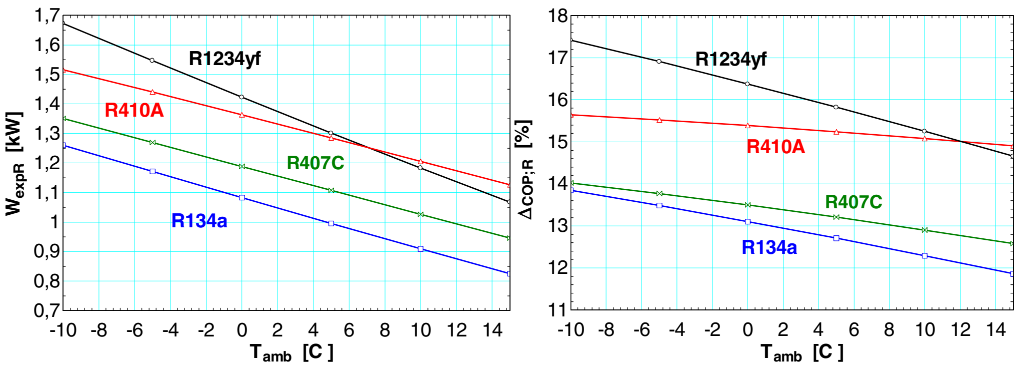


Figure 4 - Real expander power output (W_{expR}) and relative COP improvement (Δ_{COPR}).

3. Expander model

The expander technology here studied for the recovery of the expansion work in heat pumps is radial piston motors (with maximum capacity of 1-2 kW). Radial piston motors are mainly used in hydraulic applications, while they have some interesting features and many positive aspects. In fact, these machines can operate up to very high-pressure range (300 bar or even more), can handle liquids/fluids of very low temperatures (even at -40 °C), and have an high efficiency at normal operation (80-85%). They have high lifetime, low maintenance needs, and they are very reliable machines. Also, their dimensions are low, due to their compact design, and together with their high power to weight ratio, makes them suitable to be integrated in small heat pump units, requiring few space. However, there are some drawbacks. First, un-efficient lubrication can lead to motor overheating, because the usual operating pressures are higher than those of heat pumps. However, in the current application this problem can be very easily solved, since the refrigerant at the expander outlet will have low temperature; thus avoiding the overheating of the expander, and, at the same time, lubricating the motor. A second important drawback can be the cavitation caused by the presence of refrigerant vapour during expansion; cavitation, in fact, can heavily damage the machine. The motor here investigated has 9 pistons displaced in radial configurations, and has a displacement volume of 102 cc.

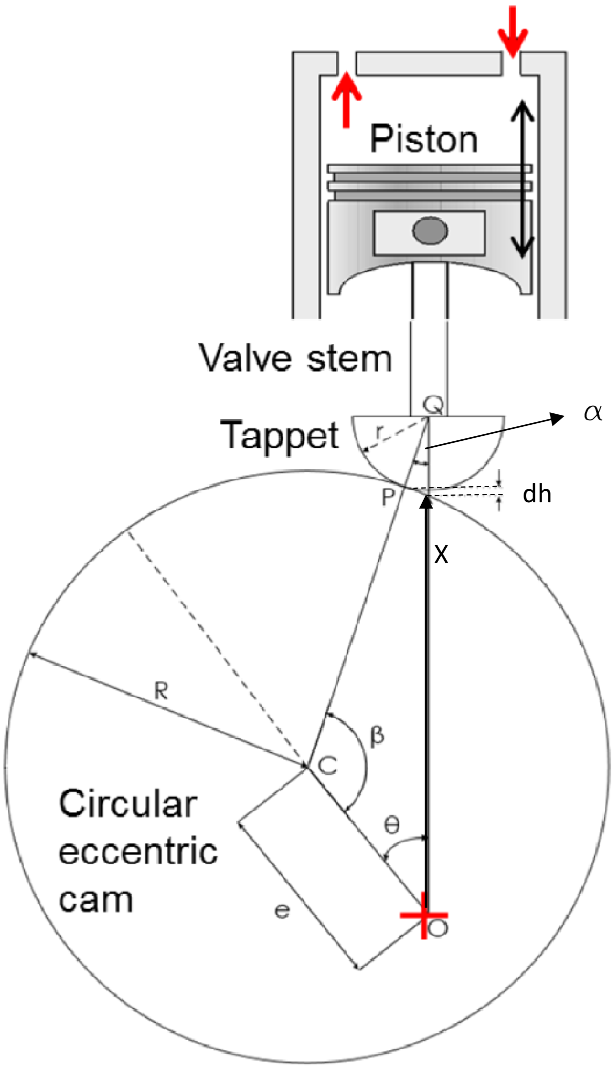
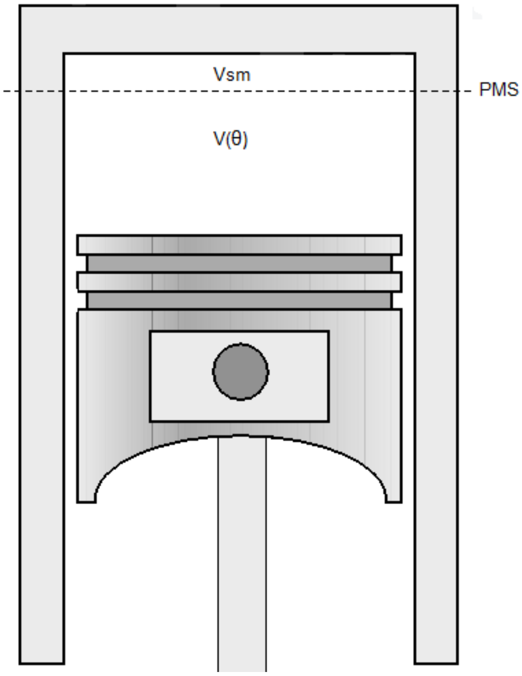


Figure 5 - Kinematic mechanism of the new designed expander.



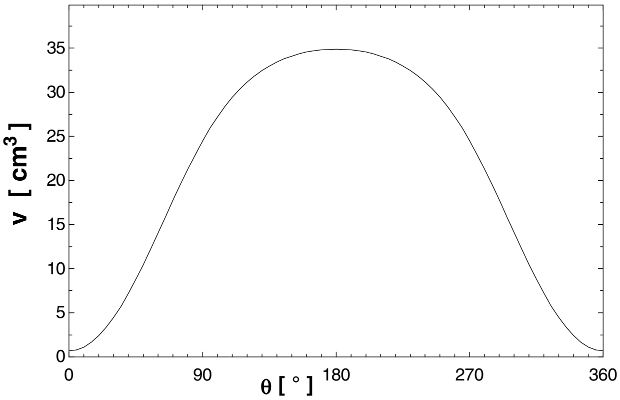


Figure 6 - Cylinder (left); piston volume law (right).

The key aspect of the motor is its kinematic mechanism when used as an expander (Figure 5). The fluid enters the cylinder and moves the piston downward; the piston is connected through a valve stem to the tappet that moves the circular eccentric cam connected to the shaft. This mechanism is essentially the opposite of alternative compressors, where the pistons are driven by the rotating shaft. The piston volume law describes the variation of the volume in the cylinder between the upper and lower dead end; and can be obtained by multiplying the piston head area for the variation of the position of the piston along the vertical axis, calculated with the following equation:

$$x = 70{,}215 - 8 \cdot \cos\theta - \sqrt{64 \cdot \left(\left(\cos\theta \right)^2 - 1 \right) + 62{,}215^2} \tag{3}$$

Where ϑ is the cranck angle and x is measured in mm. As can be seen from Figure 6, the upper dead volume (PMS) is a critical issue, since the fluid contained in it does not contribute to the expansion, and thus does not produce mechanical work.

4. Simulation model of the expander

A preliminary simulation model of the radial piston motor was developed in order to assess its performance as expander. Actually, two softwares were used. A preliminary simulation model of the ideal volumetric expander cycle working with the real fluid (i.e. limit cycle) was developed into EES®. The achieved results were compared with those provided by a model developed with LMS Imagine Lab AMESim®; the results of the two models are very close (Figure 7), which suggests the reliability of the calculations, at least in the basic limit cycle. AMESim® was then used to simulate the behaviour of the expander under nominal conditions, because EES® has some limitations to represent the actual geometry of the expander. AMESim® has numerous libraries of pre-defined and validated components (valves, etc), and also many libraries of equation of state of real gases. However, AMESim software does not allow the estimation of leakage and friction losses and the eventual heat transfer to the expander walls. Anyway, the latter should not be too high given the relatively low working temperature of fluid. The simulations were carried out with these working fluids: R134a, R1234yf, and R410a. The following input data, coming directly from the thermodynamic analysis of the heat pump cycle reported in Table 1, were adopted:

- Condenser temperature: 65°C (saturation pressure for each fluid is given in Table 1).
- Evaporator temperature -10°C (for saturation pressure for each fluid is given in Table 1).
- Heat pump power output: 20 kW.
- Heat pump cycle mass flow rate and compressor power: Table 1.

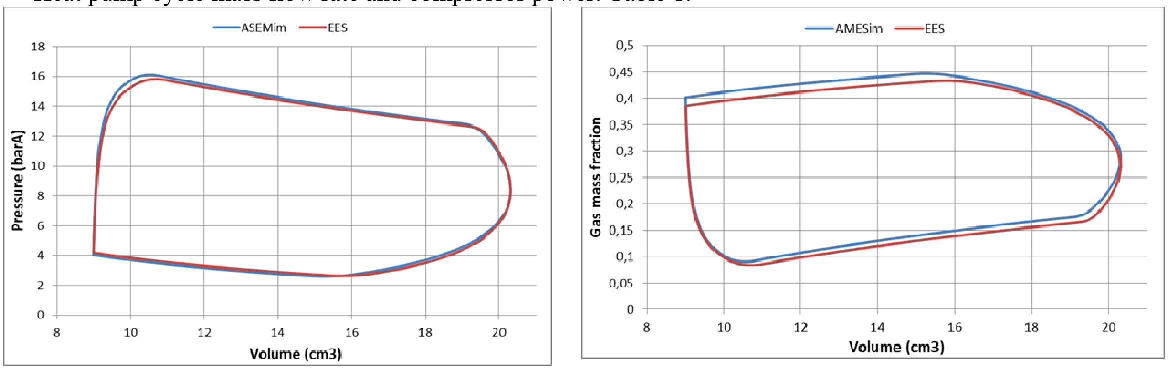


Figure 7 - Comparison of the limit cycle of the expander on volume-pressure and volume-quality planes achieved with EES® and AMESim® simulation tools

Table 2 reports the power output of the radial piston expander as calculated by the simulation model. It is important to notice that there are two results for each fluid. The results in the first column are obtained with the condenser pressure corresponding to 65 °C (Table 2); as can be seen in Table 2, the mass flow rate in the expander is significantly higher than the value calculated for the heat pump cycle for each fluid. Thus, a throttling valve is assumed located before the expander inlet in order to reduce the mass flow rate to 0,114 kg/s; this lowers the suction pressure of the expander to the values reported in the second column of each fluid (for instance, 9.5 bar for R134a). Therefore, the expander power output decreases for each fluid. In conclusion, about 7 % of the heat pump compressor power demand can be recovered by such system.

Table 2 - Expander performance for different working fluids.

	R134a		**R410a**		**R1234yf**	
Suction pressure [bar]	**18,91**	**9,5**	**42,83**	**15**	**18,35**	**9,1**
Mass flow rate in the expander [kg/s]	0,413	0,116	0,528	0,104	0,384	0,105
Expander power [kW]	1,745	0,621	3,923	0,692	1,660	0,556
Specific work [kJ/kg]	4,222	5,347	7,435	6,676	4,323	5,294
Expander/Compressor work ratio [%]	-	7,0	-	7,3		5,7

5. Conclusions

In this paper, the performance of a radial piston motor used for replacing the throttle valve normally used in an heat pump was presented. First, the boundary conditions of heat pump cycles with different working fluids were calculated. Then, the limit cycle of the heat pump with power recovered by the expander was simulated with two different softwares: EES and AMESim®. These two softwares were selected for the following reasons: on one hand, EES is an highly customizable software, on the other hand, AMESim® is commercially available and ready to use, even if it has a modest level of customizability. In both cases, the model considers the real kinematics of the radial piston motor, and uses real fluid properties. The preliminary calculations of the limit cycle of the heat pump showed a high level of agreement between the two tools. Finally, the performance of the radial piston motors used as expander with different working fluid was assessed with AMESim® tool. The results indicated that such a machine could be developed from existing units with limited modifications, and they encouraged to build a test rig to run preliminary experimental work for measuring its real performance. In the future steps, the impact in terms of efficiency of some modifications on the radial piston motor will be investigated with the simulation code developed. An expander modified with the most promising changes will be than produced and tested.

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