PROCEEDINGS OF ECOS 2023 - THE 36TH INTERNATIONAL CONFERENCE ON
EFFICIENCY, COST, OPTIMIZATION, SIMULATION AND ENVIRONMENTAL IMPACT OF ENERGY SYSTEMS
25-30 JUNE 2023, LAS PALMAS DE GRAN CANARIA, SPAIN

Investigation of a Heat Pump using Two-phase Refrigerant Compression

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Abstract:

Two-phase compression in volumetric machines shows several advantages: sealing effects brought by the liquid blocking the gap between two working chambers and it allows to get closer to an isothermal process, reducing the thermal stress on moving parts and minimizing compressor work. However, such compression also comes with some disadvantages: the mechanical reliability of the machine is questioned due to the presence of liquid during the compression, moreover, literature has shown that compressing in the two-phase region usually tends to decrease the isentropic efficiency of the machine.

The irreversibility creation in cycles such as ORC's or vapor compression cycles comes from two sources: the deviation of compression/expansion processes from internally reversible processes and the temperature difference between the hot/cold sources/sinks and the working fluid along heat exchanges (external irreversibility). Therefore, two-phase refrigerant compression/expansion can be integrated to pursue a beneficial trade-off between internal and external irreversibility, searching to increase the performance of the cycle by allowing to match as close as possible the temperature profile of the hot/cold sources/sinks. This paper presents an investigation on vapor compression heat pump cycles where the pressure increase is performed by a two-phase compressor. To this aim, a validated semi-empirical model of a scroll compressor tested with two-phase refrigerant is integrated into the cycle model. Moreover, a moving boundary model is used to model the cycle heat exchangers. The overall model is used to investigate the performance of the heat pump cycle and to quantify both internal and external irreversibility. The results show that the exergy destruction rate of the compressor at low vapor qualities are too high to be counterbalanced by the optimal heat exchange in the condenser. Therefore, the maximum coeffient of performance (COP) is located at saturated vapor compressor inlet condition. Nevertheless, the analysis is strongly dependent on the compressor used and a more optimized compressor for the application could enhance the heat pump COP at low qualities.

Keywords:

Two-phase compression, Heat Pump, Performance analysis, Exergetic analysis

1. Introduction

1.1. Two phase compression/expansion: state of the art

Two-phase compressions and expansions can be categorized in two parts: on one hand, two different fluids can be used (e.g., air and water mixture or refrigerant oil mixture), on the other hand, the state of one single fluid can be located under the saturation bell resulting in the presence of a liquid phase and a vapor phase. In the former case, experimental studies have started in the nineteen fifties, where the use of oil to lubricate air screw compressors and expanders was necessary to expand machines lifetime. In the latter case, the first studies can be found in the nineteen seventies where the expansion of flashing liquid and expansion of refrigerant started to gain interest for better machine performance and reliability purposes.

Two-phase expansion/compression have many advantages, in addition to the sealing effects brought by the liquid blocking the gap between two working chambers, it also allows to get closer to an isothermal process, reducing the thermal stress on the moving parts [1]. Nevertheless, some disadvantages can also be found: the increase of the pressure losses in suction/discharge ports due to either the reduction of the speed of sound or the increase of the density but also the loss of mechanical efficiency resulting from the higher viscosity of the liquid phase in contact with the moving parts [8].

Nevertheless, the use of two-phase compression/expansion could bring some innovations to conventional thermal systems. For instance, the Trilateral Flash Cycle shows better power outputs than classical Organic Rankine Cycles or flash steam systems resulting in a reduced system cost (in $/kW) [9]. Despite the potential of two-phase expansion for power generation, it has not been applied on an industrial scale yet, with only a few experimental studies published in the literature. The innovative cycle currently being developed in the REGEN-BY-2 European project (Horizon 2020) is also making use of two-phase refrigerant to operate [3]. In [10], the

https://doi.org/10.52202/069564-0073

investigation of a two-phase compression in a heat pump is conducted numerically using wet fluids and non-azeotropic mixtures with high vapor qualities, and the results show a better COP when no superheat occurs in the cycle.

1.2. Objectives of the paper

In most of the studies where two-phase compression/expansion is used, the performances of the cycles are better than in classical cycles where a superheated state is reached before the compression or the expansion. However, the numerical results obtained strongly rely on the performance of the compression/expansion machine used in the cycle. Using internally reversible processes or not taking into account the increase of irreversibilities in the two-phase region is not always a valid assumption. For these reasons, the present study will numerically assess the performance of a heat pump using an experimentally investigated two-phase compressor. To this aim, experimental data from a commercial compressor tested with two-phase refrigerant flows are used to validate a compressor semi-empirical model. This model is embedded in a heat pump model, making use of a moving boundary model to simulate the heat exchangers. The irreversibility creation in the heat pump comes from two sources: the deviation of compression process from internally reversible processes and the temperature difference between the heat source/sink and the working fluid along heat exchanges (external irreversibility). Therefore, the integration of two-phase refrigerant compression in the heat pump allows to follow a trade-off between internal and external irreversibility and can, in some cases, increase the performance of the cycles by allowing to better match the temperature profile of the heat source/sink.

The investigated heat pump is a theoretical water-water heat pump providing hot water (50°C) with a heat sink power of 5 kW. It uses a high temperature lift (from 15°C to 50°C) from a 15 °C heat source. It uses the refrigerant R1233zd(E) as the working fluid. A diagram of the heat pump can be found in Fig. 1. The evolution of the Coefficient of Performance (COP) of the heat pump will be investigated with a varying vapor quality up to superheated states. An example of Temperature-Entropy and Pressure-Enthalpy diagrams can be found in Figs. 2 and 3. The conventional cycle using a superheat (10 K) at the compressor inlet is the cycle 1-2-3-4 while the cycle 1′-2′-3-4 makes use of a low inlet quality compression and cycle 1″-2″-3-4 a medium inlet quality compression.

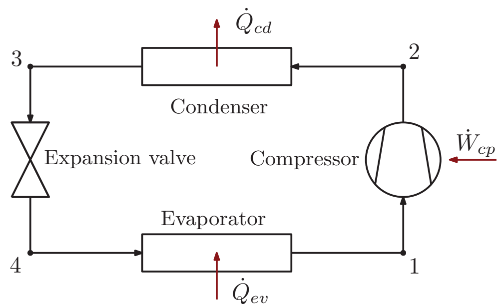

Figure 1: Heat pump cycle.

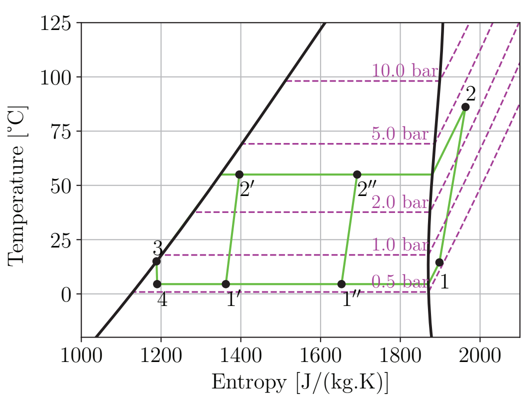

Figure 2: Temperature-entropy diagram of the heat pump cycle with different inlet qualities.

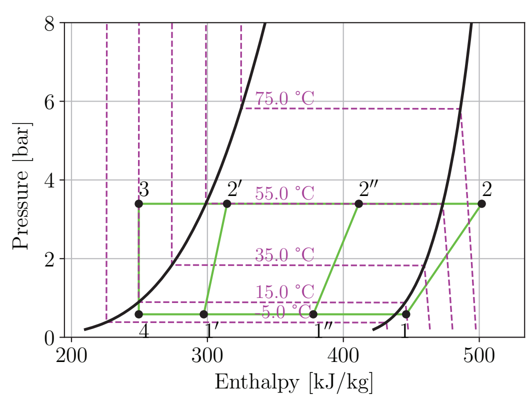

Figure 3: Pressure-enthalpy diagram of the heat pump cycle with different inlet qualities.

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2. Modeling of the heat pump cycle

As previously said, the heat pump is modeled using a semi-empirical model for the compressor and a moving boundary heat exchanger model for the condenser and the evaporator. As presented in Fig. 4, the inputs of the model are the condenser inlet/outlet water temperature ($T_{w,su,cd}$, $T_{w,ex,cd}$) as well as its heat transfer rate ($\dot{Q}_{cd}$), the inlet vapor quality or superheating of the compressor ($Q_{su,cp}$ or SH) and the evaporator water inlet temperature ($T_{w,su,ev}$) and mass flow rate ($\dot{m}_{w,ev}$).

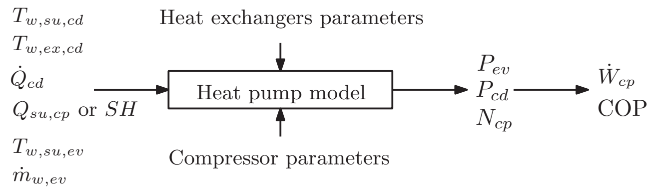

Figure 4: Model of the heat pump.

The model is then applying energy balances on the condenser and the cycle to find the evaporation pressure and the speed of the compressor giving the right working fluid mass flow rate. The condensation pressure could physically be fixed by the charge of refrigerant of the cycle, however, due to convergence problems, it has been decided to fix the condensation pressure with the condenser outlet temperature considering a pinch point of 5 K as can be seen in Fig. 15. Eventually, after fixing both level of pressure and the compressor speed, the compressor power consumption is determined and the COP can be calculated using 1. Obviously, the aforementioned procedure is an iterative process.

$$COP = \frac{\dot{Q}_{cd}}{\dot{W}_{cp}} \tag{1}$$

2.1. Compressor modeling

2.1.1. Experimental results

The compressor performance can be characterized using its volumetric and isentropic efficiencies. They can respectively be defined using the two following equations:

$$\varepsilon_v = \frac{\dot{m}_{tot}}{N_{cp} V_{disp} \rho_{su,cp}} \tag{2}$$

$$\varepsilon_s = \frac{\dot{m}_{tot}(h_{ex,cp,is} - h_{su,cp})}{\dot{W}_{shaft,cp}} \tag{3}$$

where $h_{ex,cp,is}$ is the isentropic compression exhaust enthalpy of the compressor. In particular, the isentropic efficiency (3) is used to characterize the irreversility creation of the compressor. In the case of an adiabatic and reversible evolution, the isentropic efficiency would be equal to one.

The variation of volumetric and isentropic efficiencies of the compressor with a varying pressure ratio and inlet vapor quality can respectively be found in Figs. 5 and 6, for a fixed inlet pressure, Oil Circulation Ratio (OCR) and speed. Those maps direclty comes from the experimental data used to calibrate the semi-emprical compressor model. The decrease of isentropic efficieny (thereby corresponding to an increase of irreversibilies) can clearly be observed, especially for high pressure ratios where the inlet-outlet volume ratio is getting furthest from the built-in volume ratio of the compressor. More information about the isentropic efficiency interpretation can be found in [6]. A similar behavior can be observed for the volumetric efficiency, although a sealing effect can be observed at low qualities and low pressure ratios.

2.1.2. Semi-empirical model

The scroll compressor model is a semi-empirical model inspired by [7]. The behavior of the compressor model relies on a set of physically-meaningful semi-empirical parameters to fit the experimentally tested compressor performances. The diagram of the semi-empirical model can be found in Fig. 7

This model considers a fictious isothermal envelope interacting with the fluid for supply and discharge heat transfer, taking the power losses of the compression process and rejecting ambient heat losses. It decomposes the fluid transformation inside the compressor in the following steps:

- $su \rightarrow su,1$: isobaric heat transfer from the hot isothermal envelope to the working fluid.

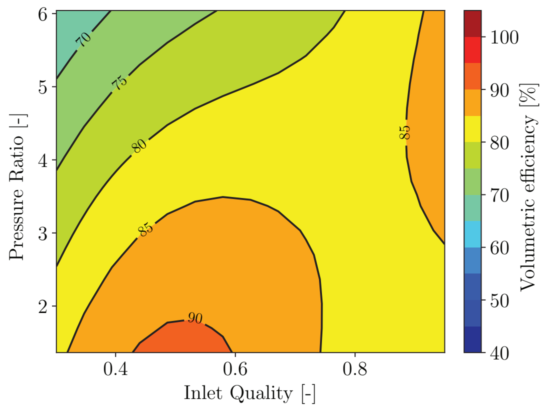

Figure 5: Evolution of the volumetric efficiency with the inlet quality and the pressure ratio for an inlet pressure of 1.5 bar, an OCR of 5% and a compressor speed of 4000 RPM.

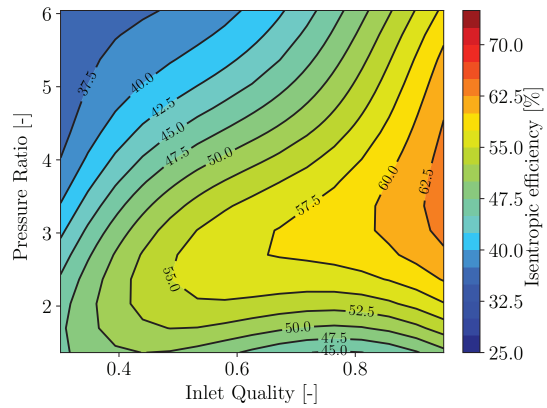

Figure 6: Evolution of the isentropic efficiency with the inlet quality and the pressure ratio for an inlet pressure of 1.5 bar, an OCR of 5% and a compressor speed of 4000 RPM.

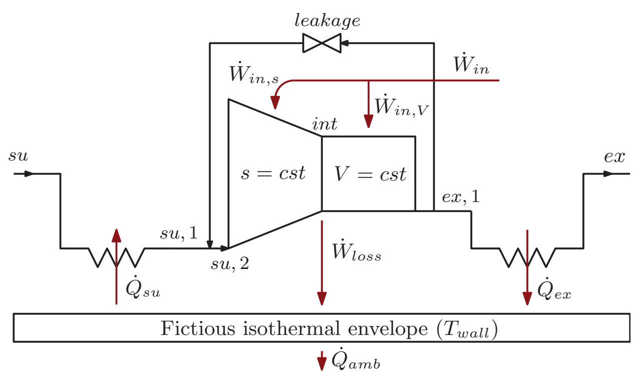

Figure 7: Model of the heat pump.

- $su,1 \rightarrow su,2$: energy balance due to the leakage flow adding to the compressor upstream flow
- $su,2 \rightarrow int$: adiabatic and reversible (isentropic) compression. First part of the compression imposed by the built-in volume ratio of the compressor r_v and consuming a power $\dot{W}_{in,s}$.
- $int \rightarrow ex,1$: constant volume compression to account for under/over compression losses. Second part of the compression consuming/producing a power $\dot{W}_{in,V}$.
- $ex,1 \rightarrow su,1$: leakage flow defined with an adiabatic pressure drop through a convergent nozzle.
- $ex,1 \rightarrow ex$: isobaric heat transfer from the hot working fluid to the isothermal envelope.

The model uses 6 semi-empirical parameters. Among them, 3 are used for the supply, exhaust and ambient heat transfers, respectively, $AU_{su,ref}$, $AU_{ex,ref}$, AU_{amb}. Two other parameters are used for the friction losses characterization $\dot{W}_{loss,ref}$ and α_{loss}. The last parameter A_{leak} is used for the characterization of the leakage flow. The subscript *ref* stands for the reference, On the heat and mechanical power transfer side, the following definitions are set:

- $\dot{Q}_{su} = \varepsilon(AU_{su,ref}, \dot{C}_{wf})\dot{C}_{wf}\left(\frac{\dot{m}}{\dot{m}_{ref}}\right)^{0.8}(T_{wall} - T_{su})$: heat transfer from the hot isothermal envelope to the working fluid. ε is the heat exchanger efficiency computed using the Number of Transfer Units (NTU) method and $\dot{C}_{mix}$ is the heat capacity rate of the mixture flow.
- $\dot{Q}_{ex} = \varepsilon(AU_{ex,ref}, \dot{C}_{wf})\dot{C}_{wf}\left(\frac{\dot{m}}{\dot{m}_{ref}}\right)^{0.8}(T_{ex,1} - T_{wall})$: heat transfer from the working fluid to the isothermal envelope.

https://doi.org/10.52202/069564-0073

- $\dot{Q}_{amb} = AU_{amb}(T_{amb} - T_{wall})$: ambient heat transfer from the isothermal envelope.
- $\dot{W}_{in,s} = \dot{m}(h_{int} - h_{su,2})$: isentropic compression power.
- $\dot{W}_{in,V} = \dot{V}_{int}(P_{ex,1} - P_{int})$: constant volume compression power consumption.
- $\dot{W}_{loss} = \dot{W}_{loss,ref}\left(\frac{N}{N_{ref}}\right)^2 + \alpha_{loss}\dot{W}_{in}$: friction power loss of the compressor.

The calculation of properties considers the oil fraction in the whole working fluid path. Thereby, every enthalpy, entropy, density calculated takes into account the oil fraction similarly to what is used in [11]. Let X be a property of a pure fluid, the mixture property calculated would use the following equation:

$$X_{mix} = QX_{r,vap}(P,T) + z_oX_o(T) + (1 - Q - z_o)X^{\sigma}_{r,liq}(T) \tag{4}$$

where the subscripts r refers to pure refrigerant properties, *vap* and *liq* respectively refer to the vapor and the liquid phases, o stands for the oil and σ refers to a saturated property. The vapor quality Q, defined as vapor mass divided by the total mass, is linked with the oil fraction z_o using the Raoult law solubility equation. As can be noticed, 4 assumes that the oil only stands in the liquid phase.

The model takes five inputs: the compressor speed (N), the oil circulation ratio (OCR, z_o), the inlet pressure (P_{su}), pressure ratio (r_p) and the inlet quality (Q). It gives as outputs the mass flow rate and the power consumption at the shaft.

2.1.3. Model validation

As already mentioned, the compressor semi-empirical model is validated against experimental data. The experimental data comes from a commercial compressor with a displacement volume of 86 cm^3 and a built-in volume ratio of 2.3, tested with two-phase conditions on 110 operating points. The model validation is performed with the shaft power consumption and the mass flow rate, as those are the two relevant variables used in the heat pump model. The validations with regards to the shaft power and the mass flow rate can be found in Figs. 8 and 9.

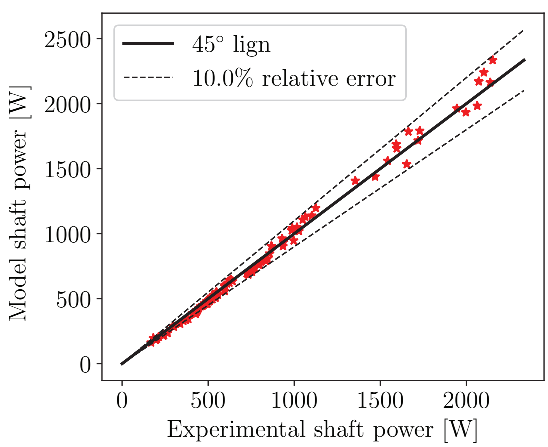

Figure 8: Shaft power prediction of the compressor model versus experimental data.

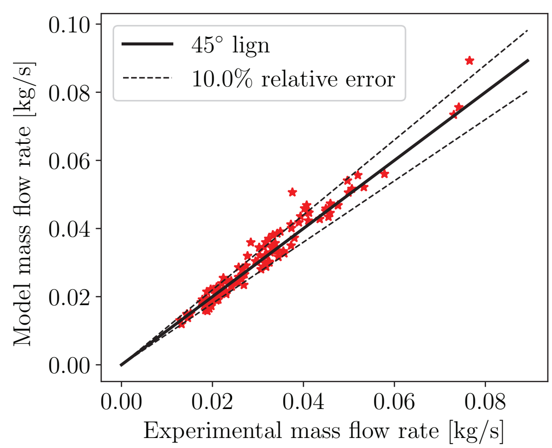

Figure 9: Mass flow rate prediction of the compressor model versus experimental data.

As can be observed in Figs. 8 and 9, the power predictions remain within the range of 10% of relative error with regards to the experimental power while the mass flow rate prediction is less precise, in particular, three points are going way outside the 10% relative error range. Nevertheless, the mean relative errors of the whole set of data are equal to 4.46% for the power and 6.47% for the mass flow rates.

Finally, the tuned parameters as well as the characteristics of the tested compressor can be found in Table 1. It can be noticed that the inlet and outlet heat transfer coefficients are abnormally high compared to what can be found in the litterature (between 1 and 10 W/K). On one hand, it can be explained by the two-phase conditions that increases the heat transfer coefficient. On the other hand, it could be explained by the lack of a suction pressure loss model. In particular, the model increases the inlet heat transfer to the fluid to increase its vapor quality to get lower mass flow rate, and adding an inlet pressure loss would have the same effect. However, the inlet pressure losses were not added because of a convergence problem. Therefore, even though the semi-empirical parameters seems to predict with a relatively good accuracy the performance of the compressor, their physical meaning can not be trusted in every aspect.

Table 1: Parameters of the compressor semi-empircal model.

Parameter	Value	Unit
$AU_{su,ref}$	72.73	W/K
$AU_{ex,ref}$	46.47	W/K
AU_{amb}	2	W/K
$\dot{W}_{loss,ref}$	75.19	W
α_{loss}	0.0848	-
A_{leak}	$3.84{\cdot}10^{-7}$	m^2
V_{disp}	86	cm^3
r_v	2.3	-

2.2. Heat exchanger modeling

The heat exchanger model used for both the condenser and the evaporator is a moving boundary model inspired from [2], it has been selected for its ability to get the temperature evolutions of both fluids inside the heat exchangers. It calculates the maximum heat rate that can be transferred based on an internal and an external pinching analysis. Then, the heat exchanger is divided into a given number of cells having a fixed power transferred and two boundaries representing the saturated liquid and vapor points are are moving to define cells with subcooled liquid, two-phase fluid and superheated vapor. The log mean temperature difference is then solved on every cell to get the outlet and inlet temperatures on both primary and secondary fluid sides (refrigerant and water). The calculation of the heat transfer coefficients in each cell is different in two-phase regime and single-phase regime. For the two-phase regime, the evaporative and condensation heat transfer coefficients are respectively calculated using the Han boiling and condensing correlations in plate heat exchangers, while for single-phase regimes, the Gnielinski pipe heat transfer correlation is used. More information about these correlations can be found in [5].

The heat exchanger used for the present analysis is a plate heat exchanger and the same heat exchanger is used for both evaporator and condenser. The plate heat exchanger characteristics used can be found in Table 2, they are based on a real heat exchanger, oversized with regards to the application.

Table 2: Parameters of the plate heat exchanger used.

Parameter	Value	Unit
Surface exchange area	5.04	m^2
Number of plates	88	-
Cross section area	$2.1 \cdot 10^4$	m^2
Dimensions (H×L×l)	524×117×232	mm

2.3. Model limitations

The results of the heat pump model presented obviously have some limitations, although the main component, i.e., the compressor, is modeled based on experimental data. These limitations can be listed as follows:

- The compressor model is taking the OCR (z_{oil}) into account, it has therefore been tested with oil and the compressor performances shown in the subsequent section depend on this OCR. However, the heat exchanger model does not take the oil into account in the heat transfer coefficient correlations. Therefore, the oil will not be considered in the heat pump model. To make the conversion between the mixture vapor quality and the pure refrigerant vapor quality and vice versa, the following equation is used for the enthalpy conversion:

$$h_{r,pure}(P) = \frac{h_{mix}(T,P) - z_{oil}h_{oil}(T)}{1 - z_{oil}} \tag{5}$$

  A slight temperature change is therefore observed when doing the conversion. Moreover, only the refrigerant mass flow rate coming from the compressor model has been taken into account, this refrigerant mass flow rate is expressed as follows:

$$\dot{m}_r = \dot{m}_{mix}(1 - z_{oil}) \tag{6}$$

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This theoretical separation is not feasible in reality, as the mechanical separation of the oil and liquid refrigerant is impossible to carry out. Nevertheless, this assumption should have a negative impact on the heat pump performance, as the oil compression work is taken into account in the compressor power. However, lower heat exchange coefficients could counterbalance this effect if the oil was circulated inside the cycle. In every simulation run, a OCR of 5% has been considered.

- The second model limitation comes from the compressor model simulated range that goes out of its testing range. In particular, the inlet pressure testing range lies in [0.8 - 2.3] bar, and the model inlet pressure goes down to 0.7 bar. This low pressure is necessary as the refrigerant used has a boiling point of 17 °C and the water of the heat source has a temperature of 15 °C. Thus, it is impossible to know if the compressor performance is well predicted or not. Nevertheless, the inlet pressure usually does not have a strong influence on the compressor isentropic and volumetric efficiencies.

  The two aforementionned reasons justify the need for a model instead of interpolations of experimental data for the compressor mass flow rate and power predictions.

- The last limitation is the fact that the model is using fixed-design components. For performance comparison, it is not always accurate to compare different conditions with the same designs, as, sometimes, the results are design-dependent. This is why the heat exchanger have been oversized, as the techno-economic aspect is not investigated in this paper. Moreover, the data to fit the compressor model comes from the testing of a commercial compressor, which has not been specifically designed for two-phase flows.

3. Heat pump performance analysis

3.1. Exergetic analysis

As stated by the second law of thermodynamics, the performance of a heat pump are limited by the efficiency of the reverse Carnot cycle. The performance gap between the ideal heat pump cycle and the real cycle can be evaluated using an exergy analysis, where the heat pump irreversibility creation can be assessed analytically. Exergy can be defined as the maximum work recoverable from a process with regards to a reference temperature and pressure. For reversible processes, the exergy is conserved, while when irreversibilites occur, a part of the exergy is destroyed. The exergy analysis allows to evaluate irreversibility losses in each component in the heat pump, the best performance of the heat pump will therefore correspond to the minimal total exergy destruction. The exergy of a point in the cycle can be seen as a thermodynamic property, defined as:

$$e = (h - h_0) - T_0(s - s_0) \tag{7}$$

where h_0, T_0 and s_0 are respectively the reference values of enthalpy, temperature and entropy. Similarly to the heat pump exergetic analysis performed in [4], the exergy destruction rate in each component of the cycle can be evaluated using the following equations:

$$\dot{E}_{D,cp} = \dot{W}_{cp} - \dot{m}\left[(h_{ex,cp} - h_{su,cp}) - T_0(s_{ex,cp} - s_{su,cp})\right] \tag{8}$$

$$\dot{E}_{D,cd} = \dot{m}\left[(h_{su,cd} - h_{ex,cd}) - T_0(s_{su,cd} - s_{ex,cd})\right] - \dot{Q}_{cd}\left(1 - \frac{T_0}{\overline{T}_{w,cd}}\right) \tag{9}$$

$$\dot{E}_{D,valve} = \dot{m}\left[-T_0(s_{su,valve} - s_{ex,valve})\right] \tag{10}$$

$$\dot{E}_{D,ev} = \dot{Q}_{ev}\left(1 - \frac{T_0}{\overline{T}_{w,ev}}\right) - \dot{m}[(h_{ex,ev} - h_{su,ev}) - T_0(s_{ex,ev} - s_{su,ev})] \tag{11}$$

where the temperatures of the heat sink $\overline{T}_{w,cd}$ and the heat source $\overline{T}_{w,ev}$ are calculated using a logarithmic mean (isentropic mean), with:

$$\overline{T}_{source} = \frac{T_{w,su} - T_{w,ex}}{\ln\left(\frac{T_{w,su}}{T_{w,ex}}\right)} \tag{12}$$

As already stated in the previous section, the irreversibility creation in the compressor mainly comes from the heat transfer and the pressure losses. Moreover, the mechanisms in contact, such as the scroll and the bearings, creates friction which is also a source of irreversibility. The tendency observed is a drop in isentropic efficiency, i.e. an increase of irreversibility creation when decreasing the inlet quality.

Regarding the heat exchangers, the irreversibility creation primarily comes from the temperature difference standing between the working fluid and the secondary fluid. Pressure losses can also create irreversibilities in the heat exchangers, however, they have not been taken into account in the present analysis. By going from a superheated state to a two-phase state down to a low vapor quality at the compressor inlet, the outlet quality of the compressor is also going to decrease, allowing to better match the temperature profile of the heat sink, which reduces exergy destruction. The objective of the analysis is therefore to optimize the system with regards to the inlet quality of the compressor to minimize the total exergy destruction.

3.2. Results

As already stated, the studied model is a water-water heat pump model. To match the profile of the subcooling at the end of the condenser, the condenser supply water temperature is the same as the heat source temperature, i.e., 15 °C, and the exhaust available water temperature is 50 °C, resulting in a temperature lift of 35 K. The condenser thermal capacity, input of the model, is varied in order to investigate the influence of the design (compressor, heat exchanger) on the heat pump performance. Varying the nominal point by keeping the same design is actually equivalent to a variation of design for a constant nominal point. As a reminder, the inlet quality of the compressor is varying and the model adapts the evaporation pressure and the compressor speed, directly impacting the working fluid flow rate, to keep a constant water exhaust temperature and condenser capacity. The evaporator water mass flow rate has been fixed to 0.3 kg/s to avoid evaporator exergy destruction and to focus the analysis on the compressor-condenser exergy destruction trade-off.

The evolution of the COP with the compressor inlet condition can be found in Fig. 10. The corresponding compressor speeds and pressure ratios (the evaporation pressure being an output of the model) can be found in Fig. 11. The COP evolution tendency is clear, the maximum of the COP stands near the saturated vapor point. Furthermore, for a high condenser capacity, the performance seems to follow a plateau between an inlet quality of 0.6 and 1, on the contrary to a low condenser heat transfer rate where the COP sees a consequent drop at low qualities. For a high condenser capacity, the speed of the compressor is higher, which increases its isentropic efficiency at low qualites. Regarding the evolution of the pressure ratio, it seems that higher pressure ratios are needed when having a superheated inlet state, simply because the water supply temperature is fixed and must be higher than the compressor inlet temperature.

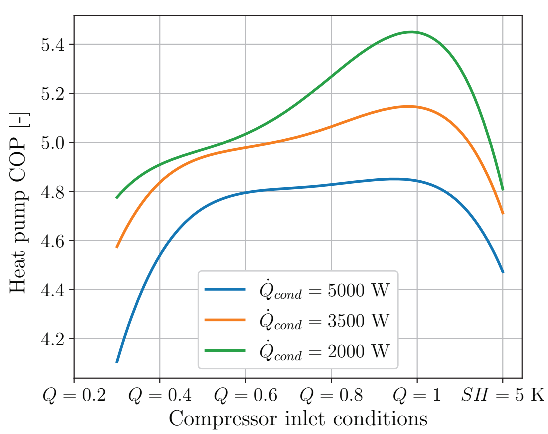

Figure 10: Heat pump COP for a varying compressor inlet condition for three different condenser powers.

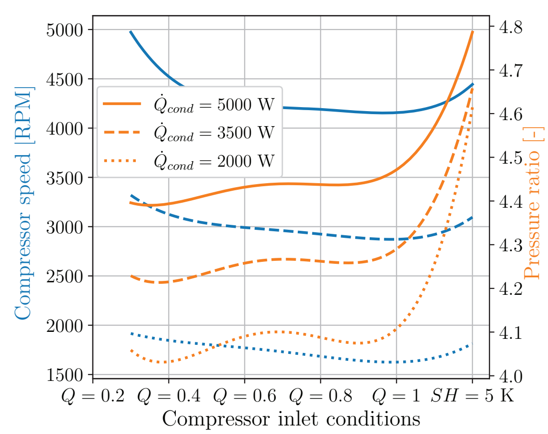

Figure 11: Compressor speed and pressure ratio for a varying compressor inlet condition.

The exergy destruction rate of each component for a 5000 W and a 2000 W condenser capacity can respectively be found in Fig. 12 and 13. In both cases, the exergy destruction of the evaporator and the valve are negligible, and the trade-off is well focused on the compressor and the condenser. When starting from a low vapor quality, the compressor exergy destruction rate is the highest, and it reaches a maximum at a vapor quality of 1 to eventually re-increase due to the increase of pressure ratios for superheated inlet states. Regarding the condenser curve, it is clearly lower for low vapor qualities, due to the better match with the condenser water curve, and increases when increasing the vapor qualities. The total exergy destruction is perfectly matching the tendency of the COP in Fig. 10, the COP maximum corresponds to the total exergy destruction minimum. Furthermore, Figs. 14 and 15 show the evolutions of the water temperature inside the condenser and the evaporator, respectively for a 0.3 quality and a 5K superheated inlet point. The temperature difference inside the evaporator can clearly justify the low exergy destruction rate in this heat exchanger. The same conclusion can be drawn on the condenser, where the exergy destruction rate is lower for the low quality compressor inlet condition due to the proximity between the temperature curves.

Finally, it seems that for both condenser capacities, the compressor exergy destruction rate variations with the

https://doi.org/10.52202/069564-0073

inlet conditions will always remain higher than the condenser exergy destruction rate variations. Therefore, the total exergy destruction rate tendancy is following the shape of the compressor exergy destruction rate. The present analysis is therefore very sensitive to the compressor performance. Hence, the compressor design has a strong influence on the results, and the current commercial compressor may not be the best candidate to be used with two-phase refrigerant flow. A compressor with a higher built-in volume ratio could better fit for the presented model as its performance would be better for higher pressure ratios. It should also be noted that reducing the water condenser outlet temperature could reduce the pressure ratio and enhance the heat pump performance.

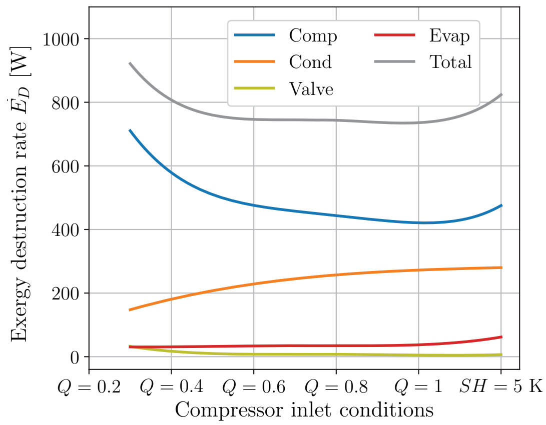

Figure 12: Exergy destruction rate of each heat pump component for a condenser power of 5000 W.

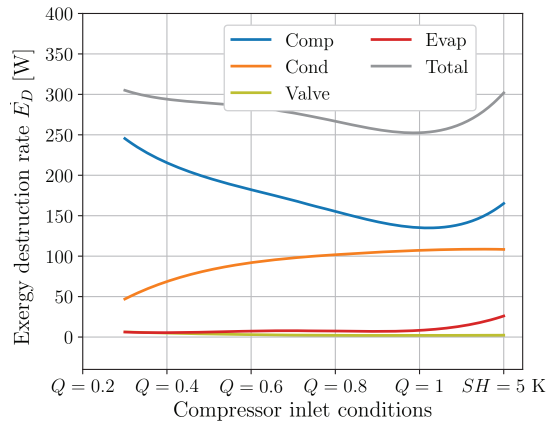

Figure 13: Exergy destruction rate of each heat pump component for a condenser power of 2000 W.

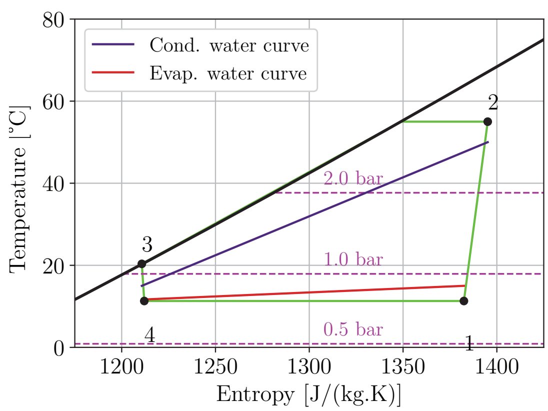

Figure 14: Temperature-entropy diagram of the heat pump with a condenser power of 5000 W and a compressor inlet quality of 0.3.

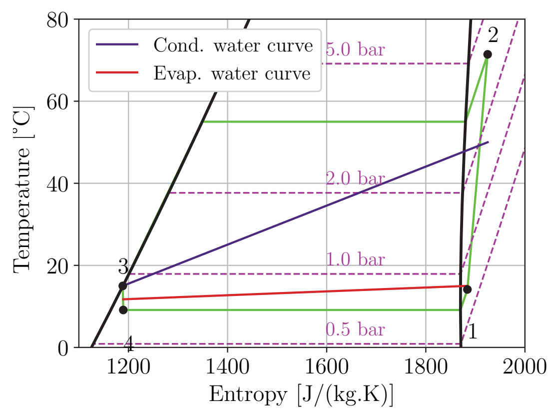

Figure 15: Temperature-entropy diagram of the heat pump with a condenser power of 5000 W and a compressor inlet superheat of 5 K.

4. Conclusions

The current study presents an investigation on a fixed-design water-water high temperature lift heat pump making use of two-phase compression. The objectives of the investigation is the heat pump performance assessment under varying compressor inlet conditions with two-phase refrigerant vapor qualitites as low as 0.3, up vapor with 5 K of superheat. To that end, the heat pump system has been described by coupling a validated scroll compressor semi-empirical model with moving boundaries heat exchanger models.

The model predicts good system efficiency. The performances of the heat pump at different operating conditions are governed by a trade-off between heat exchange and compressor irreversibilities. On one hand, when a low inlet vapor quality is used, the condenser temperature profiles of both fluid (refrigerant R1233zd(E) and water) are getting closer, reducing the irreversibilities. On the other hand, the compressor irreversibility creation is increased when decreasing the inlet quality.

The heat pump model has been tested with a temperature lift of 35 K, from a water condenser inlet temperature of 15 °C to 50 °C and condenser capacities from 2 kW to 5 kW. The results have shown a maximum in the COP curve around a compressor inlet quality of 1. The compressor being the most important source of exergy desctruction, the trade-off could not find better performance at low inlet qualities because the exergy destruction rate of the compressor is too high. Finally, it is important to highlight that the results are very design-sensitive, and the use of a more optimized compressor for two-phase refrigerant flows could enhance the performance of the heat pump and the trade-off could find a lower optimal inlet quality. This potential scenario is the object of future research, as another compressor, specifically designed for two-phase flows with a higher built-in volume ratio is going to be experimentally tested. The integration of the new two-phase compressor performance will allow to see if the trade-off conclusion can be changed.

Aknowledgement

The project source of the results presented in this paper has received funding from the European Union's Horizon 2020 research and innovation programme under grant agreement N° 851541.

Nomenclature

Letter symbols

$\dot{Q}$ heat transfer rate, W

Q vapor quality, $-$

SH superheat, K

T temperature, °C

$\dot{m}$ mass flow rate, kg/s

P pressure, Pa

$\dot{W}$ power, W

COP coefficient of performance, $-$

$\dot{C}$ heat capacity rate, W/K

N rotation speed, RPM

$\dot{E}$ exergy transfer rate, W

V volume, m^3

h enthalpy, J/kg

s entropy, J/(kgK)

e exergy, J/kg

AU heat transfer coefficient, W/K

OCR oil circulation ratio, $-$

Greek symbols

ε efficiency

ρ density

Subscripts and superscripts

cd condenser

ev evaporator

cp compressor

w water

su supply

ex exhaust

v volumetric

s isentropic

tot total

mix mixture

$disp$ displacement

V constant volume

ref reference

amb ambient

r refrigerant

vap vapor

liq liquid

o oil

D destruction

σ saturation

https://doi.org/10.52202/069564-0073

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