SIMULATION OF AN AIR-TO-WATER REVERSIBLE HEAT PUMP

José M. Corberán[1] José Gonzálvez[1] Javier Urchueguía[2]
Ana M. Lendoiro[1]
[1]Department of Applied Thermodynamics,[2]Department of Applied Physics
Universidad Politécnica de Valencia
Camino de Vera 14, ES 46022 VALENCIA,SPAIN
Tel. 34 963877323, Fax. 34 963877329, E-mail corberan@ter.upv.es

Abstract

In this paper a detailed simulation of an air-to-water reversible heat pump working with propane is presented.

The global model of the heat pump is divided in the following submodels; compressor, heat exchangers, expansion valve, connecting pipes and accessories. The governing equations of each submodel are discussed.These equations forms a system of non-linear equations that is solved using a standard newton-like solver.

The results of the model are compared with measurements in a instrumented heat pump. The test are made for a wide range of air temperatures in both modes, cooling and heating. Finally, the model is used to test various designs trying to improve the performance of the heat pump working with propane.

Nomenclature

$\dot{m}$	Mass flow rate	kg/s
A	Area	m^2
h	Specific enthalpy	J/Kg
Nu	Nusselt number	$(-)$
p	Pressure	Pa
Pr	Prandtl number	$(-)$
T	Temparature	K
ρ	Density	kg/m^3
Subscripts		
i	inlet	
o	outlet	

INTRODUCTION

The paper presents a model for calculating the operation point of a vapor compression system. The aim of this model is to combine a high accuracy with a low CPU time in order to be a feasible tool to support the design of new refrigerating plants. To accomplish this objective, the idea is to use simple sub-models for the components that have been well characterized with other more complex models or by experimental data. The model solves the resulting system of non-linear equations by a Newton-like technique. This procedure differs from other simulation programs, like the Frigosim code [7] where the solution is find by iteration.

DESCRIPTION OF THE HEAT PUMP TEST RIG

The heat pump test rig was designed in order to perform the characterization of a medium size reversible air-to-water heat pump. The whole range of climate conditions typical of Southern Europe, were simulated in a climatic chamber in both modes, heating and cooling.

The test rig components included a 50 m^3 climatic chamber, an air handling unit, a hydraulic unit and the heat pump unit to be characterized. All these elements are shown in figure 1. The heat pump unit was situated outside the climatic chamber and the pressure difference between inlet and outlet air was monitored to have nearly constant pressure drop. The air-handling unit, situated on the top of the climatic chamber, was used to create the desired environmental conditions inside the chamber. The outdoor air and the return air from the HP were distributed and mixed into the climatic chamber. The temperature regulation in the climatic chamber was done by means of a control unit controlled the outdoor damper and the recirculating damper using the air temperature measurements in the duct and outdoor. Temperature measurement was accurately done using humidity and temperature transmitter Vaissalla HMP 143. The water loop was connected to the brazed plate heat exchanger of the heat pump unit. The water inlet temperature could be controlled by acting a three-way valve, which controlled the water flow. The water outlet temperature was also controlled using a by-pass.The testing and definition of requirements in heating and cooling mode have been performed according to EN-255-1 and prEN 12055 Standards, respectively. The deviations of the measured values from set conditions were always lower than the permissible specified values in standards.

The heat pump unit was thoroughly instrumented with 12 resistance temperature detectors type PT-100, 12 pressure transducers for measuring absolute pressure Setra 280 E and 2 for relative pressures Setra 228, and a Coriolis mass flowmeter Danfoss M2100di15, in order to perform the characterization. Figure 2 shows the heat pump elements, the piping and all the instrumentation and Table 1 shows the different elements of the heat pump.

Compressor	Coil	BPHE	TEX	Fan	Subcooler
Maneurop MT100HS	ALfa Laval d=9mm	Alfa Laval CB52-46HX	Danfoss TDEX 6	Ziehl-Abegg FB065	Danfoss Coaxial HE

Table 1: Main elements of the tested heat pump

HEAT PUMP MODEL

The heat pump is treated basically as composed of four main elements(compressor,evaporator,condenser and expansion device),the subcooler,the liquid receiver and the connecting tubes among these elements. For the purpose of the modeling, every element is treated as a black box, i.e. relation between the input variables and output variables. The independent variables chosen are pressure and enthalpy in each point.This choice assures a smooth variation of the variables, not given by other choices like pressure and temperature. The main characteristics of the employed models are described in the following sections. Then, these models are coupled to form a global model of the heat pump, which takes into account the matching problem of the global set of equations with special care.

Compressor

For the compressor,two equations are required to characterize its behavior: one for the mass flow rate and one for the outlet enthalpy. The mass flow rate can be calculated as:

$$\dot{m} = \rho_i \dot{V}_s \eta_v \tag{1}$$

where ρ_i is the inlet density $\dot{V}_S$ is the displacement of the compressor in m^3/s and η_v is the volumetric efficiency. The second governing equation of a compressor is:

$$h_o = h_i + \frac{h_{is} - h_i}{\eta_{is}} \tag{2}$$

where η_{is} is the isentropic efficiency, and h_i,h_o and h_{is} are the inlet, outlet and isentropic enthalpy respectively.

A third equation is needed to know the power input to the compressor.

$$\dot{W}_{el} = \frac{\dot{m}\Delta h_{is}}{\eta_{ov}} \tag{3}$$

where Δh_{is} is the enthalpy difference if the compression process was isentropic, η_{ov} is the overall isentropic efficiency that takes into account the power input to the compressor and $\dot{W}_{el}$ is the power input to the compressor.

Measurements on the compressor using propane are used to correlate the volumetric,isentropic and overall isentropic efficiency versus pressure ratio [1]

Evaporator and condenser

Both elements are regarded as heat exchangers in which the mass flow rate and the inlet conditions of both fluids are known and whose model must provide the fluid conditions at the outlet.

For the purpose of the present analysis a simple e-NTU characterization has been chosen, however much more complex and detailed models can also be used in the described methodology. The condenser and the evaporator have been treated as single stream heat exchangers and using the appropriate $\epsilon - NTU$ relations for these exchangers. The temperature of the refrigerant is taken as the saturation temperature corresponding to the inlet pressure and the pressure drop is neglected.

$$h_o = h_i + \frac{\left(1 - e^{NTU}\right) C_s \left(T_{s,i} - T_{sat}\right)}{\dot{m}} \tag{4}$$

where $T_{s,i}$ and C_s are the inlet temperature and thermal capacity of the secondary fluid respectively, and T_{sat} is the refrigerant saturation temperature defined above.

The overall heat transfer coefficient U needed in the evaluation of NTU is defined using the mean logarithmic temperature as:

$$U = \frac{1}{A} \frac{\upsilon_o - \upsilon_i}{\ln \frac{\upsilon_o}{\upsilon_i}} \tag{5}$$

where υ_i and υ_o are the temperature differences between the two streams at the inlet and the outlet respectively. The refrigerant temperature is constant in this model and equal to the saturated temperature defined by the outlet pressure.

The overall heat transfer coefficient is provided by a more detailed model of the HEXs [3]. Both HEX, the coil and the Plate Heat Exchanger, are simulated over a wide range of circulating mass flow rates working as evaporator and condensers.Then, the overall heat transfer coefficient is calculated for each simulation using equation 5. These values are correlated against the Reynolds number of the saturated liquid,$Re_{lo} = GD/\mu_l$. This procedure allows to characterize both HEX.

Expansion device

The expansion device used in this heat pump is a thermostatic expansion valve. It has two governing equations, the first one is

$$T_{ev,o} - T_{ev,sat} = SH \tag{6}$$

where $T_{ev,o}$ is the temperature at the evaporator outlet,$T_{ev,sat}$ is the saturation temperature of the evaporator and SH is the superheat.

The second equation is the assumption of isoenthalpic flow through the valve.

$$h_o = h_i \tag{7}$$

Connecting tubes

The connecting tubes are regarded as heat exchangers with the surrounding air taking into account the pressure drop. This model assumes the inlet conditions and the mass flow rate known, and calculates the outlet enthalpy using an $\epsilon - NTU$ analysis like the heat exchangers. The model distinguishes if the refrigerant entering the tube is in single-phase or in two-phase.

In the first case, the outlet enthalpy is calculated using the $\epsilon - NTU$ analysis for a single stream heat exchanger assuming the air temperature as constant. The refrigerant side HTC is calculated using the

Gnielinski correlation when the flow is turbulent. In laminar flow, the Nusselt number has a constant value of 3.66 for circular tubes. The air HTC is calculated depending on the type of convection process. If there is forced convection, the model uses the correlation of Churchill and Bernstein for external forced convection across a cylinder; in case of natural convection, the model uses the correlation of Churchill and Usagi when the tube is vertical and the correlation of Churchill and Chu when the tube is horizontal. All these correlations can be consulted in [6]

The pressure drop is calculated taking into account the viscous drag and the pressure drop required to accelerate the fluid.

$$p_o = p_i - f\frac{1}{2}\rho_m V_m^2 \frac{L}{D_h} + \rho_i V_i^2 - \rho_o V_o^2 \tag{8}$$

where ρ_m and V_m are the mean values of density and velocity of the refrigerant inside the tube,L is the length of the tube and D_h is the hydraulic diameter.

The friction factor is obtained from the moody chart.The outlet pressure must be obtained by iteration because of the use of the mean velocity and density in equation 8. The model takes the mean density and velocity as the inlet values in the first iteration and then updates the values in the following iterations. The low variation of the properties in single-phase flow makes the model to solve the problem in only needs three or four iterations.

The second case,where the inlet of the tube is in two-phase flow is more complex. The pressure is linked with the temperature and a small pressure drop affects the temperature and therefore the heat transfer process. The procedure used in this case is to solve the problem by iteration. First, the pressure drop is calculated using equation 8 taking the mean values as the initial ones.The friction factor is obtained assuming a liquid flow in the tube as suggested in [6]. The knowledge of the outlet pressure allows to calculate the outlet saturation temperature. An integration of the energy equation gives the outlet enthalpy as:

$$h_o = h_i + \frac{U\mathcal{P}L}{\dot{m}}\left(\overline{T}_{ref} - T_{air}\right) \tag{9}$$

where $\mathcal{P}$ is the tube perimeter.

The next step is to update the mean values of the properties and the next iteration begins. This procedures is used until the outlet enthalpy and pressure do not change. There is a third case considered by the model where the refrigerant changes from two-phase flow to single-phase flow or vice versa. This case is not common in the practice, it only occurs during a bad piping design, but it is implemented in the model because the search a solution can force these conditions in a tube. First,the model evaluates the length needed by the refrigerant to reach the saturated state. If the length is lower than the tubes' length then the tube is calculated like two different tubes, each of them with its corresponding procedure.

Numerical procedure

The selected unknowns of the system are the pressures and enthalpies in all the points(in our case there are nine points). The inlet temperature and mass flow rate of the air and water are imposed as well as the speed of the compressor. The superheat is imposed by the TEX and a subcooling is provided by the subcooler. All these parameters acts as boundary conditions of the system.

A balance between equations and unknowns shows that the system needs a closure equation. The refrigerant mass inventory gives this equation, but as discussed in [2], the existence of a liquid receiver at the condenser outlet fixes this point as saturated and therefore the closure equation is the saturated state of the condenser outlet.

The governing equations of each component described above can be posed in the form $f(p_i, h_i, p_o, h_o, \dot{m}) = 0$. There is not an explicit form of the equations in terms of pressure and enthalpy, but they can be easily programmed using the thermodynamics functions defined in the Refrop Database[4].

The input values to each sub-model are the pressure, enthalpy and mass flow rate at the inlet and the outlet. Then, using the thermodynamic functions,any other thermodynamic property (temperature,density,...)can be known and the output value of the function $f(p_i, h_i, p_o, h_o, \dot{m})$ can be calculated.

At the end, the problem is reduced to calculate the solution to a non-linear system of equations $\vec{f}(\vec{x}) = \vec{0}$. This system of equations is solved using a standard solver provided by the IMSL^{tm} libraries. The routine is based on the MINPACK subroutine HYBRD1, which uses a modification of M.J.D. Powell's hybrid algorithm. This algorithm is a variation of Newton's method, which uses a finite-difference approximation to the Jacobian and takes precautions to avoid large step sizes or increasing residuals. For further description, see [5].

VALIDATION AND PARAMETRIC STUDIES

The model of the heat pump has been validated with the measurements in both modes,cooling and heating.

In cooling mode, the inlet and outlet temperatures of water was set at 7°C and 12°C respectively in all the tests. The air temperature in the climatic chamber was varied from 30°C to 46°C. The superheat was fixed to 3.5°C in all tests and the subcooling was around 5°C in order to assure the entrance of liquid to the Coriolis Massflowmeter.

The water temperatures was set to 45°C at the inlet and 50°C at the outlet in heating mode. The air temperatures was varied from 7°C to 15°C and the relative humidity to 80%. The superheat was 9°C and the subcooling was around 5°C.

The comparison in shown in Figure 3.The predicted COP is within 5% of relative error with respect to measurements.The model under-predicts the capacity of the heat pump although the relative errors are less than 10%.The reason of this mismatch can be the simple models used for the HEX where the pressure drop is neglected. The HEX's models are predicting lower evaporation temperatures than the tests and therefore the inlet density to the compressor is lower, giving less mass flow rate and capacity than the experiments.

After the validation, the model has been used to study possible optimizations in the heat pump working with propane. As commented in the introduction, propane is an alternative to the refrigerant R22. Propane gives less capacity(≈-12%) and a slightly better COP than R22 when retrofitted. Then, the possible optimizations to the heat pump are to obtain the capacity as R22 giving the highest COP possible.

Three different changes to the original heat pump have considered to accomplish this objectives:

Compressor The replacement of the original compressor, Maneurop MT100Hs (171 cm^3 swept volume) by the Maneurop MT125HU(215.3 cm^3) with higher capacity is considered.

Recuperator The use of a recuperator between the superheated vapor and the subcooled liquid is a promising option for propane. From a theoretical point of view, the use of a recuperator in cycles working with propane has a positive effect in the COP. This theoretical analysis is made under the assumption that an increase of the superheat at the inlet of the compressor does not change the evaporating and condensing temperatures. This assumption is not true in a real system and the effect of a recuperator has to be evaluated with the aid of the model.The recuperator tested with the model provides 20°C of subcooling.

Suction tube Propane has a lower molecular mass than other refrigerants like R22 and therefore the pressure drop through the components of the heat pump is lower. The reduction of the suction tube diameter and its impact in terms of efficiency is studied with the model. A reduction of the suction tube from 26.5mm to 9.52mm is considered.

These parametric studies have been performed for the operating air temperatures in both modes, heating and cooling.The results are compared with the original heat pump working without subcooling(the use of a subcooler introduces energy to the system that has to be accounted for).

The comparison in terms of COP and capacity are shown in figure 4. Figure 5 shows the thermodynamic cycles in an enthalpy-pressure diagram.

The use of a different compressor increases the capacity as expected but it has a negative effect in the efficiency of the system. Figure 5 shows that an increase in the capacity leads to higher condensing temperatures and lower evaporating temperatures and therefore a less efficient cycle. The results of the model indicates us that more area of the HEXs is needed if we want to maintain the COP.

The studies with the recuperator shows that the performance of the heat pump working with a recuperator is similar to the original heat pump. The evaporating and condensing temperatures are different with respect to the original heat pump and therefore the benefit in terms of COP predicted by the theoretical analysis is lost. The capacity increases slightly in cooling mode with the recuperator, mainly due to lower vapor quality at evaporator's inlet.

The third study shows a significant decrease of the capacity and COP when the suction tube is changed from 26.5mm to 9.52mm diameter. The pressure drop at the inlet of the compressor leads to less mass flow rate and compressor efficiency than the original heat pump. An intermediate diameter of 15.875 mm has been tested with the model. In this case, the performance of the heat pump is not affected. This test suggest the existence of a critical diameter, below it the performance is affected.

CONCLUSIONS

The development of a model for vapor compression systems is reported in this paper. This model uses simple submodels for each component. The compressor is characterized by its volumetric and isentropic efficiency and the heat exchangers are modeled using a $\epsilon - NTU$ approach.

The model is validated simulating a reversible air-to-water heat pump working with propane. The results of the model are compared with measurements giving a good agreement although a better modeling of the HEXs is desirable to improve the accuracy.

The model has been used to study different designs of the heat pump working with propane. Three different designs have been considered: a different compressor with a higher displacement, the use of a recuperator and the reduction of the suction tube. These designs have been compared with the original heat pump giving the following results:

- The use of a compressor with more displacement increases the capacity as expected but the COP decreases a 10% over the original heat pump. Therefore, a special care must be taken when replacing a compressor to recover capacity because the efficiency can be greatly affected.

- The benefit in COP predicted by a theoretical analysis when a recuperator is used is not shown in the simulation. The differences in predicted COP when using a recuperator are negligible. The capacity increases in cooling mode mainly due to lower vapor quality induced by the recuperator.

- A reduction of the suction tube diameter below a certain value can decrease dramatically the performance of a heat pump because of the lower density at the compressor inlet and the higher pressure ratio.

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References

[1] J. Urchueguia et Al. Performance of an alternative hermetic refrigerant compressor using propane as working fluid. In *$4^t h$ IIR-Gustav Lorentzen Conference on Natural Working Fluids*, 2000.

[2] J.M. Corberán et Al. The matching problem on the modeling of vapor compression systems. a tool to analyze the system behavior. In *International Refrigeration Conference at Purdue*, 1998.

[3] J.M. Corberán et Al. Detailed modeling of evaporators and condensers. In *International Refrigeration Conference at Purdue*, 2000.

[4] Mark O. McLinden et al. *NIST Thermodynamics and Transport Properties of Refrigerants and Refrigerant Mixtures–REFPROP Version 6.0*, January 1998.

[5] More et al. *User guide for MINPACK-1*, 1980.

[6] A.F. Mills. *Heat and Mass Transfer*. Richard D. Irwin,Inc., 1995.

[7] Even Thorbergsen. *Frigosim Instruction Manual Version 2.0*, 1998.

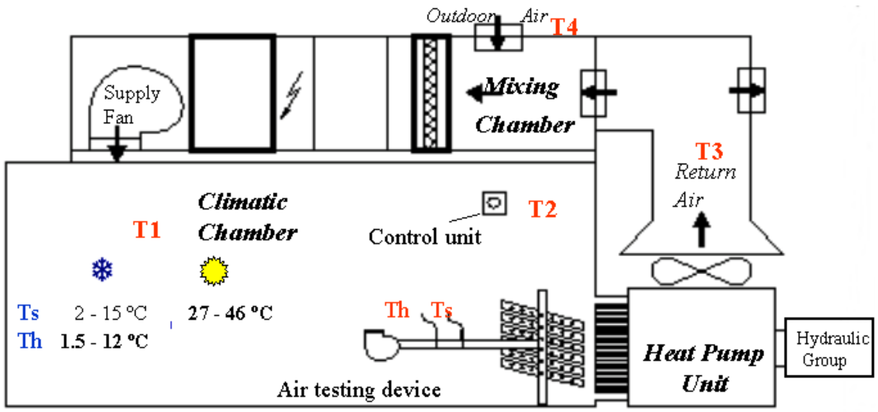
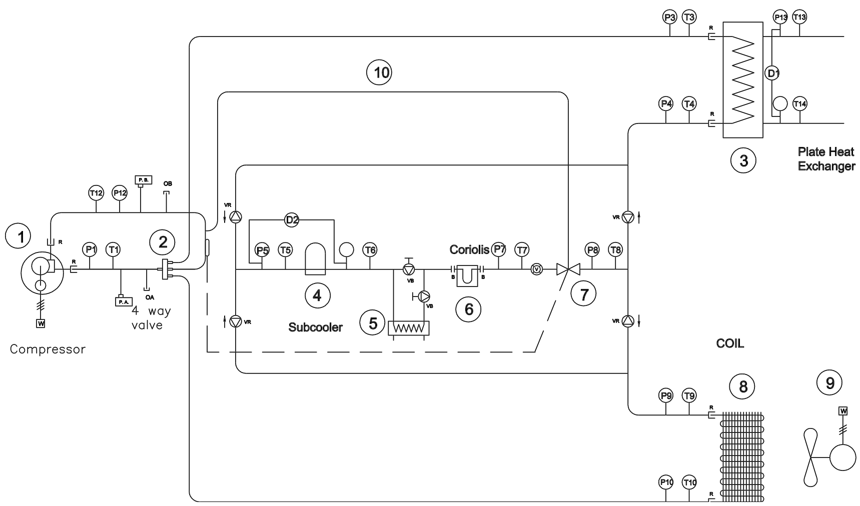


Figure 1: Heat Pump Test Rig



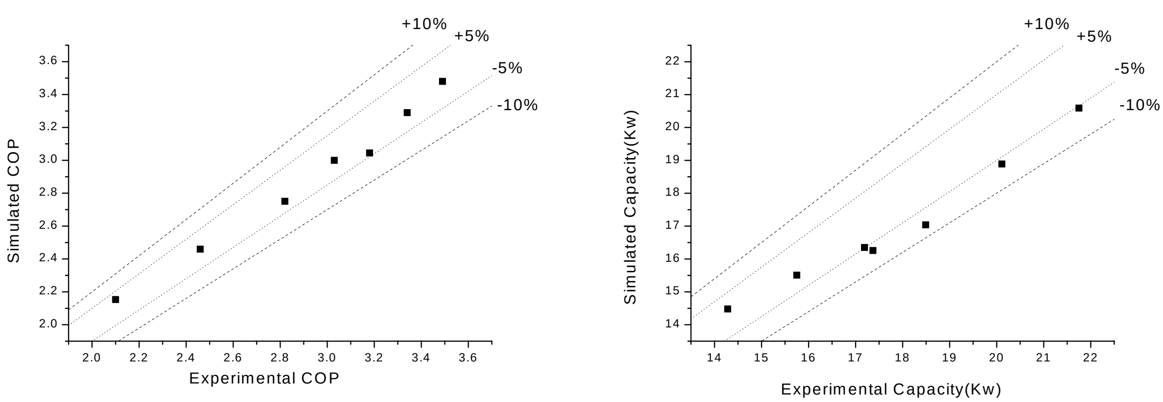
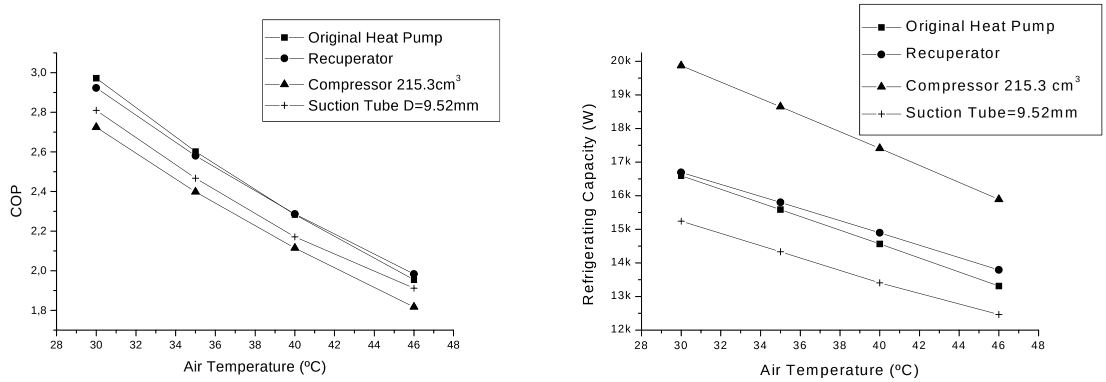
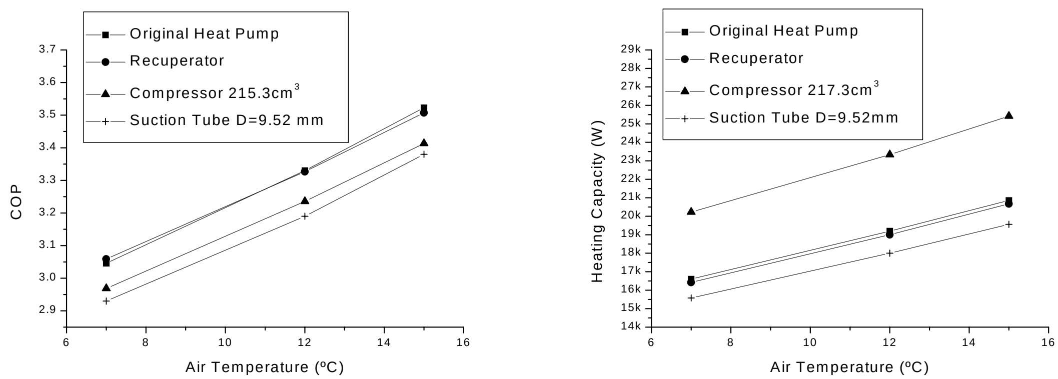


Figure 3: Comparison between model and experimental results

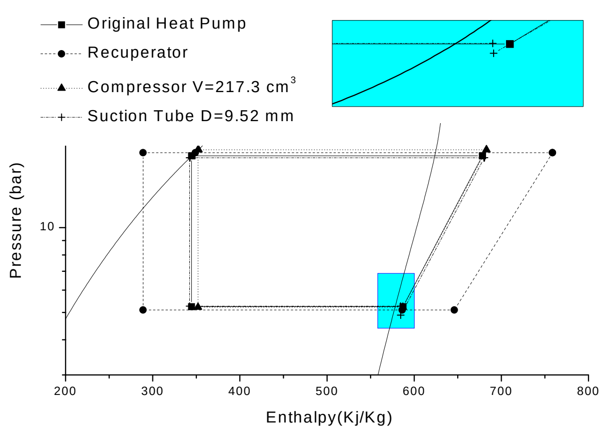


(a) Cooling mode

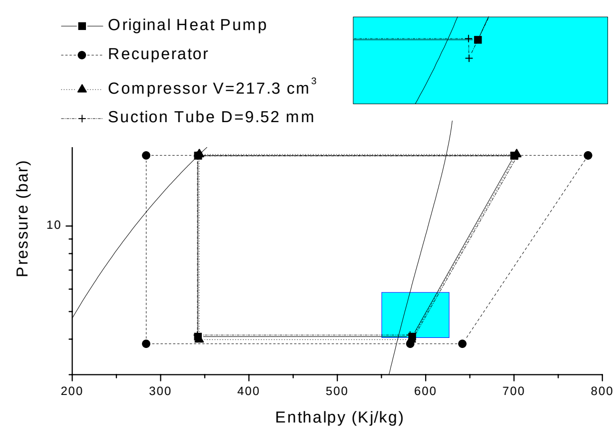


(b) Heating mode

Figure 4: Optimization studies



(a) Cooling mode



(b) Heating mode

Figure 5: Optimization studies