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Research Publications

INFLUENCE OF THE SOURCE AND SINK TEMPERATURES ON THE OPTIMAL REFRIGERANT CHARGE OF A WATER-TO-WATER HEAT PUMP

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ABSTRACT

This paper presents the results of a simulation study that was carried out in order to elucidate the influence of the source and sink temperatures on the optimal charge of a water-to-water heat pump. A water to water chiller previously employed in a charge optimisation study carried out by the authors, was modelled in detail with IMST-ART software and a full simulation study was performed, first as a chiller, keeping constant the outlet water temperature from the evaporator at 7 ºC replicating the experimental campaign, and then as a heat pump, keeping constant the hot water supply temperature at the outlet of the condenser, replicating the test campaign carried out with a water to water heat pump at KTH (Stockholm). The results of the simulations were in very good agreement with the test results obtained at UPV when the sink temperatures (condensation temperatures) were varied, and show the same trends found at KTH when the source temperature (evaporation temperature) was progressively decreased. The simulation allowed to study the refrigerant distribution among the different components of the system at different operating conditions and showed that the great variation of the optimal charge with the variation of the evaporation temperature is mainly due to the variation of the amount of refrigerant into the oil. Regarding the refrigerant distribution among the components, most of the charge was found in the condenser (50-80%), a considerable part in the evaporator (around 15%) and a substantial amount in the compressor oil (10-35%).

1. INTRODUCTION

The use of optimised minimum-charge units is an important issue in the development of future sustainable refrigeration and A/C equipment. The charge of the systems is directly related with the direct emissions of substances with high GWP in the case of using synthetic refrigerants, and of potential hazardous substances in the case of employing natural refrigerants as ammonia or hydrocarbons. On the other hand, minimisation of charge cannot be realised in detriment of the COP of the systems since lower efficiencies imply higher electricity consumption and therefore higher CO_2 global production. An integral optimisation of the equipment design and of the charge is therefore necessary to lead to the minimum CO_2 global emissions.

A good understanding and modelling of the distribution of the charge among the different components of a refrigeration unit and of the influence of the charge on the unit performance are essential to better design and charge future equipment. In a previous experimental study at the UPV (Corberan, Martinez, and Gonzalvez 2008), the authors found that for a water-to-water chiller the condenser subcooling corresponding to the optimal charge (leading to maximum COP) was almost constant regardless of the condensation temperature. The optimal charge at each condensation temperature was not exactly the same but the variation was small.

In contrast,(Fernando et al. 2004), at KTH, arrived to a different conclusion when testing a water-to-water heat pump under different evaporation temperatures. They found that the optimal charge decreased with the evaporation temperature. This paper presents the simulation study that was performed in order to elucidate the influence of the source and sink temperatures on the optimal charge of a water-to-water heat pump. The original water-to-water chiller employed in the previous study at UPV was modelled in detail with IMST-

ART software and a full simulation study was performed, first as a chiller, keeping constant the outlet water temperature from the evaporator at 7 ºC, replicating the experimental campaign carried out at UPV, and then as a heat pump, keeping constant the hot water supply temperature at the outlet of the condenser, replicating the test campaign carried out at KTH.

When the model results were compared with the experimental ones for both applications, chiller and heat pump, a good agreement was found and then the simulation results were employed to better analyse and understand the distribution of the charge among the different components of the unit and the influence of the source and sink temperatures on the optimal charge.

2. METHODOLOGY

2.1. Model description

The simulation study has been performed by means of IMST-ART software. A short description of the main characteristics of the model in what matters the present paper is given in this section. For a full description of its characteristics and capabilities visit: www.imst-art.com.

The global model of the system is divided in submodels: compressor, heat exchangers, expansion device, accessories, and piping. Each submodel involves a series of non-linear equations and in the case of the heat exchangers, the solution of a system of ODEs, which is discretized with a finite volume technique. Then, the submodels are coupled to form a global model of the heat pump. The global set of equations forms a complex system of non-linear equations AEs and DAEs, which is solved globally by a Newton-like solver.

The independent variables chosen for the global set of equations are pressure and enthalpy at each inlet and outlet point. This choice assures a smooth variation of the variables, not given by other choices like temperature or quality.

Calculation of the refrigerant thermodynamic and transport properties is performed by REFPROP subroutines from (NIST) for each refrigerant. The corresponding properties are then conveniently stored in a refrigerant data library. The required properties during the simulations are estimated by interpolation from the corresponding data file. Additionally, built-in tables allow the calculation of the properties of any usual secondary fluid, i.e. dry air, humid air, water and common brines.

The global system of equations is solved using a standard solver based on the MINPACK subroutine HYBRD1, which uses a modification of M.J.D. Powell's hybrid algorithm. The program continuously surveys the convergence of the method and a special strategy to find appropriate initial solutions and also to carefully bound variables and functions have been worked out. The result of all this is an extremely robust software.

2.1.1. Compressor

For the compressor, three equations are required to characterize its behaviour: one for the mass flow rate, eq. (1); one for the power input, eq. (2); and one for the outlet enthalpy, eq. (3). In IMST-ART these equations are normally expressed as a function of the volumetric efficiency η_v, the compressor efficiency η_c and the fraction of power input ξ which is lost to the environment from the outer shell of the compressor.

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

$$E = \frac{\dot{m}(i_{is} - i_i)}{\eta_c} \tag{2}$$

$$i_o = i_i + \frac{i_{is} - i_i}{\eta_c}(1 - \xi) \tag{3}$$

For this simulation study the ARI performance polynomials provided by the compressor manufacturer were employed to estimate the compressor mass flow rate and the power input instead of equations (1) and (2) and a fraction of 5% was assumed for ξ.

Of special importance for this study is the estimation of the amount of refrigerant dissolved in the lubricant oil. The software includes built in correlations with the refrigerant into oil solubility for some typical combination of refrigerant and oil, which allow the estimation of the amount of refrigerant dissolved in the oils as a function of the pressure at the compressor suction and of the oil temperature. Figure 7 shows the solubility curves of propane in mineral oil, which have been extracted from (ASHRAE 2006). For this purpose, the compressor model also includes an empirical correlation of the oil temperature as a function of the condensing temperature and the compressor discharge temperature, see (Navarro et al. 2005).

2.1.2. Heat Exchangers

In the case of the evaporator or condenser, a 2-phase flow with phase change occurs. A steady 2-phase flow similar to the annular pattern in tubes is considered to occur along the channels in between plates in IMST-ART BPHE model. The separated two-fluid 2P flow model is assumed. The mass flow rate and the mass velocity all along the refrigerant path are considered to be known at each iteration of the HE but unknown for the global system of equations.

At a fluid cell, the number of equations to be considered is two (energy and momentum). The unknown variables at the outlet of the cell are therefore: x, α, ρ_l, ρ_v, i_l, i_v, p and T. The assumption of thermodynamic equilibrium is adopted, so that the state equations and a model for the void fraction allow closing the system of equations.

The numerical method employed for the heat exchangers solution is called SEWTLE (for Semi Explicit method for Wall Temperature Linked Equations). Basically, this method is based on an iterative solution procedure. First a guess is made about the wall temperature distribution, then the governing equations for the fluid flows are solved in an explicit manner, getting the outlet conditions at any fluid cell, from the values at the inlet of the HE and the assumed values of the wall temperature field. Once the solution of the fluid properties are got at any fluid cell, then the wall temperature at every wall cell is estimated from the balance of the heat transferred across it. This procedure is repeated until convergence is reached. The numerical scheme developed for the calculation of the temperature at every wall cell is also explicit, so that the global strategy consists in an iterative series of explicit calculation steps. For further description, see (Corberan et al. 2001). Dedicated empirically adjusted correlations are employed for the evaporation and condensation heat transfer coefficients in the BPHEs (Garcia-Cascales et al. 2007).

2.1.3. Expansion device

The expansion device employed in the experimental tested unit was a thermostatic expansion valve so it involves two governing equations, the first one is the assumption of isoenthalpic flow through the valve and the second one is the statement of the imposed superheat.

2.1.4. Charge inventory

The water to water unit to be simulated has no liquid receiver. In that kind of systems the subcooling at the outlet of the condenser depends on the amount of refrigerant that must be allocated in the condenser in liquid form and therefore becomes subcooled. The closing equation of the global system of equations is then the charge inventory, i.e. the addition of the refrigerant mass at every portion of the refrigerant circuit must be equal to the refrigerant charge of the system. IMST-ART is able to calculate under this mode, i.e. the refrigerant charge is an input and the subcooling is an output.

2.2. Experimental validation of the model

As explained in the introduction section, the research presented in this paper is a continuation of a previous experimental study in which we studied the influence of the charge on the performance of 16 kW chiller working with propane under different condensation temperatures, for more details see (Corberan, Martinez, and Gonzalvez 2008). The main components of the experimental unit are summarised in table 1.

Table 1. Components of the experimental heat pump used for the experimental validation

Compressor	Evaporator	Condenser	Expansion device	Oil type
Scroll ZR72KCE	BPHE V80x26	BPHE B80x26	Thermostatic bidirectional	1.5 litres MO-ISO68

The geometry and characteristics of the different components were input to the model. Then, the heat transfer coefficient at the evaporator and condenser where tuned to reproduce the evaporation and condensation temperature at the nominal operating conditions (12-7, 30-35), corresponding to a ground coupled chiller, and optimal charge. The tuning required a degradation factor of 0.85 at both, condenser and evaporator. Those factors were then kept constant along the complete study. The following figures show the comparison between the calculated and measured results. Figure 1 shows the deviation between the measured and simulated performance of the experimental unit in a number of points with different working conditions and two different charges. As can be observed the unit performance is predicted with high accuracy all over the range of variation of the experiments.

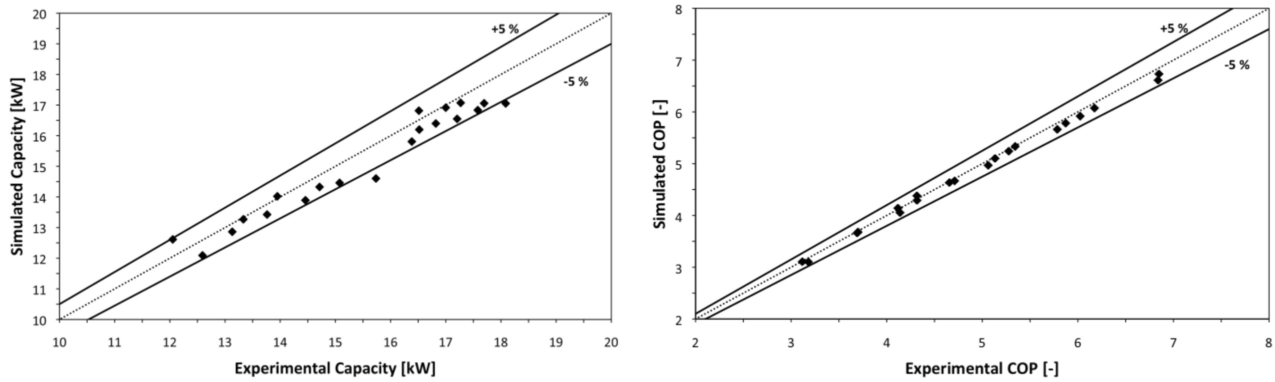

Figure 1. Experimental vs. calculated cooling capacity and COP with a refrigerant charge of 460g and 540g.

3. INFLUENCE OF THE SINK TEMPERATURE

Figure 2 shows the comparison between the experimental and predicted capacity and COP at the evaporator for different refrigerant charges at different operating conditions. The evaporator was always operated with constant inlet and outlet water temperatures corresponding to a chiller: 12 - 7 ºC.

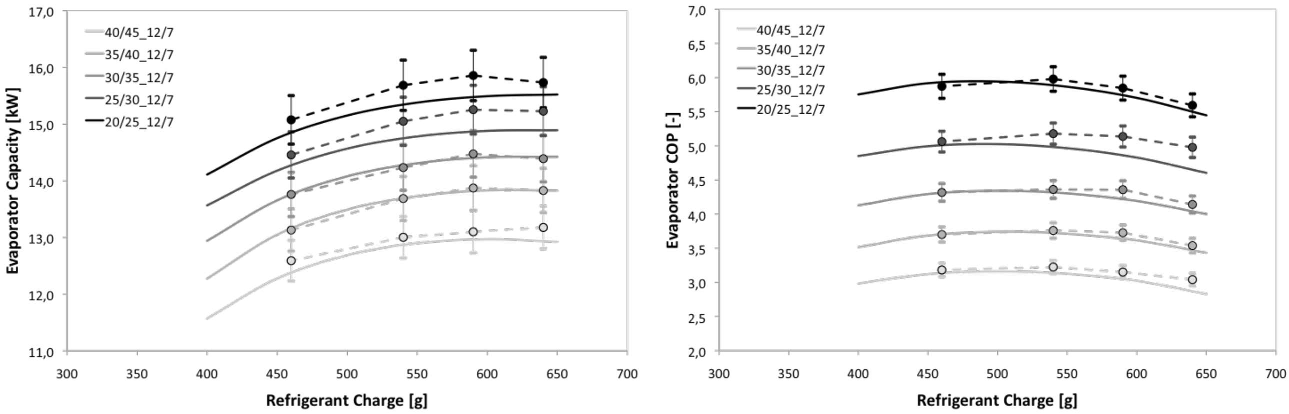

Figure 2. Experimental and calculated cooling capacity and COP at different working conditions for the chiller application

As can be seen, the model results (continuous lines) show a close similarity with the experimental results. (dotted lines). The decrease of COP due to the increase of the water temperature at the condenser is very well predicted. Also the shape of the curves is very well predicted proving that the main mechanisms happening in the condenser are well captured. The experimental optimal charge is around 550g while the calculated one is lower (around 480). This difference is still small if one takes into account the large number of assumptions that the model involves, mainly those concerning the heat transfer calculations inside the evaporator and condenser and the void fraction evaluation all along the different components. It is interesting to note that the optimal charge is almost constant and independent on the condensation temperature.

The experimental results in figure 2 show the associated uncertainty bars due to the employed instrumentation. In regard to the uncertainty on the refrigerant charge, this could be high but it is very difficult to estimate. The charging of the refrigerant is a critical issue when the amount of charge is relatively small, as it is the case, especially due to leakages and refrigerant stored in the fittings during the charging process. The measurements were always carried out after several hours of continuous operation under the same water temperature conditions at evaporator and condenser in order to let the refrigerant being dissolved in the oil to reach a reasonable steady state concentration.

4. INFLUENCE OF THE SOURCE TEMPERATURE

In order to have an experimental basis for comparison for this part of the study (heat pump application with hot water production at 40ºC), the simulations were performed by selecting the different heat source temperatures, with constant heat sink temperature, that were employed by (Fernando et al. 2004) at KTH. Table 2 shows the simulation test matrix.

Table 2. Simulation test matrix

Indoor side		Outdoor side	
Water inlet Tª [ºC]	Water outlet Tª [ºC]	Brine inlet Tª [ºC]	Brine outlet Tª [ºC]
33	40	-10	-13
		-2	-5
		6	2.25
		12	7.6

The employed model was exactly the same, except that the secondary fluid at the outdoor side employed in this case was a water/ethylene glycol mixture as the one employed in the experiments at KTH. The refrigerant was again propane and the superheat at the compressor inlet was set to 7K as in the experiments.

Figure 3 shows the calculated heating capacity and COP of the unit, working now as a heat pump, as a function of the refrigerant charge at different evaporator water temperatures.

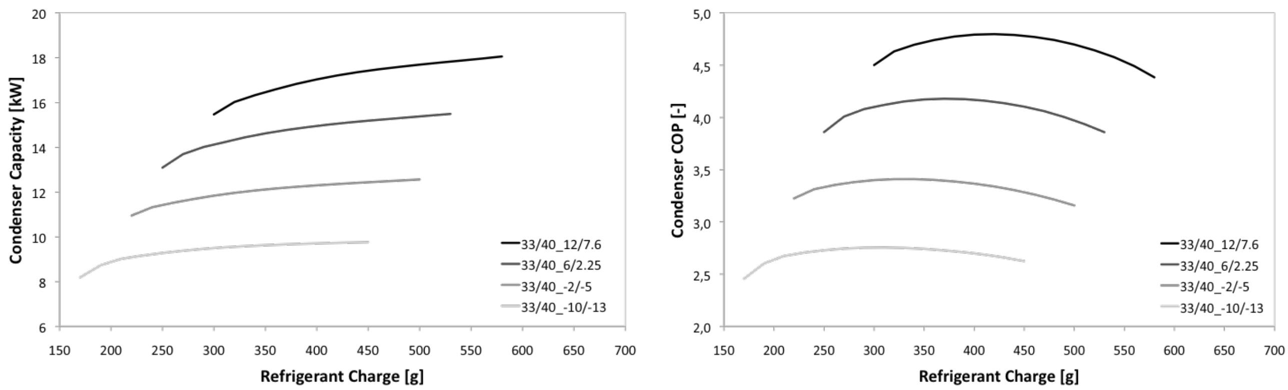

Figure 3. Condenser capacity and COP as a function of the refrigerant charge for different outlet brine temperatures.

As observed, for a given heat source temperature, the heating capacity increases with the charge, first quite importantly and then more gradually. The COP presents a clear maximum, which clearly depends on the source temperature, in contrast with the case of variation of the sink temperature previously studied. On the other hand, the maximum becomes less pronounced at the lowest temperatures. These results agree qualitatively well with the results obtained by (Fernando et al. 2004) showing very similar behaviour. The optimal refrigerant charges leading to the maximum COP for the different tested source temperatures are 310g, 340g, 370g, and 420g respectively.

Figure 4 shows the cooling COP for the unit working as a chiller (left) and the heating COP (right) for the unit working as a heat pump, vs. the condenser subcooling. As it was found in (Martínez Galván 2007) and reported in (Corberan, Martinez, and Gonzalvez 2008), the optimum COP is reached when the subcooling is around 5-7 K regardless the sink temperature (left graph). Figure 4 (right) proves now that the optimum subcooling is again around the same value regardless the source temperature. As discussed in Corberan *et al.* (2008) the optimum subcooling approximately corresponds to the subcooling that the condenser is able to reach, without significant variation of the condensation temperature, which is a figure a bit lower than the temperature difference between the condensation temperature and the inlet water temperature (source temperature). This value would depend on the condenser design but no great differences are expected among typical BPHEs designs. In fact, the reason why the optimal subcooling slightly moves to a lower value when the evaporator temperature decreases is because this decrease is related with a significant decrease on the refrigerant mass flow and therefore the temperature difference between the condensation temperature and the inlet water temperature slightly decreases.

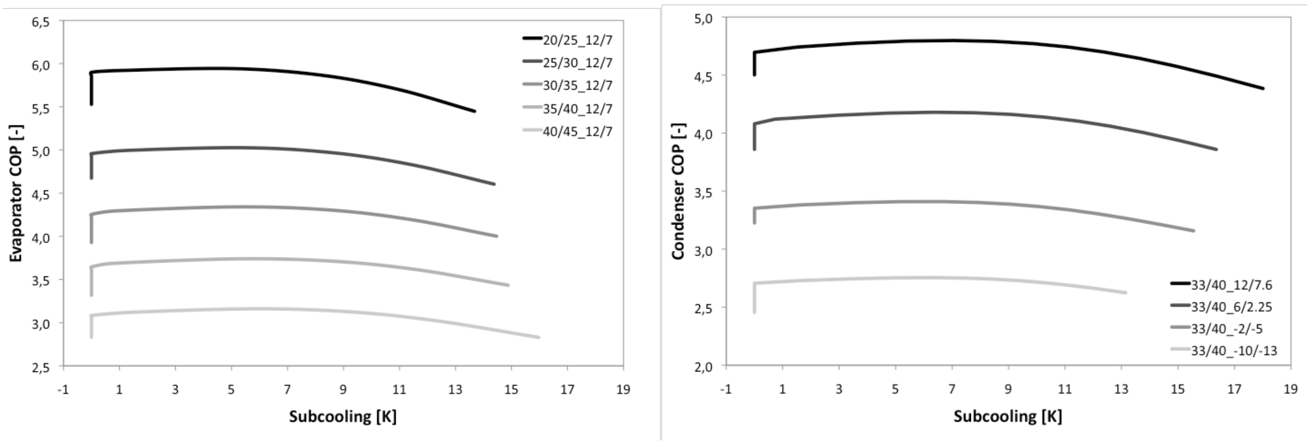

Figure 4. Evaporator COP working as a chiller (left), and Condenser COP working as heat pump (right), as a function of subcooling under different working conditions

The decrease in COP at 0K subcooling is due to the lack of refrigerant in the system, leading to a two-phase flow inlet to the expansion device. The two-phase flow is represented also as 0 subcooling.

As a conclusion, the optimal charge of a water to water chiller can be easily found by setting the subcooling to approximately the optimal one 5-7 K regardless the sink temperature could be. This is also recommended for a heat pump, but given that the optimum charge is not constant with the source temperature, the optimum subcooling should be set at the most frequent operating source temperature. Normally this corresponds to quite mild temperatures. At the lowest source temperatures the unit will be working above the optimum charge, nevertheless the lower the source temperature the more horizontal is the COP curve for high charges so that the total penalty for not being at the optimal charge is low. This result is also of practical interest since the optimization of the unit through the evaluation of the subcooling is much easier than through the exact measurement of the charge.

5. CHARGE DISTRIBUTION RESULTS

The model allows the estimation of the distribution of the refrigerant charge in the main components of the system. Figure 6 shows the estimated refrigerant charge distribution among the main components of the system for both studied applications: chiller (left) and heat pump (right), i.e. charge fraction in the component vs. the total charge, for the different studied working conditions.

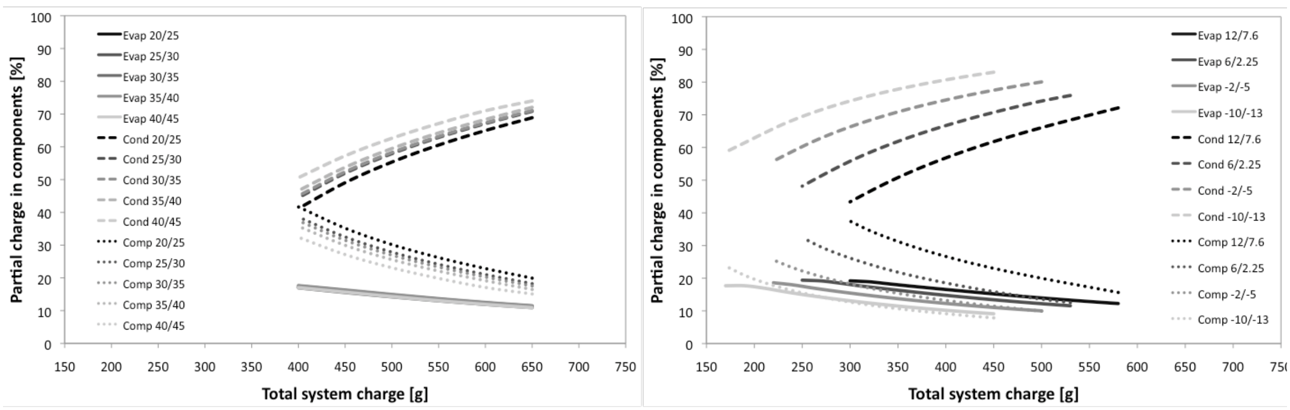

Figure 6. Charge distribution in the components of the system. Left: chiller case. Right: heat pump case

As observed in the figure, the most of the refrigerant charge is in the condenser (50-80%) increasing with the system charge. The extra charge of refrigerant which is injected into the unit moves into the condenser and stays in it as subcooled liquid refrigerant. The subcooling region in the condenser then increases and the condensation temperature must increase so that the condensation can take place in a smaller region. The increase in condensation temperature is always detrimental for the COP of the cycle. On the other hand the increase of the subcooling increases the COP. These two opposite trends lead to the existence of the observed COP maximum.

At the heat pump case, the lower the heat source temperature the lower is the amount of the refrigerant in the evaporator and in the compressor, consistently, the greater the charge in the condenser. The amount of refrigerant in the evaporator is quite the same for all the tests (12-20%), independently of the source or sink temperatures. At the same charge, the lower the source temperature the lower the density of the refrigerant in the evaporation and the higher the inlet quality so that the lower the amount of refrigerant in the evaporator as seen in figure 6.

Figure 7 shows the propane solubility in mineral oil for the simulated heat pump conditions (left) as a function of the refrigerant charge in the simulated system, and the general propane solubility curves in mineral oil (right) as a function of the oil temperature and pressure (ASHRAE 2006).

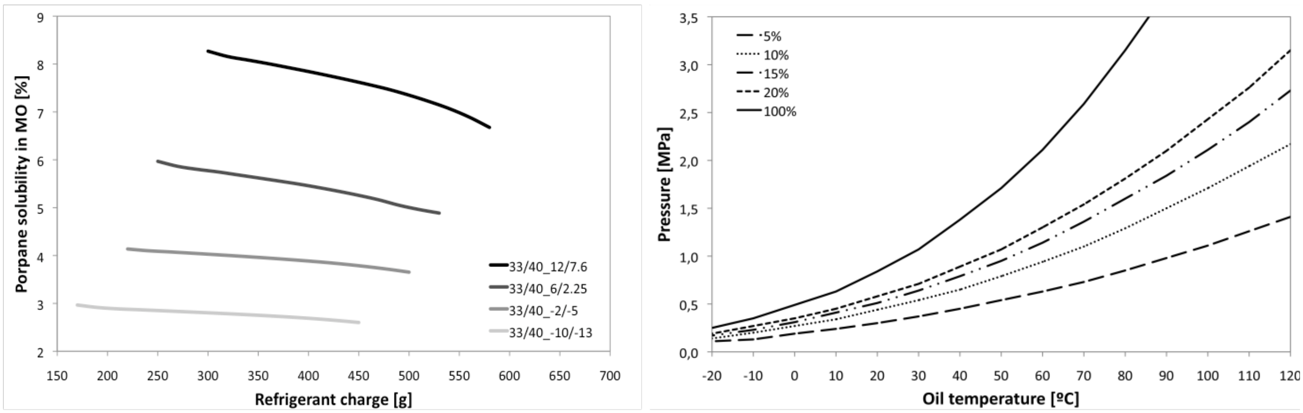

Figure 7. Propane solubility in mineral oil iso-68, as a function of the refrigerant charge for the simulation results (left), and as a function of the oil temperature (right).

Therefore, the amount of refrigerant dissolved in the compressor oil is important (10-35%), and responds to the solubility of the refrigerant into the oil; the lower the source temperature, the lower the suction pressure and therefore, the solubility of propane into the oil decreases. At the same time, the lower the suction temperature, the higher the discharge temperature so that the discharge temperature increases producing an oil temperature increase and hence again a decrease of the solubility of propane into the oil; both mechanisms leading to a decrease of the amount of propane into the oil. At a constant source temperature, the decrease of the fraction of refrigerant charge in the compressor oil when the charge is increased is also due to the increase in the oil temperature due to the increase in condensation temperature. Conversely, at high source temperatures and low charges the amount of refrigerant in the oil can be very high (35% in the figure for 12/7.6 case).

CONCLUSIONS

The main conclusions that can be drawn from the presented study are the following:

- The model accurately predicts the performance of the unit under different working conditions and is able to reproduce the influence of the charge on the performance under different sink and source temperatures, for both applications: chiller or heat pump.
- Basically, the extra charge of refrigerant injected in a unit moves into the condenser and stays in it as subcooled liquid refrigerant. The subcooling region in the condenser then must enlarge and therefore the condensation temperature must increase so that the condensation can take place in a smaller region. The increase in condensation temperature is always detrimental for the COP of the cycle. On the other hand

the increase of the subcooling increases the COP. These two opposite trends lead to the existence of a maximum COP.

- The optimal charge for a chiller is almost constant independently of the sink temperature.
- The optimal charge for a heat pump depends on the source temperature. The lower the source temperature the lower the optimal charge. However, the COP decrease with extra charges becomes small.
- The optimum COP is reached when the subcooling is around 5-7 K regardless the sink temperature or the source temperature. This optimum approximately corresponds to the subcooling that the condenser is able to reach, without significant variation of the condensation temperature, which is a figure a bit lower than the temperature difference between the condensation temperature and the inlet water temperature (source temperature). This value would depend on the condenser design but no great differences are expected among typical BPHEs designs.
- Therefore, the optimal charge of a water to water chiller or heat pump can be easily found by setting the subcooling to approximately the optimal one 5-7 K at the most frequent source and sink temperatures of the targeted application.
- The fraction of refrigeration charge in each component of the system has also been investigated with the model. The results show that in this kind of equipment which does not include a liquid receiver the most of the refrigerant charge is in the condenser (50-80%), the fraction increasing with the total charge. Furthermore, the fraction of refrigerant in the compressor is very important (10-35%), and plays an important role in the behaviour of the unit when the source or sink temperatures are modified.

ACKNOWLEDGMENTS

This research has been partially funded by the Spanish Ministry for Science and Innovation (MICINN) through the project “Estudio de optimización de equipos frigoríficos y de aire acondicionado para su funcionamiento con refrigerantes naturales hidrocarburos” with reference DPI2008-06707-C02-01. The authors gratefully acknowledge their financial support.

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