SIMULATION-BASED DESIGN AND OPTIMIZATION OF A RESIDENTIAL AIR CONDITIONER RUNNING WITH R-410A

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Abstract. *A simulation-based optimization methodology to size the heat exchangers (i.e., condenser and evaporator) of a unitary residential air conditioner with focus on both performance enhancement and cost savings is presented. A steady-state system simulation model was put forward for a 2.5-ton nominal cooling capacity split air conditioning unit operating with R410A as working fluid. The model predictions for cooling capacity, sensible heat ratio (SHR) and coefficient of performance (COP) were compared with experimental data, when it was found that the model is able to predict the experimental trends within a ±5% error band. In addition, the model was used to find out the condenser and evaporator geometries (face area, heat transfer area) that enhance the system COP for a fixed cost, and vice-versa. On one hand, it was observed that the original COP can be increased by 5% if the cost is held fixed. On the other hand, cost savings of 5% were achieved in cases when the system COP was remained unchanged.*

Keywords: *air conditioning, modeling, simulation, optimization, energy performance, cost*

1. INTRODUCTION

Albeit the HCFC working fluids have already been phased out in most of developed countries (Calm and Domanski, 2004), R-22 replacement for static air conditioning applications is still an up-to-date concern in emerging countries like Brazil as the phase out deadline is approaching. Among the several replacement candidates under consideration, the most prospective ones seem to be the binary quasi-azeotropic mixture named R-410A and propane (R-290) (Mohanraj et al., 2009). Whereas the former is seen as a drop-in solution for most small-capacity air conditioning units – although its working pressures are much higher than those found when R-22 is used – the latter will require a new system design focused on charge minimization because of its high flammability. As learned from the CFC phase out in the late 1980s (Calm and Didion, 1998), a series of component matching and performance enhancement studies will be required before new air conditioning systems running with either R-410A or R-290 come onto the Brazilian market.

In general, the component matching exercise is carried out by testing a prototype according to a standardized test procedure (e.g., ARI 210/240 Standard, 2006). The experimental tests are costly and time demanding not only by themselves, but also because of the cost and time associated with prototype assembling and transporting. It has been advocated in the open literature that the development costs of vapor compression refrigeration systems may be reduced if proper simulation tools are adopted (Hermes et al., 2009; Waltrich et al., 2010) as hundreds of design options can be evaluated within a couple of minutes without the need of a large number of prototypes.

Both steady-state (Domanski and Didion, 1983; Domasnki and McLinden, 1992; Rossi and Braun, 1995) and transient (Chi and Didion, 1983; Murphy and Goldshmidt, 1985; Sami and Duong, 1987; Vargas and Parise, 1995) simulation models have been proposed in the past decades for predicting the performance of air conditioning and heat pump systems. In spite of the large number of publications in the field, optimization studies are scarce, and the few available are focused on a single component (Pira et al., 2000; Stewart et al., 2005; Domanski and Yashar, 2007). Recently, Waltrich et al. (2010) presented a model-based methodology to optimize the energy and cost of a refrigeration cassette for light commercial applications by varying not only the heat exchangers (condenser and evaporator) characteristics but also the compressor capacity. Later, Negrão and Hermes (2011) carried out an energy and cost optimization of a household freezer taking into account not only the refrigeration system characteristics (heat exchangers, compressor), but also the compartment insulation. So far, there is no evidence in the open literature of a multi-component, multi-objective optimization of an air conditioner.

Therefore, the present study is focused on the optimization of a residential air conditioning unit by properly sizing and selecting the air-supplied heat exchangers based on both COP and cost. Bearing in mind that the best simulation model is the one which provides the desired results with a minimum computational effort, it was decided to overlook some minor effects (i.e. two-phase heat transfer coefficient, refrigerant-side pressure drop, coil circuitry) and place all modeling efforts on factors that actually play important roles on the system performance, such as heat exchanger geometry (e.g., number of fins and tubes), air-side pressure drop and fan pumping power. In a future publication, the model proposed herein will be used not only to assess but also to enhance the system performance in case R-410A and R-290 are adopted as R-22 replacements.

2. SIMULATION AND OPTIMIZATION MODELS

2.1 Refrigeration Loop Modeling

In general, the system simulation model follows the approach of Hermes et al. (2009) originally developed for household refrigerators and freezers. The refrigeration system under study is a unitary (split) residential air conditioner comprised of a hermetic scroll compressor, two fan-supplied tube-fin heat exchangers (a condenser and an evaporator) and a thermostatic expansion device, as depicted in Fig. 1. A 4-way valve is used to turn the system into a heat pump in case the heating mode is desired. In this work, the system was modeled and tested in the cooling mode.

The model was devised assuming that, on one hand, there is no need to account for the refrigerated room as did by Hermes et al. (2009) since the indoor and outdoor temperatures are standardized conditions (e.g., ARI 210/240 Standard, 2006). On the other hand, the latent heat transfer in the evaporator coil should be properly calculated.

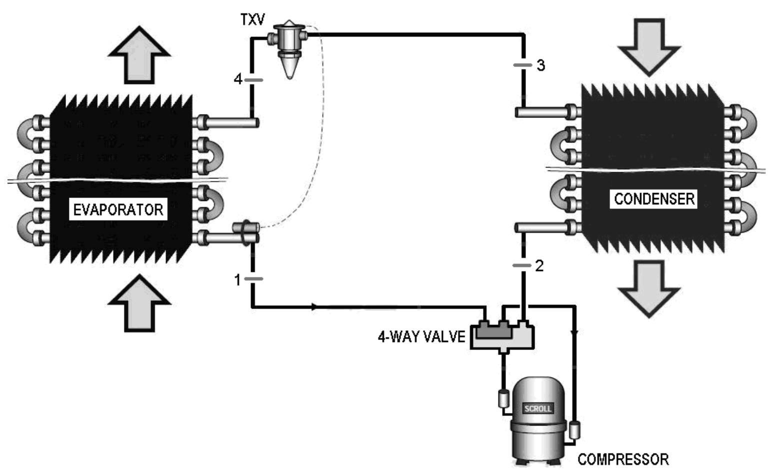

Figure 1. Schematic representation of the refrigeration system (Adapted from Kim et al., 2006).

The compressor was modeled based on mass and energy balances following a lumped approach. Therefore, the refrigerant enthalpy at the compressor discharge is calculated from

$$h_2 = h_1 + W_k / m_r \quad (1)$$

where m_r and W_k are the refrigerant mass flow rate and the compression power, respectively. They both are calculated as bi-quadratic functions of the condensing (t_c) and evaporating (t_e) temperatures from the following empirical polynomials fitted to experimental data obtained from the manufacturer's compressor maps:

$$m_r = \beta\left(a_0 + a_1 t_e + a_2 t_c + a_3 t_e t_c + a_4 t_e^2 + a_5 t_c^2 + a_6 t_e^2 t_c + a_7 t_c^2 t_e + a_8 t_e^2 t_c^2\right) \quad (2)$$

$$W_k = \beta\left(b_0 + b_1 t_e + b_2 t_c + b_3 t_e t_c + b_4 t_e^2 + b_5 t_c^2 + b_6 t_e^2 t_c + b_7 t_c^2 t_e + b_8 t_e^2 t_c^2\right) \quad (3)$$

where β is a scaling factor used during the optimization task (β=1 for the original compressor), whereas the coefficients a_i and b_i were adjusted using experimental data obtained from the compressor maps.

Bearing in mind that the best heat exchanger simulation model is the one which provides the desired results with a minimum computational effort, some minor effects were overlooked and, thus, the heat exchanger modeling efforts were placed on factors that actually play important roles on the system performance. Therefore, the following simplifying assumptions were taken into account: (i) the heat exchangers were modeled following a lumped approach and, therefore, no distinction was made between the single and two-phase refrigerant flow regions; (ii) the air side pressure drops in the condenser and evaporator do not affect the air flow rate supplied by the fans, but the required pumping power, and (iii) the refrigerant side pressure drops on both condenser and evaporator coils were not taken into account.

Therefore, the refrigerant enthalpy at the condenser (3) and evaporator (1) outlet ports were calculated respectively from the following energy balances,

$$h_3 = h_2 - Q_c / m_r \quad (4)$$

$$h_1 = h_4 + Q_e / m_r \quad (5)$$

The heat transfer rate released by the condenser, Q_c, was calculated from

$$Q_c = \varepsilon_c \rho_a c_{p,a} V_c (t_c - t_o) \tag{6}$$

where V_c is the air flow rate supplied by the condenser fan [m^3/s], $c_{p,a}$ and ρ_a are the specific heat [J/kg$_a$K] and density [kg$_a$/m^3] of dry air, respectively, t_o is the temperature of the outdoor environment [°C] (standardized condition), t_c is the condensing temperature and ε_c is the condenser heat exchanger effectiveness calculated from

$$\varepsilon_c = 1 - \exp\left(-\eta_c \hbar_c A_c / \rho_a c_{p,a} V_c\right) \tag{7}$$

where $\hbar_c$ is the air side heat transfer coefficient [W/m^{2}K] calculated from the correlation of Wang et al. (2000), A_c is the condenser heat transfer area ($=A_{f,c}+A_{t,c}$) [m^2], and η_c is the condenser surface efficiency calculated from the method of Schimidt (1945). The overall heat transfer rate absorbed by the evaporator, Q_e, was calculated from

$$Q_e = Q_{e,sen} + Q_{e,lat} \tag{8}$$

where the indexes *sen* and *lat* stand for sensible and latent heat transfer, respectively. The overall heat transfer rate was calculated based on the enthalpy potential as follows

$$Q_e = \varepsilon_e \rho_m V_e (h_i - h_e) \tag{9}$$

where V_e is the air flow rate supplied by the evaporator fan [m^3/s], $c_{p,m}$ and ρ_m are the specific heat [J/kg$_a$K] and density [kg$_a$/m^3] of moist air, respectively, $h_i=h(t_i,\ \phi_i,\ p_{atm})$ is the air enthalpy of the indoor environment [J/kg$_a$] which is a function of the air temperature, t_i, the relative humidity of the indoor environment, ϕ_i, and the local atmospheric pressure, p_{atm}. Furthermore, h_e is the enthalpy of the saturated air at the coil temperature, t_e, [J/kg$_a$], and ε_e is the evaporator heat exchanger effectiveness calculated from

$$\varepsilon_e = 1 - \exp\left(-\eta_e \hbar_e A_e / \rho_m c_{p,m} V_e Le^{2/3}\right) \tag{10}$$

where $\hbar_e$ is the air-side heat transfer coefficient [W/m^{2}K] calculate from the correlation of Wang et al. (2000), A_e is the condenser heat transfer area ($=A_{f,e}+A_{t,e}$) [m^2], η_e is the evaporator surface efficiency (Schimidt, 1945), and Le is the Lewis number assumed to be equal to the unity. The latent heat transfer rate was calculated from

$$Q_{e,lat} = \eta_{wet} \varepsilon_e \rho_m V_e (\omega_i - \omega_e) h_{lv} \tag{11}$$

where ω_i and ω_e are the humidity ratios at the evaporator inlet (standardized condition) and at the coil surface (saturated air), respectively, h_{lv} is the latent heat of condensation [J/kg$_a$], and η_{wet} is the wet surface efficiency obtained from the empirical data. The sensible heat transfer rate is then obtained from Eq. (8) and the sensible heat ratio is calculated from

$$SHR = Q_{e,sen} / Q_e \tag{12}$$

The refrigerant flow through the expansion device was modeled as an isenthalpic process, $h_3=h_4$, and the mass flow rate model was replaced by a prescribed evaporator superheating degree, Δt_{sh}, so that the evaporating pressure was calculated from

$$p_e = p_{sat}(t_1 - \Delta t_{sh}) \tag{13}$$

The same procedure was applied to the refrigerant charge model, which was replaced by a prescribed condenser subcooling degree, Δt_{sc}, so that the condensing pressure was calculated from

$$p_c = p_{sat}(t_3 + \Delta t_{sc}) \tag{14}$$

The overall coefficient of performance, *COP*, is calculated as follows:

$$COP = (Q_e - W_{fan,e}) / (W_k + W_{fan,c} + W_{fan,e}) \tag{15}$$

where $W_{fan,e}$ and $W_{fan,c}$ are the power consumed by the evaporator and condenser fans, calculated from

$$W_{fan,x} = \Delta p_x V_{fan,x} / \eta_{fan,x} \tag{16}$$

where the index x stands for either the condenser (x=c) or the evaporator (x=e), $\eta_{fan,x}$ is the fan efficiency obtained from the empirical data, $V_{fan,x}$ is the air flow rate, and Δp_x is the pressure drop in the heat exchanger coil calculated from,

$$\Delta p_x = f_x \frac{G_x^2}{2\rho} \left(\frac{A_f + A_t}{A_{face}} \right)_x \tag{17}$$

where f_x is the friction factor obtained from the correlation of Wang et al. (2000), and $G_x = \rho V_{fan,x} / A_{min,x}$. The model is fed with indoor and outdoor air conditions, the working fluid properties, the condenser and evaporator geometries (fin pitch, number of fins and number of longitudinal and transversal tubes), air flow rates and fan efficiencies. The model was implemented using the EES platform (Klein and Alvarado, 2004) to solve the equation set simultaneously through the Newton-Raphson method. All required thermodynamic and thermophysical properties were calculated using the REFPROP7 software (Lemmon et al., 2002).

2.2 Optimization Scheme

The optimization task is aimed at finding out the condenser and evaporator heat transfer areas that either maximizes the system *COP* for a fixed cost or minimizes the cost for a target *COP*. Whereas the *COP* is calculated from Eq. (15), the heat exchangers costs are based upon the price of the commodity (the tubes are made of copper and fins of aluminum), as follows:

$$C = c_{Al} M_{Al} + c_{Cu} M_{Cu} \tag{18}$$

where M is material mass, c_{Al} and c_{Cu} are the unitary costs of aluminum (=0.4 $\$/kg_{Al}$) and copper (=1.8 $\$/kg_{Cu}$).

3. DATA REDUCTION AND MODEL VALIDATION

In this study, a R-410A residential heat pump equipment operating in the cooling mode (2.5-ton nominal cooling capacity) and comprised of a scroll compressor and a thermostatic expansion valve was selected to explore the proposed simulation-based design methodology. The heat pump unit was tested by Kim et al. (2006) in order to investigate the effect of artificially imposed faults on the system performance (Kim et al., 2006; Kim et al., 2009). In the present study, 14 no-fault data points available from Kim et al. (2006) were used for the model calibration and validation exercises.

Equations (2) and (3) were fitted to experimental data obtained from the compressor map. The resulting coefficients are depicted in Table 1. Figure 2 shows a comparison between the predictions of equations (2) and (3) and the experimental counterparts, where it can be noted that discrepancies within 3% and 7% error bands were achieved for the power consumption (Fig. 2.a) and mass flow rate (Fig. 2.b), respectively.

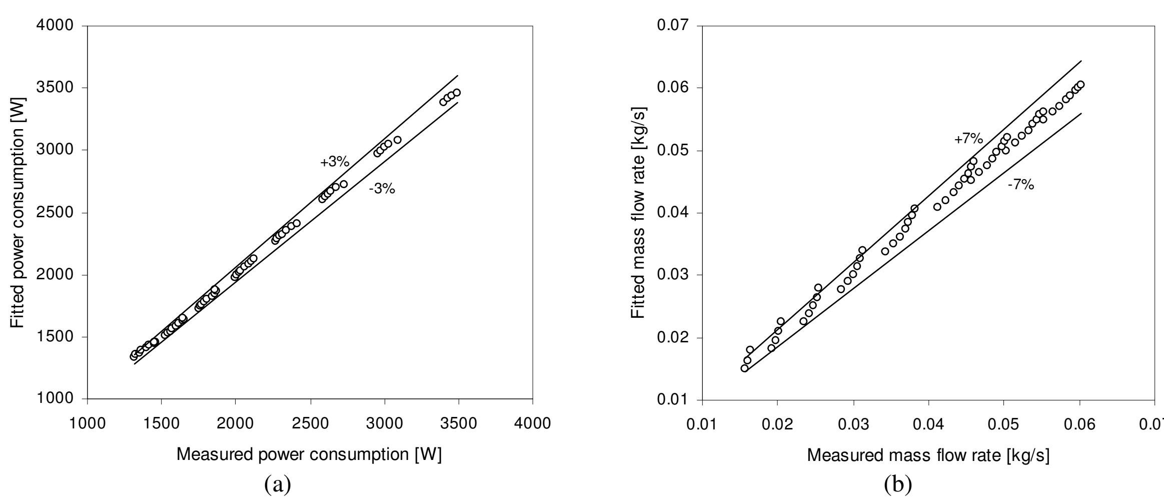

Figure 2. Compressor model validation: (a) mass flow rate; (b) compression power

Table 1. Coefficients of Eq. (2) and Eq. (3)

a_0	a_1	a_2	a_3	a_4	a_5	a_6	a_7	a_8
0.0467	0.00101	-0.00016	1.39 10^{-5}	9.34 10^{-6}	-4.8 10^{-7}	7.12 10^{-8}	-1.2 10^{-7}	4.45 10^{-10}
b_0	b_1	b_2	b_3	b_4	b_5	b_6	b_7	b_8
1160	-5.02	-8.5	0.0464	-0.444	0.673	0.0178	-0.0011	-0.00021

The evaporator wet surface effectiveness was best fitted to experimental data in order to minimized the RMS error between calculated and measured evaporating and condensing pressures and sensible and latent cooling capacities, so that η_{wet}=0.58 was achieved. In this case, only 6 out of 14 data points were used, as the other conditions led to a dry evaporator coil. In all cases, it was considered that the evaporator and condenser fans supplied an average air flow rate of 0.5 L/s and 1.1 L/s, respectively. Figures 3 to 6 summarize the model validation exercise, where it can be seen that model is able to predict the working pressures (Fig. 3) within a ±2% error band, and the overall cooling capacity (Fig. 4), the sensible heat ratio (Fig. 5), and the system COP (Fig. 6) within a ±6% error band.

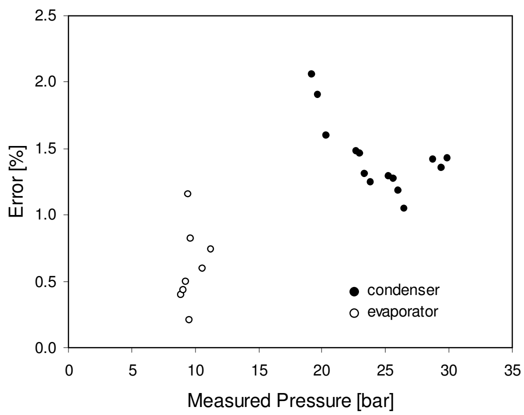

Figure 3. Model validation for the working pressures

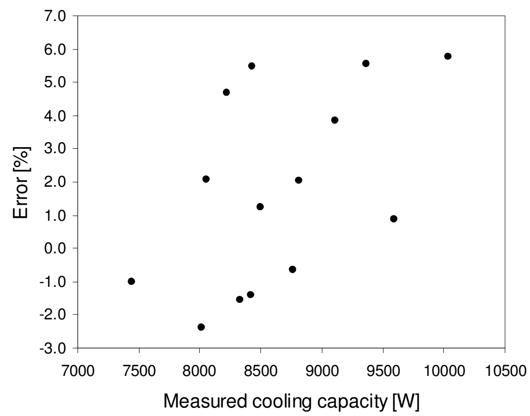

Figure 4. Model validation for the overall cooling capacity

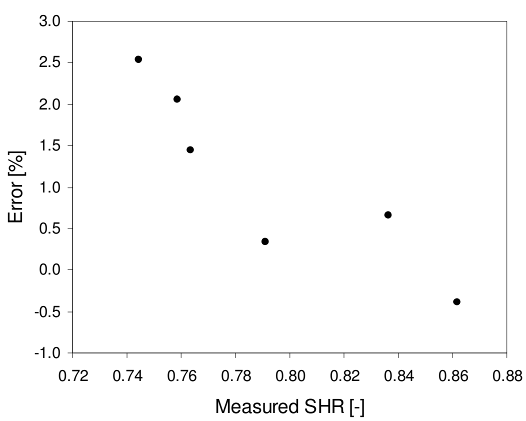

Figure 5. Model validation for the sensible heat ratio

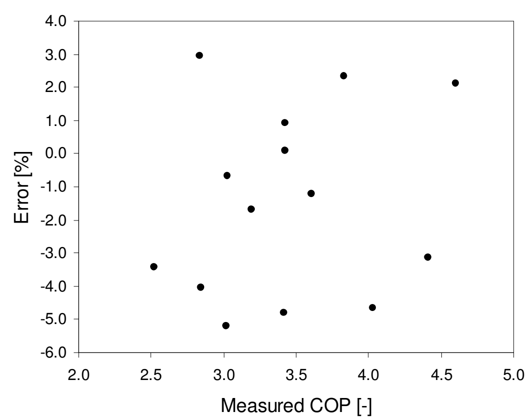

Figure 6. Model validation for the system COP

4. DISCUSSION

First, the effect of the number of transversal tubes (i.e., face area) of both condenser and evaporator on the system COP and cost was explored, as shown in Fig. 7. In this analysis, the cooling capacity was kept unconstrained. As can be noted, the system COP increases with the number of transversal tubes, so does the heat exchangers cost. Whereas the latter behaves linearly, the COP presents a curvature, indicating that higher COP systems can be designed without any cost penalties by increasing the number of evaporator tubes and reducing the number of condenser tubes.

The optimization was carried out in order to find out the numbers of condenser and evaporator tubes that minimize the cost in case the COP was held fixed. In this analysis, the numbers of condenser and evaporator tubes are not independent parameters anymore. Moreover, the compressor was also resized through a scaling factor, β, in Eq. (2) and Eq. (3) to hold the cooling capacity constrained at 8400 W. Figure 8 shows the minimum cost for each possible COP value. As expected, the higher the COP the higher the optimum cost. The values above the curve are not optima, which is the case of the original system represented with a bullet. Conversely, the values below the curve do not fulfill the system constraints. It is worthy to note that the curve of Fig. 8 represents both the minimum cost and maximum COP. On one hand, following the vertical arrow from the bullet (original system), the minimum cost for a fixed COP can be found. Moving along the horizontal arrow, on the other hand, the maximum COP for a specified cost can be obtained. Whereas the optimum cost is 5% smaller than the original one, the optimum COP is 4% higher than its original counterpart.

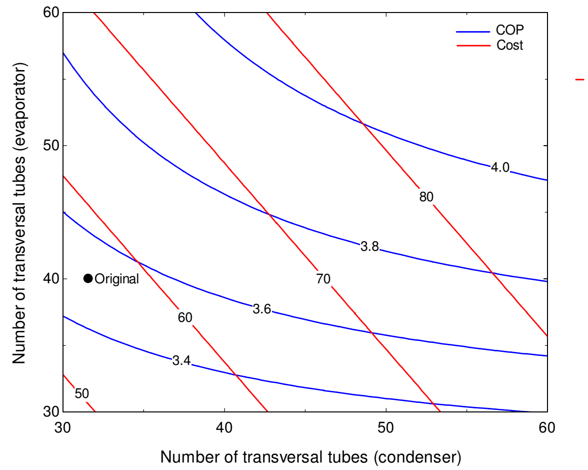

Figure 7. Effect of the number of transversal tubes on system COP and cost

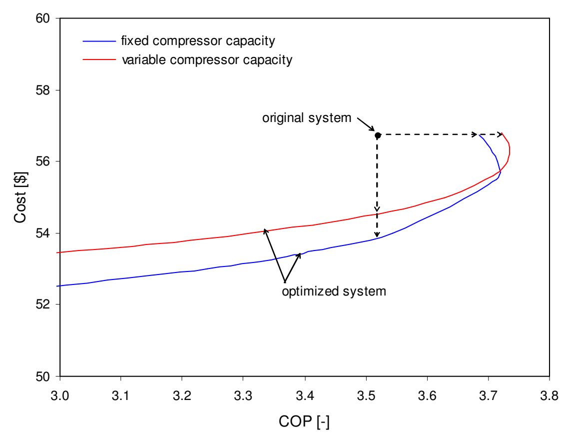

Figure 8. Optimized cost and COP by varying the number of transversal tubes

Figure 9 explores the effect of the numbers of evaporator and condenser fins on the system COP and cost. Whereas the cost increases linearly with the fins, it can be seen that the higher COP values are observed for a higher number for condenser fins and a lower number of evaporator fins, indicating that the pumping power required for the evaporator fan (fan efficiency ~ 1%) is much higher than that required by the condenser fan (fan efficiency ~ 5%). Therefore, for the same cost, the COP can be enhanced by reducing the number of evaporator fins and increasing the number of condenser fins. Figure 10 shows the minimum cost for a given COP in case when the numbers of condenser and evaporator fins were varied. Similarly to the analysis carried out by varying the number of tubes, the numbers of condenser and evaporator fins are not independent as the COP was held fixed to obtain the minimum system cost. Again, the cooling capacity was held constrained at 8400 W. Figure 10 shows that the cost increases with the COP until an inflexion point is reached. This is so as the free flow passage decreases for higher fin densities, increasing the pressure drop and so the pumping power required by the fans, which in turn affects the system COP (see Eq. 15). In this case, a cost reduction of 5% or a COP improvement of 5% were also observed.

5. CONCLUSIONS

A simulation-based design methodology for optimizing both COP and cost of residential unitary air conditioning equipment was presented. A simplified system simulation model was proposed and validated against experimental data obtained for a 2.5-ton nominal cooling capacity unit, with the model predictions for cooling capacity, SHR, COP and working pressures agreeing with the experimental data within a ±6% error bands. The model was then used to investigate the influence of some heat exchanger design parameters (number of tubes, number of fins) on the system COP and cost. For a constrained cooling capacity, the heat exchangers geometry and the compressor were resized to find out the minimum cost that can be achieved for a given COP. The simulations have pointed out that a COP enhancement up to 5% and a cost reduction of 5% can be achieved in cases when either the number of transversal tubes or the number of fins were changed individually. Of course, further analyzes are still required to assess the combined effects of free flow area and heat transfer surface, and also the influence of the pressure drops upon the fan-supplied air flow rates. These issues will be addressed in a further publication.

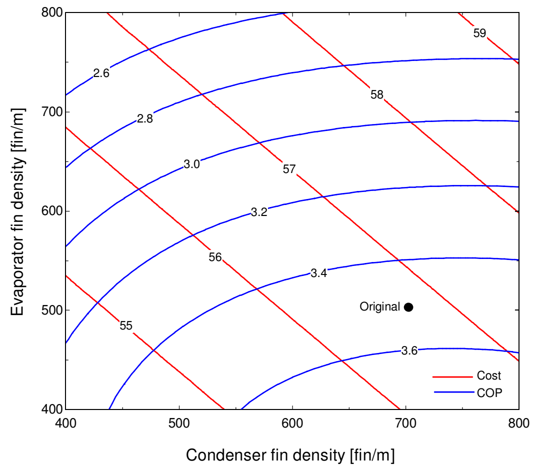

Figure 9. Effect of the number of fins on system COP and cost

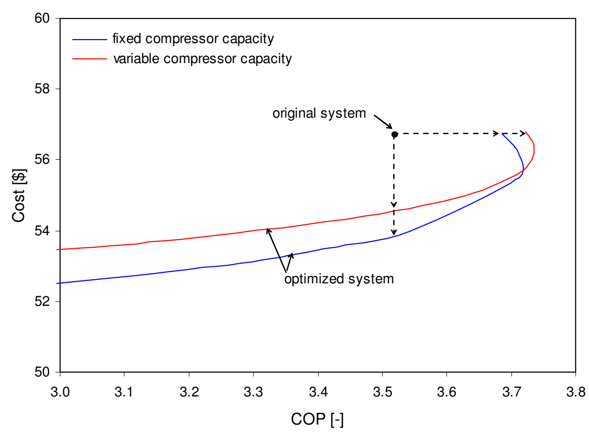

Figure 10. Optimized cost and COP by varying the number of fins

6. ACKNOWLEDGMENTS

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8. NOMENCLATURE

Roman

A, area (m^2)
a, b, compressor empirical coefficients (-)
C, cost ($)
c, unitary cost ($/kg)
COP, coefficient of performance (-)
c_p, specific heat (J/kgK)
f, friction factor (-)
G, maximum air mass flux (kg/m^{2}s)
$\hbar$, heat transfer coefficient (W/m^{2}K)
h, specific enthalpy (J/kg)
Le, Lewis number (-)
m, mass flow rate (kg/s)
M, material mass (kg)
p, pressure (Pa)
Q, heat transfer rate (W)
SHR, sensible heat transfer ratio (-)
t, temperature (K)
t_c, condensing temperature (K)
t_e, evaporating temperature (K)
V, air flow rate (m^3/s)
W, power cosumption (W)

Greek

β, compressor scaling coefficient
ε, heat exchanger effectiveness (-)
η, efficiency (-)
ρ, specific mass (kg/m^3)
ω, humidity ratio (kg$_s$/kg$_a$)

Subscripts

a, dry air
c, condenser
e, evaporator
f, fin surface
i, evaporator inlet
k, compressor
m, moist air
o, outdoor
r, refrigerant
sat, saturation
sub, subcooling degree
sup, superheating degree
t, tube surface
x, heat exchanger (condenser or evaporator)