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ON THE MODELING OF A ROOM AIR CONDITIONER

Gustavo Cherem-Pereira, gustavo.perereira@electrolux.com.br
Electrolux of Brazil
Pontifical Catholic University of Parana - PUCPR
Mechanical Engineering Graduate Program
Nathan Mendes, nathan.mendes@pucpr.br
Pontifical Catholic University of Parana - PUCPR
Mechanical Engineering Graduate Program

Abstract. *Evaporator and condenser of a room air conditioner is semi-empirically modeled based on test data and some parameters, mixing both analytical and empirical method. The methodology allows to obtain a comprehensive model to be integrated to building energy simulation models and to be an interesting tool to analyze and optimize the system as well as learn about the influence of changing or modifying components on the global behavior of the air conditioner. The experimental data, from which the heat exchangers models were created, were obtained in calorimeters according to ISO standard. The results show good agreement when compared to an extensive experimental database where various parameters are varied.*

Keywords: *Room Air Conditioners, Heat Exchangers, Semi-empirical Modeling*

1. INTRODUCTION

In order to reduce energy consumption, since the energy crisis in the 70's, simulation models have been developed to reduce the energy consumption on HVAC systems, especially in central air conditioning systems. In general, air conditioners models found in the literature can be analytical, semi-empirical or empirical. The semi-empirical modeling is based on models for each component of the refrigeration system, using a formulation based on physical principles and correlations obtained from experiments providing an interesting tool to optimize the refrigeration system.

However, there is still a lack of models to predict equipment performance, especially of room air conditioners. Building energy simulation programs employ empirical modeling of air conditioning systems, based on performance data map normally provided by equipment manufacturers (Cherem-Pereira and Mendes, 2012 and Korolijaa *et al.*, 2011). Alternatively, using semi-empirical modeling, more detailed information can be provided and other results can be examined such as the energy consumption reduction impact when a less efficient compressor is replaced

In this way, this work presents a development of a semi-empirical model of a room air conditioner. The effectiveness concept and experimental data are used in order to develop new semi-empirical models for predicting condenser and evaporator behaviors. The models presented are integrated to compressor and capillary tube models and each component model is validated from tests carried out in calorimeters according to ISO standard (ISO 5151: 1994). These calorimeters and associated measurement uncertainties are also described in details in the present paper.

2. EVAPORATOR MODEL

The model does not describe the evaporator mathematically detailed but describes it with a reduced number of experiments. Defining the heat transfer global coefficient as function of a mean air enthalpy difference for wet finned-tube evaporators, Eq. 1, a semi-empirical method of modeling was developed.

$$\Delta i_m = \frac{(i_i - i_i^s) - (i_o - i_o^s)}{\ln\left(\frac{i_i - i_i^s}{i_o - i_o^s}\right)}. \tag{1}$$

With few approximations the heat transfer rate can be expressed by Eq.2:

$$\dot{q}_{ev} = U_{ev} A_t \Delta i_m . \tag{2}$$

Therefore, the evaporator effectiveness can be expressed by Eq. 3:

$$\varepsilon = \frac{\dot{q}_{ev}}{\dot{q}_{\max}} = \frac{U_{ev} A_t \Delta i_m}{\dot{m}_a (i_i - i_{a,te}^s)}. \tag{3}$$

Substituting Δi_m for its definition showed in Eq. 3, considering equals the fictitious air saturated enthalpies at refrigerant temperature on the evaporator inlet and outlet, using the effectiveness definition of Eq. 4 and summing with a few of algebra, the evaporator effectiveness and the evaporator heat transfer global coefficient can be expressed by Eqs. 5 and 6.

$$\varepsilon = \frac{(i_i - i_o)}{(i_i - i^s_{a,te})}. \tag{4}$$

$$\varepsilon = 1 - \exp\left[\frac{-U_{ev}A_t}{\dot{m}_a}\right]. \tag{5}$$

$$U_{ev}A_t = \dot{m}_a \ln\left[\frac{(i_o - i^s_{a,te})}{(i_i - i^s_{a,te})}\right]. \tag{6}$$

The term $U_{ev}A_t$ from Eq. 6 may be obtained experimentally defining the model that requires at least three experimental points for calculating the evaporator total heat transfer rate. Four characteristics must be measured or calculated starting from experimental data, they are: $i_{int,i}$, $i_{int,o}$, T_{ev} and $\dot{m}_a$. With them the term $U_{ev}A_t$ can be calculated by Eq. 8 for the experimental points and also the evaporator effectiveness and the heat transfer rate can be calculated by Eq. 7 and 9, respectively.

The room air conditioner experimental tests were performed in a psychrometric calorimeter (ISO 5151, 1994) where the evaporator and condenser data were obtained from. The $U_{ev}A_t$ mean value from experimental data can be used to predict the evaporator performance. Aiming to reduce predicting errors the term $U_{ev}A_t$, in the present work, was correlated to the inlet air enthalpy of evaporator thus the accuracy of heat transfer rate was improved in 50%. Fig. 1 shows the correlation obtained. After reached the correlation for some experimental data, the $U_{ev}A_t$ value can be predicted for any environment conditions in both indoor and outdoor sides. Forty nine experimental data were created, in eight of them it was not occurred water condensation on the fin surfaces. In those cases, the term $U_{ev}A_t$ is overestimated because the lower difference between inlet and outlet air enthalpies. The error can be up to 40% and it must be corrected systematically.

After calculated $U_{ev}A_t$, the evaporator effectiveness must be calculated by Eq. 6 and finally the total heat transfer rate can be obtained by Eq. 7:

$$\dot{q}_{ev} = \varepsilon \cdot \dot{m}_a (i_i - i^s_{a,te}). \tag{7}$$

The air flow to be used in Eq. 7 can be the experimental data mean value but to minimize heat transfer rate errors, the air flow can be correlated to the characteristic enthalpy.

Figure 2 shows the relative error found between calculated and predicted heat transfer rate for the 49 test points, where 95.9% of points remains between 10% of error.

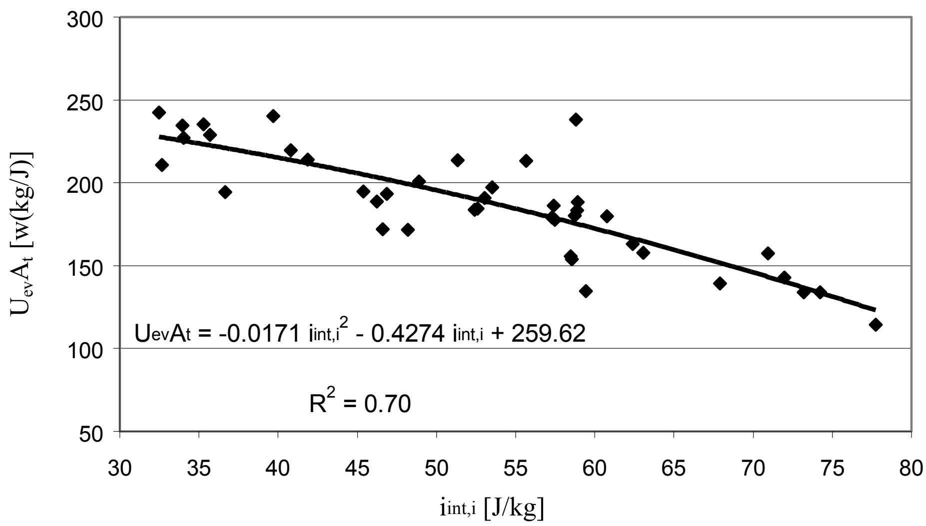

Figure 1. $U_{ev}A_t$ correlation.

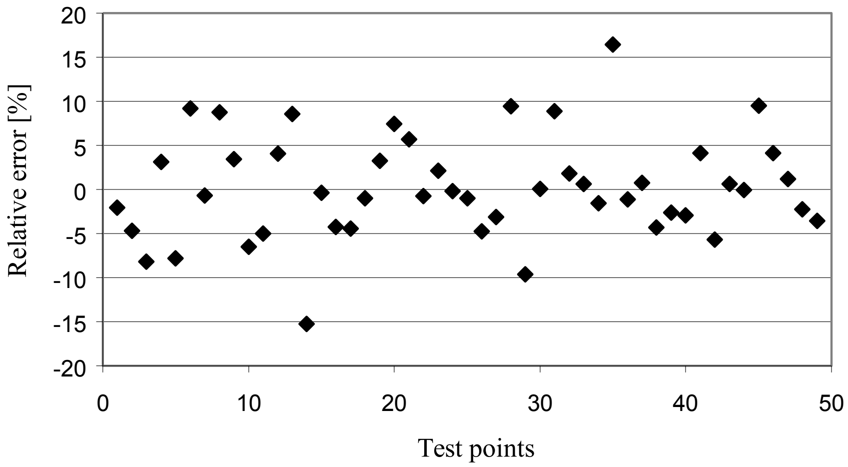

Figure 2. Evaporator heat transfer rate relative errors.

3. CONDENSER MODEL

Similarly to the evaporator, the air side condenser effectiveness is the ratio of the maximum possible heat transfer rate to the real heat transfer rate:

$$\varepsilon = \frac{\dot{q}_{cd}}{\dot{q}_{\max}} = \frac{U_{cd} A_t \Delta T_{\ln}}{C_{\min}(T_{cd} - T_i)}, \tag{8}$$

where:

$$\Delta T_{\ln} = \frac{(T_{cd} - T_i) - (T_{cd} - T_o)}{\ln\left[\frac{(T_{cd} - T_i)}{(T_{cd} - T_o)}\right]}. \tag{9}$$

Substituting the definition of condenser effectiveness of Eq. 10 and also substituting Eq. 9 in Eq. 8 and then working with few of algebra the condenser effectiveness and the term $U_{cd}A_t$ can be expressed by Eq. 10 and 11.

$$\varepsilon = \frac{(T_o - T_i)}{(T_{cd} - T_i)}. \tag{10}$$

$$\varepsilon = 1 - \exp\left[\frac{-U_{cd} A_t}{\dot{m}_a c_{p,a}}\right]. \tag{11}$$

$$U_{cd} A_t = \dot{m}_a c_{p,a} \ln\left[\frac{(T_{cd} - T_i)}{(T_{cd} - T_o)}\right]. \tag{12}$$

The required parameters to calculate $U_{cd}A_t$ must be obtained, in minimum, with three experimental points. The average of the three $U_{cd}A_t$ must be used to estimate the condenser effectiveness by Eq. 11 and then the total heat transfer rate by Eq. 13.

$$\dot{q}_{cd} = \varepsilon \cdot \dot{m}_a c_{p,a} (T_{cd} - T_i) \tag{13}$$

Figure 3 shows the errors between measured and calculated heat transfer rate, where 90.9% of points remain within 10% relative errors.

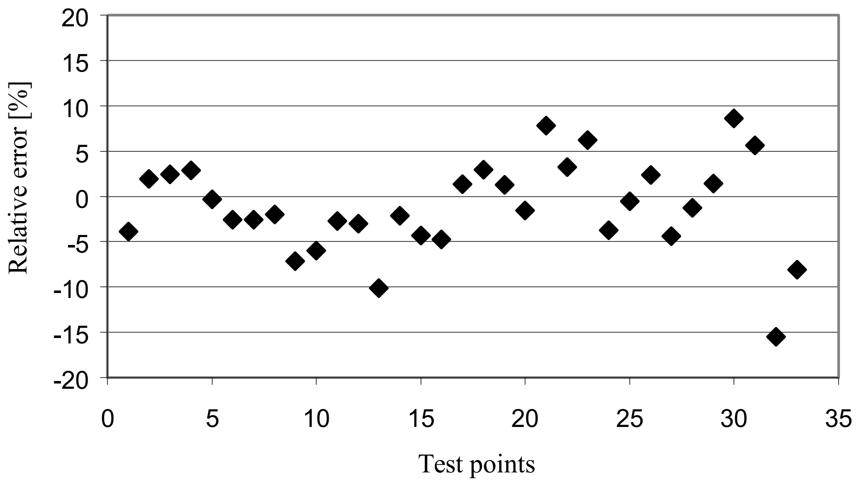

Figure 3. Condenser heat transfer rate relative errors.

4. MODEL RESULTS ON THE AIR CONDITIONER MODELING

In the previous sections the modeling and validation for the heat exchangers have been presented. The component models are integrated and results are obtained in terms of cooling capacity, compressor power, condenser heat rejection, evaporating temperature and energy efficiency ratio, using models presented by Cherem-Pereira (2003).

In order to assemble the four component models, an integration algorithm must be implemented so that the energy and mass transfer between them be respected, according to one or more convergence criteria, which need to be compatible with each model.

The calculation program starts with the compressor simulation, which provides the refrigerant mass flow and the thermodynamic state of the refrigerant fluid at the compressor outlet. Then the condenser simulation is run obtaining the refrigerant thermodynamic state at the condenser outlet and capillary inlet. In the sequence, the capillary tube model estimates the refrigerant mass flow and the refrigerant state at the capillary outlet. In the case, the two mass flows - predicted by the compressor and by the capillary tube - are different, adjustments on the condensing pressure are carried out until respecting the mass flow balance between the compressor and capillary tube. This step is the first iterative process, since a rise on the condensing pressure causes an increase on the mass flow predicted by the capillary tube model and a decrease on the mass flow obtained by the compressor model.

After reaching the mass balance, the next calculation is on the evaporator heat exchange to obtain the refrigerant thermodynamic state at the evaporator inlet from the refrigerant state at the compressor inlet. If those thermodynamic refrigerant states do not match, the second iterative process starts varying the evaporating pressure as the evaporator energy performance increases when the evaporating temperature decreases.

All the testes carried out in the calorimeter were also simulated, so that the results are presented for all cases. Figure 4 shows the relative errors between the measured and simulated cooling capacity and Figure 5 shows the same errors for the E.E.R. (Energy Efficiency ratio). For the cooling capacity, 85.3% of the points are within a 15% error band, which can be considered good since the points shown in Figs. 4-5 include tests for five different 1.6-mm capillary tubes, which length was varied from 615 mm and 1355 mm. The testes also include an extensive range of temperature and relative humidity..

We can notice in Fig. 4 that the errors are positive in most cases. This tendency is because the model converges to lower condensing and evaporating pressures and as consequence to higher cooling capacities, higher E.E.R.'s and higher supply air temperatures. This phenomenon is credited to the hypothesis used in the present modeling of adiabatic capillary tube. Even in this way, the model could be calibrated so that the pressures would be predicted at higher values, providing in that way 97% of test points within a 12.5% relative error band and the errors would fluctuate around the zero value.

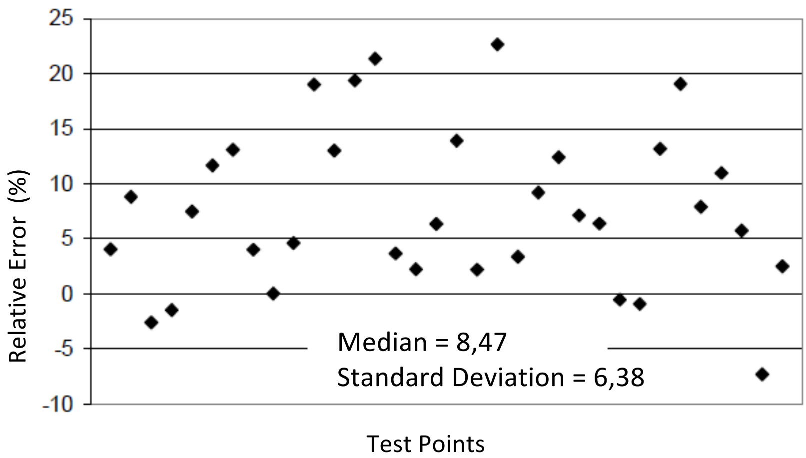

Figure 4. Relative errors for simulations – Cooling Capacity.

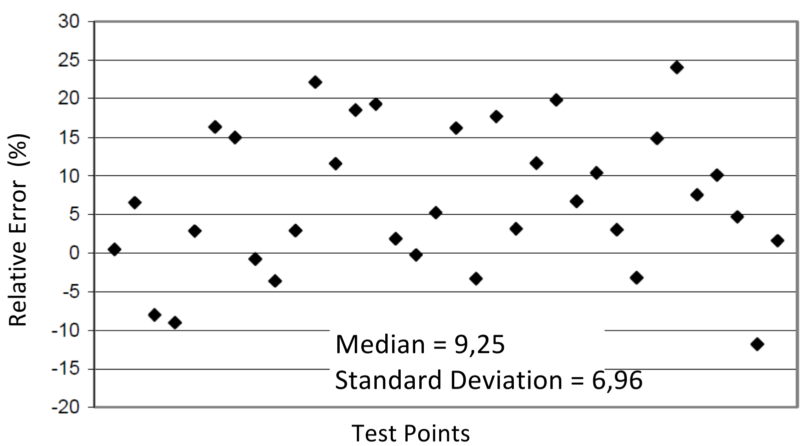

Figure 5. Relative errors for simulations – E.E.R..

Figures 6-7 present a sensitivity analysis carried out on cooling capacity, condensation temperature, compression power, evaporation temperature, condenser heat loss and EER, by varying just one parameter at a time. The reference values (= 1) are the ones calculated under standardized conditions for cooling capacity tests: Tdb,in= 26,7°C, Twb,in=19,4°C, Tdb,out=35°C – according to ANSI/AHAM RAC-1(1992) - for each variable. We can notice in Fig. 6 that an increase on the indoor temperature increases the difference of temperature between air and refrigerant and therefore increases the cooling capacity. The higher the heat exchange at the evaporator the higher the evaporating and condensing pressures. The evaporating pressure has been shown more sensible. However, the compression power rises due to a higher refrigerant mass flow caused by a higher condensing pressure and is more representative than a lower compression rate. Despite the increase on the compressor power, the cooling capacity increase is more noticeable and therefore the air-conditioner efficiency (E.E.R.). Finally the total rejected heat at the condenser rises due to higher evaporating temperature, cooling capacity and compression power.

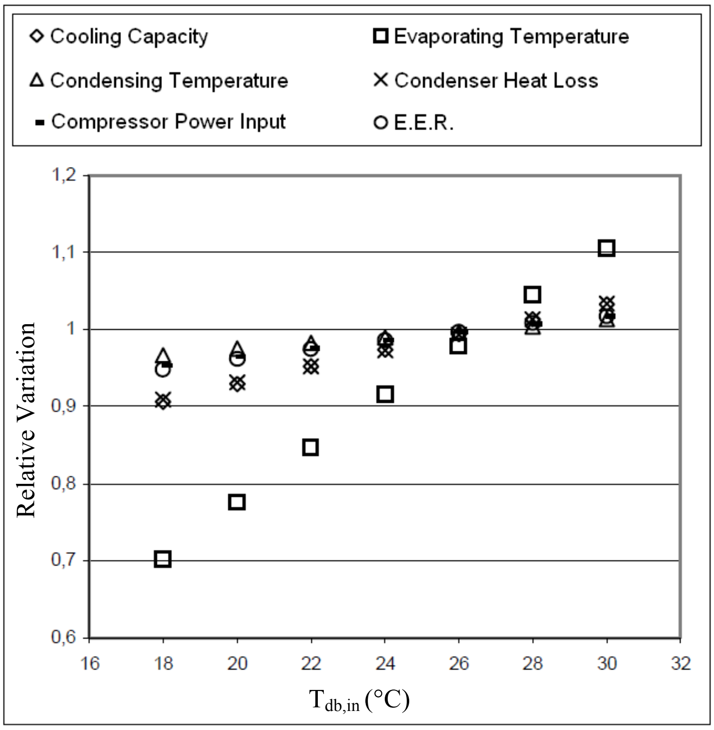

Figure 6. Variation of characteristics as function of indoor dry-bulb temperature (W_{out}=0,010kg/kg, $T_{db,out}$ = 35°C).

Figure 7 illustrates the relative variations according to changes on the outdoor dry-bulb temperature. It has been notice that increasing this temperature, the condenser heat loss decreases due to a lower temperature difference between air and refrigerant fluid. As a consequence, the evaporating and condensing temperatures rise. Increasing the evaporating temperature, the cooling capacity and the refrigerant specific volume at the compressor inlet decrease.

However, the higher the condensing temperature the higher refrigerant flow, which strongly influences the compression power. The E.E.R. decreases because the compression power rises and the cooling capacity decreases when there is an increase on the outdoor dry-bulb temperature.

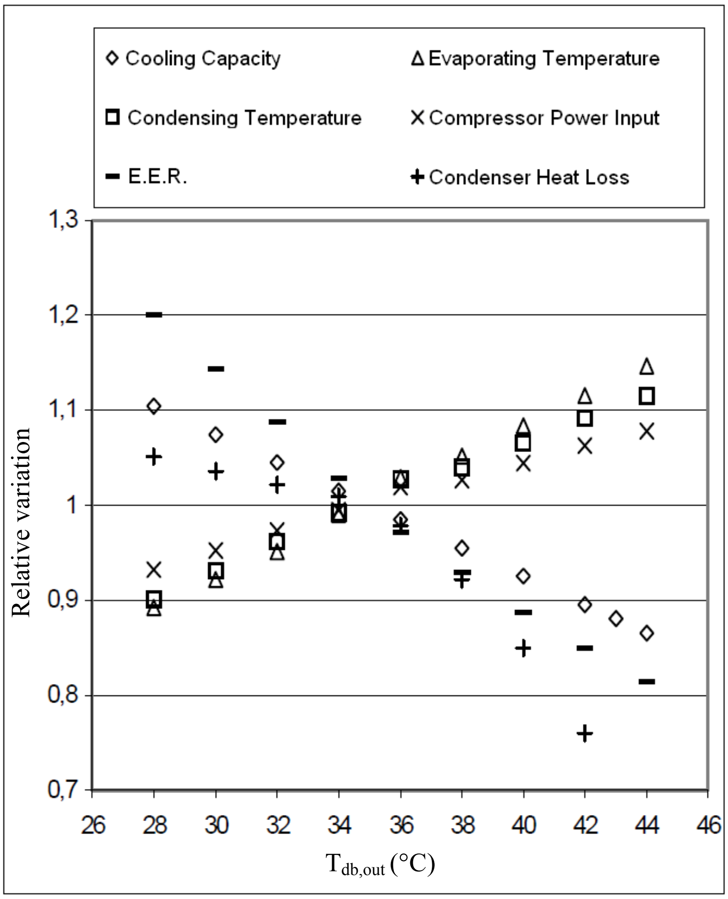

Figure 7. Variation of characteristics as function of outdoor dry-bulb temperature ($T_{db,out}$ =35.0°C, $T_{db,in}$ = 26.7°C).

It is also noticed that despite the impact mentioned above with the increase on the condenser overall heat transfer coefficient or condenser heat exchange area, the condenser heat loss remains barely unchanged since the condensing temperature decreases due to both an increase on the cooling capacity and a decrease on the compressor power.

Therefore it is clear that the condenser is the most important component and an improvement on its design may result on a great building energy consumption reduction, once the compressor performance varies enormously in the market according to their price.

The model allows the analysis of several other parameters such as total area of heat exchangers, indoor wet-bulb temperature, capillary tube length, pressure loss at suction and discharge chamber compression valves and superheating degree before compression chamber among others.

5. FINAL REMARKS

The semi-empirical evaporator model results were compared to 49 test points, where 95.9% of points had relative errors below 10.0%. For the condenser, the semi-empirical model results were compared to 31 test points, where 90.9% of the points have errors of up to 10.0%.

The semi-empirical model has the advantage of monitoring various parameters and characteristics of the system compared to modification of other features and it is easier to be integrated into building energy simulation tools, allowing to speed up simulations and decreasing numerical divergence problems. On the other hand, they do not allow to determine the optimum refrigerant mass load and the thermodynamic state of the refrigerant fluid.

6. ACKNOWLEDGEMENTS

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