International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2008

# A Model for the Analysis of Compact Heat Exchangers

Josef Ramon Garcia-Cascales
*Technical University of Cartagena*

F. Vera-Garcia
*Technical University of Cartagena*

J. Gonzalvez-Macia
*Technical University of Valencia*

J. M. Corberan-Salvador
*Technical University of Valencia*

M. W. Johnson
*Modine Manufacturing LTD*

***See next page for additional authors***

**Authors**
Josef Ramon Garcia-Cascales, F. Vera-Garcia, J. Gonzalvez-Macia, J. M. Corberan-Salvador, M. W. Johnson, and G.t. Kohler

# A model for the analysis of compact heat exchangers

JR García-Cascales[1*], F Vera-García[1], J Gonzálvez-Maciá[2], JM Corberán-Salvador[2]
MW Johnson[3], GT Kohler[3]
[1]Technical University of Cartagena, Thermal and Fluid Engineering Department, Cartagena, Murcia, Spain (+34 968 325 991, jr.garcia@upct.es, +34 968 325 987, francisco.vera@upct.es)
[2]Technical University of Valencia, Applied Thermodynamic Department, Valencia, Spain (+34 963 877 323, corberan@ter.upv.es, +34 963 879 127, jgonzalv@upv.es)
[3]Modine Manufacturing LTD, Heat Exchanger Division, Racine, Wisconsin, US (M.W.Johnson@na.modine.com, g.t.kohler@na.modine.com)

## ABSTRACT

In this paper, a model for the analysis of compact heat exchangers working as evaporators or condensers is studied. It has been implemented in a code for their selection. We are interested in the type of heat exchangers mostly used in air-conditioning and refrigeration applications. The heat exchangers are discretized in cells and analyzed following the path imposed by the refrigerant flowing through the tubes. The compact heat exchanger is formed by finned multiport tubes with microchannels through which the refrigerant flows. They may have two-rows. In this case, an iterative procedure is performed in order to obtain the unknowns of the problem design. Some experiments are performed and the data obtained are used to evaluate the results provided by the model. The working fluid is R134a and the secondary fluid is air. The facility is briefly described and some conclusions are finally drawn.

## 1. INTRODUCTION

The objective of this paper is to develop a model for the analysis of compact heat exchangers and implement it in a code for the selection of this type of heat exchanger in air-conditioning and refrigeration system design. In this case, our main concern is to have a robust model for the analysis of compact heat exchangers working as condensers and evaporators. In addition, the model is to deal with multiple pass and multiple row heat exchangers. In this work, each heat exchanger tube is discretized into cells so the analysis process follows the path imposed by the refrigerant flow and takes into account the heat exchanged with the air flowing through the heat exchanger. This methodology sequentially studies the heat exchanger, after defining a suitable sequence of cells in it. The discretization procedure follows that developed for the analysis of coils by Corberán and García (1998).

An iterative procedure is proposed when the heat exchanger has two rows. It is necessary in order to make the sequential method converge to a correct solution. To understand this procedure, a two row heat exchanger is considered. To calculate the heat exchanger outlet variables, the following variables are known: Air inlet temperature $T_{ai}$ and air specific humidity $\phi_{ai}$, at the inlet of the heat exchanger (just before the second row) and refrigerant inlet conditions $p_{ri}$ and $h_{ri}$ (Figure 1). The problem arises if a sequential procedure is followed. The air inlet conditions to the first row are not known and an iterative procedure is needed.

This has been implemented in a program called M-Power described by Kohler et al. (2006) in the last 2006 Refrigeration and Air Conditioning Conference. This can work with different refrigerants. Their thermodynamic properties are internally evaluated by means of the NIST REFPROP subroutines. The correlations used to characterize the heat transfer and pressure drop at each side of the heat exchanger (refrigerant and air sides) are typical correlations used in the existing literature for single and two-phase flow. The outputs of the energy balance are the fluid variables at the outlet of the cell and the heat exchanged in the cell. These values are the inlet of the following cell in the sequential process introduced above.

So, the work presented in this paper has been structured as follow, firstly the model developed and the algorithm implemented in the code are described. The topology of the compact heat exchangers studied is briefly described. After, the facility and the experimental results measured in the laboratory are presented. The results provided by the code and some comments on these results are explained. It is completed comparing with some experimental results obtained at Modine laboratories. Finally, some conclusions are drawn.

## 2. MODELING OF A MULTIPLE-ROW MULTIPLE-PASS HEAT EXCHANGER

The aim of this model is to develop a robust model able to study heat exchangers with multiple-rows and multiple-passes. To do so, the heat exchangers (evaporators and condensers) are discretized in elements, so that each tube is divided in a number of cells following the path imposed by the refrigerant flow. After defining the sequence of cells in the heat exchanger, the system is sequentially studied. Energy and momentum balances are considered for each cell so the output variables of a cell are sequentially used as input variables of the following cells.

This discretization methodology is inspired in that developed by Corberán and García (1998).for the analysis of coils. When the heat exchanger has two rows, an iterative procedure has been proposed. It is necessary in order to make the sequential method converge to the correct solution. To understand this procedure, let us consider the two row heat exchanger in Figure 1. To calculate the heat exchanger outlet variables, the following variables are known: Air inlet temperature $T_{ai}$ and air specific humidity $\phi_{ai}$, at the inlet of the heat exchanger (just before the first row) and refrigerant inlet conditions $p_{ri}$ and $h_{ri}$. A problem arises if a sequential procedure is followed. The air inlet conditions to the second row are not known and an iterative procedure is needed.

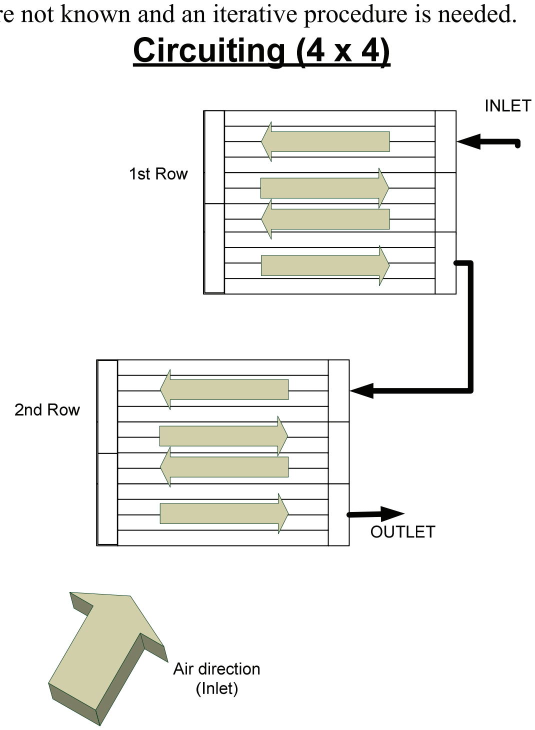


Figure 1. Two row heat exchanger.

The algorithm implemented in the code is schematically described in the diagram shown in Figure 1. As a first guess, the model supposes an inlet temperature for the first row equal to that in the second row and a value of the heat exchanged in the heat exchanger. Convergence is easily reached, by following an algorithm which recalculates the temperature at the inlet of the first row (outlet of the second one) and the heat exchanged in the heat exchanger.

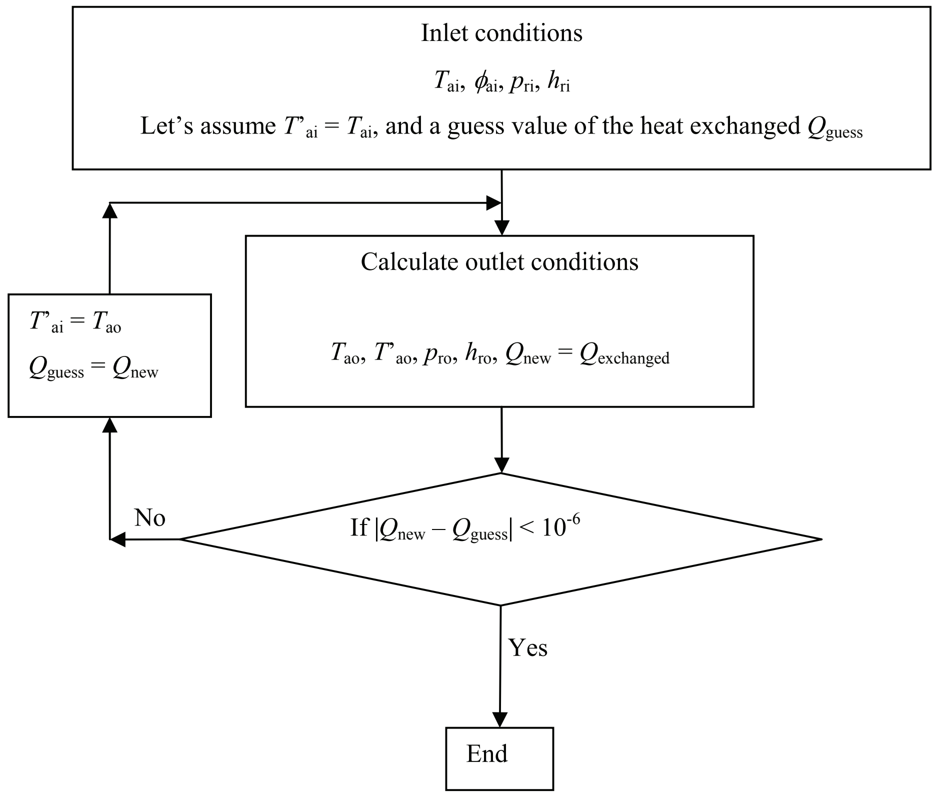


Figure 2. Algorithm for the evaluation of the outlet conditions in the two row heat exchanger.

As far as the code is concerned, it is able to work with different refrigerants. Their thermodynamic properties are evaluated by means of the NIST REFPROP subroutines. The correlations used to characterize heat transfer and pressure drop at each side of the heat exchanger (refrigerant and air sides) are typical correlations used in the existing literature for single and two-phase flow. The outputs of the energy and momentum balances are the fluid variables at the outlet of the cell and the heat exchanged in this cell. As was mentioned above, these values are the inlet of the following cell in the sequential process introduced above.

## 3. EXPERIMENTAL FACILITY

The experimental facility is briefly described in Figure 3. The condenser has been put in a wall so air goes through the heat exchanger coming from a climate chamber at the conditions gather in Table 2 and 3. In order to test the model, the following variables have been measured:
In the case of the secondary fluid – air, the Inlet air dry-bulb temperature (ºC), the inlet air wet-bulb temperature (ºC), the outlet air dry-bulb temperature (ºC), the atmospheric pressure (kPa), the volumetric flow rate at standard conditions ($m^3/h$) and in the case of the refrigerant – 134a, the inlet temperature (ºC), the outlet temperature (ºC), the inlet saturation temperature (ºC), the outlet saturation temperature (ºC), the outlet subcooling (ºC), the mass flow rate (kg/s), the pressure drop (kPa), and the heat transferred (kW).

In addition to the geometrical information of the heat exchanger, the program needs the following input variables: inlet air dry-bulb temperature, inlet air wet-bulb temperature, the atmospheric pressure and the volumetric flow rate in the case of the air. In the case of the refrigerant, the following variables are required: Inlet temperature, inlet saturation temperature and mass flow rate. The input data dialog box is shown in Figure 4.

Figure 3. Test facility.

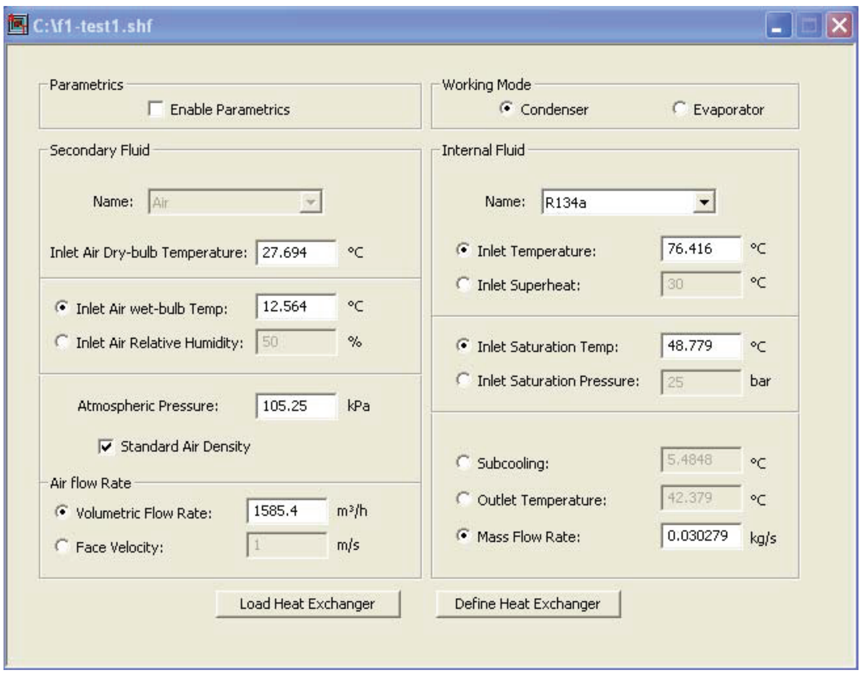


Figure 4. Input data dialog box.

Two heat exchangers have been studied; their geometries are described in Table 1. Both of them have two rows, four passes per row and are made of aluminum.

Table 1. Geometric description of the heat exchangers.

| Case | Heat exchanger 1 | Heat exchanger 2 |
|---|---|---|
| Tubes per row | 33 | 40 |
| Fin type | Louvered fins | Louvered fins |
| Fins per inch | 12 | 10 |
| Core width (mm) | 340 | 250 |
| Finned length | 410 | 250 |

The cross section of the tube is rectangular; a reference schematic of the tube is shown in Figure 5. The tube has 19 mm in length (L) and 1.9 mm in height (H).

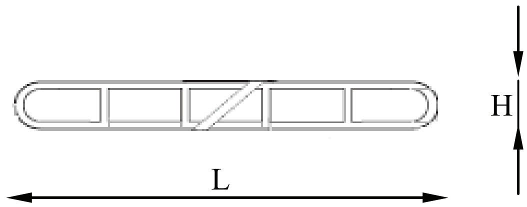


Figure 5. Cross section of the tube.

A schematic picture of one of the heat exchangers studied in this paper is included in Figure 6,

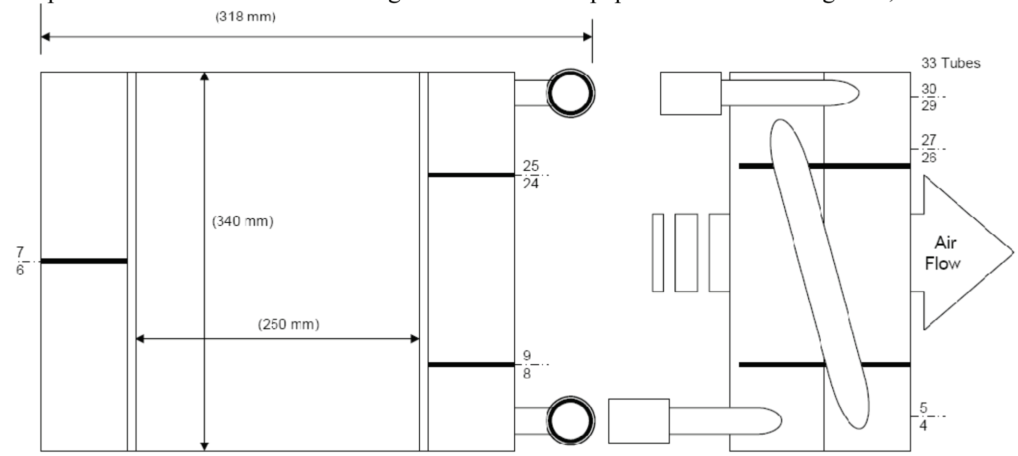


Figure 6. Schematic picture of the 33 tube per row heat exchanger.

## 4. RESULTS

The data measured in the facility corresponding to the two heat exchangers we have studied are summarized in Tables 2 and 3.

Table 2. PF2 – HE1. Experimental results.

| **Test** | **pf2 - t1.1** | **pf2 - t1.2** | **pf2 - t1.3** | **pf2 - t1.4** |
|---|---|---|---|---|
| **Secondary fluid – air** | | | | |
| Inlet air dry-bulb T (ºC) | 27.69 | 27.81 | 27.85 | 27.79 |
| Inlet air wet-bulb T (ºC) | 12.56 | 12.39 | 12.32 | 12.26 |
| Outlet air dry-bulb T (ºC) | 38.93 | 39.97 | 41.11 | 42.33 |
| Atmospheric pressure (kPa) | 105.25 | 105.26 | 105.27 | 105.27 |
| Volumetric flow rate ($m^3/h$)* | 1585.39 | 1304.26 | 1024.36 | 748.00 |
| **Refrigerant – 134a** | | | | |
| Inlet temperature (ºC) | 76.42 | 76.58 | 76.63 | 76.43 |
| Outlet temperature (ºC) | 41.98 | 42.38 | 42.35 | 42.58 |
| Inlet saturation (ºC) | 48.78 | 48.77 | 47.70 | 48.07 |
| Outlet saturation (ºC) | 47.46 | 47.70 | 48.07 | 48.23 |
| Outlet subcooling (ºC) | 5.48 | 5.32 | 5.72 | 5.65 |
| Mass flow rate (kg/s) | 0.03 | 0.03 | 0.02 | 0.02 |
| Pressure Drop (kPa) | 0.68 | 0.62 | 0.56 | 0.50 |
| Heat Transferred (kW) | 5.99 | 5.33 | 4.57 | 3.66 |

Table 3. PF2 – HE2. Experimental results.

| **Test** | **pf2 - t2.2** | **pf2 - t2.3** | **pf2 - t2.4** |
|---|---|---|---|
| **Secondary fluid – air** | | | |
| Inlet air dry-bulb T (ºC) | 34.93 | 34.98 | 34.98 |
| Inlet air wet-bulb T (ºC) | 15.12 | 15.10 | 15.14 |
| Outlet air dry-bulb T (ºC) | 43.21 | 43.89 | 44.90 |
| Atmospheric pressure (kPa) | 105.33 | 105.33 | 105.33 |
| Volumetric flow rate ($m^3$/h)* | 1304.59 | 1024.07 | 745.24 |
| **Refrigerant – 134a** | | | |
| Inlet temperature (ºC) | 76.42 | 76.81 | 76.57 |
| Outlet temperature (ºC) | 43.02 | 42.68 | 42.75 |
| Inlet saturation (ºC) | 48.89 | 48.93 | 48.55 |
| Outlet saturation (ºC) | 48.38 | 48.55 | 48.57 |
| Outlet subcooling (ºC) | 5.36 | 5.86 | 5.81 |
| Mass flow rate (kg/s) | 0.02 | 0.02 | 0.01 |
| Pressure Drop (kPa) | 0.48 | 0.45 | 0.43 |
| Heat Transferred (kW) | 3.64 | 3.07 | 2.49 |

(*) At standard conditions.

The results obtained with the model are shown in Table 4.

Table 4. Results obtained with the model.

| **Test** | **pf1 - t1** | **pf1 - t2** | **pf1 - t3** | **pf1 - t4** | **pf2 - t2** | **pf2 - t3** | **pf2 - t4** |
|---|---|---|---|---|---|---|---|
| Capacity (kW) | 5.65 | 5.00 | 4.28 | 3.52 | 3.60 | 3.07 | 2.50 |
| Exchanger Efficiency (%) | 91.26 | 91.84 | 92.51 | 93.32 | 92.75 | 93.35 | 94.08 |
| UA (W/K) | 325.36 | 287.64 | 232.41 | 212.00 | 216.81 | 186.67 | 162.12 |
| NTU | 0.61 | 0.66 | 0.68 | 0.85 | 0.50 | 0.54 | 0.65 |
| **Refrigerant R134a** | | | | | | | |
| Internal surface area ($m^2$) | 0.38 | 0.38 | 0.38 | 0.38 | 0.46 | 0.46 | 0.46 |
| De-superheating area % | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 |
| Two Phase Area (%) | 39.46 | 39.46 | 49.14 | 45.91 | 69.19 | 74.19 | 74.19 |
| Subcooling Area (%) | 60.55 | 60.55 | 50.86 | 54.09 | 30.81 | 25.81 | 25.81 |
| Mass Flow rate (kg/s) | 0.03 | 0.03 | 0.02 | 0.02 | 0.02 | 0.02 | 0.01 |
| Inlet Temperature (ºC) | 76.42 | 76.58 | 76.63 | 76.43 | 76.42 | 76.81 | 76.57 |
| Outlet Temperature (ºC) | 33.71 | 34.07 | 35.53 | 34.10 | 41.65 | 41.68 | 41.00 |
| Outlet Sat. Temperature (ºC) | 48.37 | 48.41 | 47.02 | 47.80 | 48.62 | 48.71 | 48.35 |
| Inlet Superheat (K) | 27.64 | 27.81 | 28.93 | 28.37 | 27.53 | 28.26 | 28.02 |
| Outlet Subcooling (K) | 14.66 | 14.33 | 11.81 | 13.71 | 6.96 | 6.63 | 7.35 |
| Pressure Drop (kPa) | 13.41 | 11.71 | 11.35 | 8.73 | 8.79 | 7.73 | 6.28 |
| **Secondary Fluid Air** | | | | | | | |
| Total core surface area ($m^2$) | 2.06 | 2.06 | 2.06 | 2.06 | 2.14 | 2.14 | 2.14 |
| Flow rate ($m^3$/h) | 1562.80 | 1286.10 | 1010.10 | 737.44 | 1315.90 | 1033.20 | 751.84 |
| Face Velocity (m/s) | 5.14 | 4.23 | 3.32 | 2.42 | 3.58 | 2.81 | 2.05 |

In order to illustrate the results calculated by the model, Figure 7 shows a comparison between the heating capacities measured in the experiment with those provided by the model. We also compare the refrigerant outlet temperatures obtained in the experiments and those obtained by the program in Figure 8.

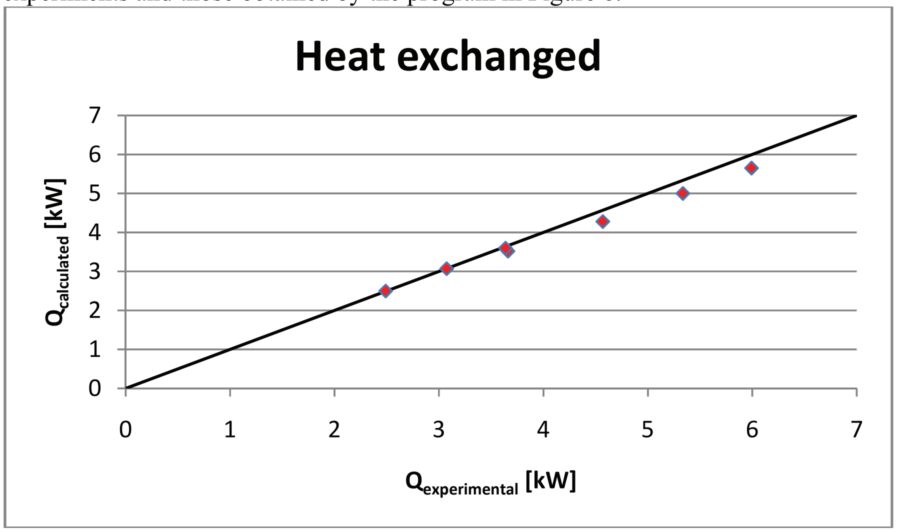


Figure 7. Comparison between measured and calculated heating capacity.

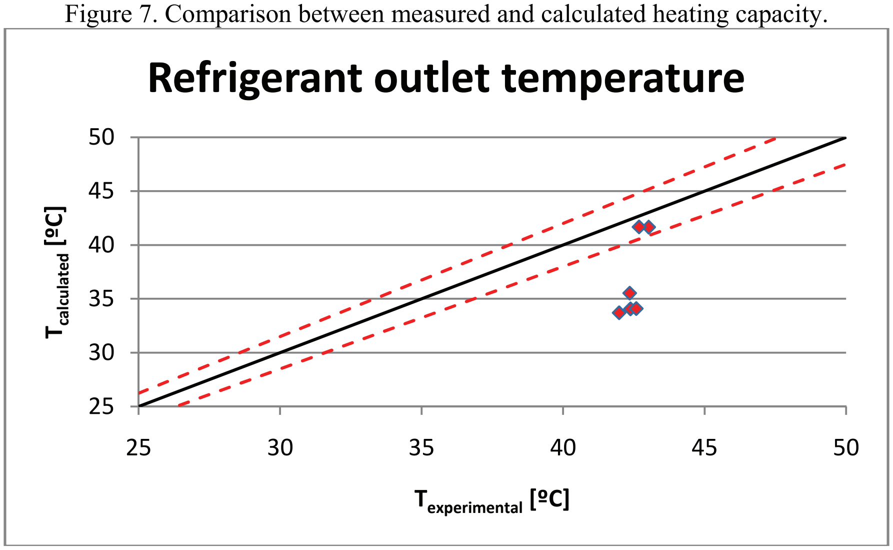


Figure 8. Comparison between measured and calculated outlet temperatures.

As far as the heat exchanged is concerned, we see good agreement between the calculated results and the experimental results. Other parameters such as the subcooling or the refrigerant outlet temperature are more sensitive and the differences are greater. We see that the latter is somewhat underestimated by the model. In order to show these differences, lines corresponding to an error of ±5% have also been depicted. We think that this might be due to the correlations used for the heat transfer coefficient and for the friction factor. In García-Cascales et al (2007a, 2007b) or Vera-García et a. (2007), the authors showed that condensation temperature (outlet saturation temperature) is more suitable in order to validate a heat transfer or a pressure drop model as it is quite sensitive to small variations of these coefficients. In Figure 9, the experimental values of the condensation temperature are compared to the calculated ones. Despite those differences in other variables, in the case of the outlet saturation temperature the calculated results are approach quite well the experimental results.

## 4. CONCLUSIONS

A model to study compact heat exchanger with multiple passes and multiple rows has been developed. It is iterative and as a first guess it supposes an inlet temperature for the second row equal to that in the first row and a value of the heat exchanged in the heat exchanger. Convergence is easily reached by following the algorithm presented above.

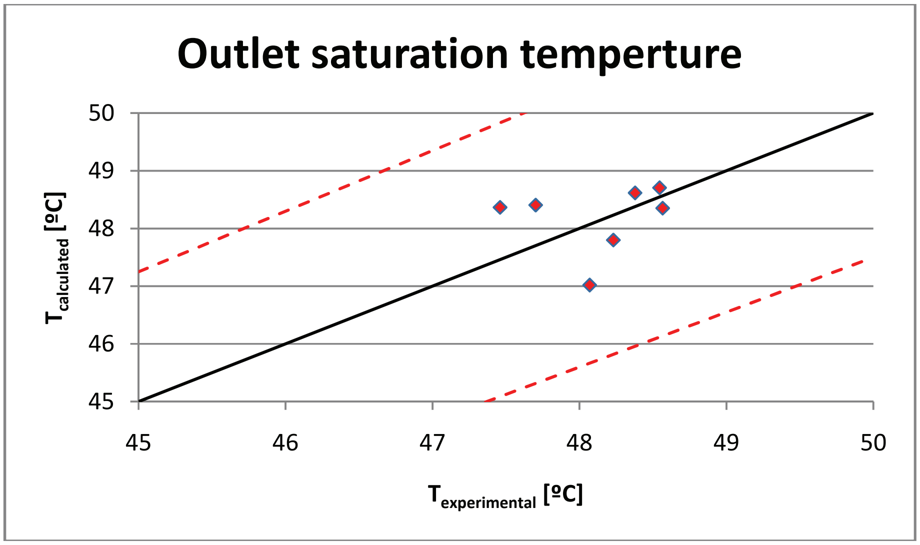


Figure 9. Comparison between measured and calculated condensation temperatures.

This has been implemented in a commercial code in order to assist the design of refrigeration equipments. Some results obtained at Modine laboratories have been compared with those provided by the code. The results are satisfactory although we think that the model included for the characterization of the heat transfer and the pressure loss should be improved. This will be the subject of a future research.

## NOMENCLATURE

| | | | | |
|---|---|---|---|---|
| h: | Specific enthalpy | (J/kg) | | **Subscripts** |
| H: | Tube height | (mm) | a: | air |
| L: | Tube length | (mm) | i: | inlet |
| p: | Presssure | (Pa) | o: | output |
| Q: | Heat transferred | (W) | r: | refrigerant |
| T: | Temperature | (ºC, K) | | |
| ϕ: | Relative humidity | % | | |

## REFERENCES

Corberán, J.M., García, M., 1998, Modelling of plate finned tube evaporators and condensers working with R134a, *Int. J. of Refrig.*,vol. 21, no. 4: p. 273-284.

García-Cascales J.R., Vera-García F., Corberán-Salvador J.M., Gonzalvez-Macia J., Fuentes-Díaz D., 2007a, Assessment of boiling and condensation heat transfer correlations in the modelling of plate heat exchangers. *Int. J. of Refrig.*, vol. 30, no. 6, p. 1004-1017.

García-Cascales J.R., Vera-García F., Corberán-Salvador J.M., Gonzalvez-Macia J., Fuentes-Díaz D., 2007b, Assessment of boiling heat transfer correlations in the modelling of fin and tube heat exchangers. *Int. J. of Refrig.*, vol. 30, no. 6, p. 1029-1041.

Kohler G., Johnson M., Gonzalvez J., Corberan J.M., 2006. 'Mpower', A Simulation Code to Assist the Design of Refrigeration and A/C Equipment Using Round Tube Plate Fin and PF2, in Proceedings of the *2006 Refrigeration and Air Conditioning Conference*, Purdue University, USA.

Vera-García F., García-Cascales J.R., Corberán-Salvador J.M., Gonzalvez-Macia J., Fuentes-Díaz D., 2007, Assessment of condensation heat transfer correlations in the modelling of fin and tube heat exchangers. *Int. J. of Refrig.*, vol. 30, no. 6, p. 1018-1028.