COMPUTATIONAL SIMULATION OF DIRECT EXPANSION EVAPORATORS

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Abstract. *This work presents a model to simulate the performance of a typical refrigeration equipment, the direct expansion evaporator. The resultant code is capable to predict the evaporator thermal capacity, the influence of the several overall tube layouts, the importance of all geometrical variables, the dehumidification process and the frost formation. Also some results are presented in order to show the influence of the heat exchanger compactness.*
The analysis for the different refrigerants used nowadays is performed by the software REFPROP 6.0, which was included in this code.
With the tube-by-tube algorithm used, it was possible to study the performance of the relative flow between the air and the refrigerant.

Key-words: *Evaporator, Heat Exchanger, Refrigeration.*

1. INTRODUCTION

This work shows the development and use of a computational tool that intend to facilitate the task of specifying direct expansion evaporators. It also presents some results that indicates the best approach for an optimum project, besides some of them are at our knowledge new in the open literature.

2. THE MODEL

2.1. Tube-by-tube method

The DX code, written in Visual Basic , was based on a tube-by-tube approximation. Each of them, with their fins has its own identity. The tubes behavior are analyzed in a proper sequence related to the overall arrange of the evaporator. The heat transfer and the thermal balance of the external humid air in cross flow, regarding the tube axis, are calculated simultaneously with the forced boiling convection, occurring in the inner part of the tube.

The program permits the choice of several tube configurations in order to evaluate the overall performance of the heat exchanger.

It assumes as given, the inlet thermodynamic state of both, air and refrigerant. Also a geometrical configuration for the evaporator is proposed (length, diameters, fins dimensions and frequency, etc.).

Through the balance equations and the proper correlations the code returns the outlet thermodynamic state of the fluids involved.

2.2. Heat transfer balance.

Considering the steady state heat transfer process for a single tube, we may write,

$$q = m_{ar}\left(h_2 - h_1\right) = m_{refr} \cdot \left(h_{rf} - h_{ri}\right) \tag{1}$$

$$q = U_{ow} A_o \Delta h_m \tag{2}$$

$$h = h_a + W \cdot h_g \tag{3}$$

Those equations are applicable to both dry and wet cooling of the air, special care for the U definition must be taken.

For the dry situation we consider

$$U_o = \left[\frac{A_o}{h_i A_{pi}} + \frac{(1-\Phi)}{h_o \cdot \left(\frac{A_{po}}{A_F} + \Phi\right)} + \frac{1}{h_o}\right]^{-1} \tag{4}$$

The analysis for the wet situation is applicable when the average surface temperature of the tube is bellow the dew point and above the freezing point. In this case, we follow the approximation presented in Threlkeld et al. (1999) . Where the global heat transfer coefficient is defined as,

$$U_{ow} = \left[\frac{b_r \cdot A_o}{h_i A_{pi}} + \frac{b_{wm} \cdot (1-\Phi_w)}{h_{ow}\left(\frac{A_{po}}{A_F} + \Phi_w\right)} + \frac{b_{wm}}{h_o}\right]^{-1} \tag{5}$$

The discussion of the physical meaning of the new parameters, appearing in the above definition was presented in Gatica (1999).

There is also the possibility of superheating the refrigerant vapor at the end of the evaporator. In this case the internal heat transfer coefficient drops drastically, controlling the heat exchange and by consequence using a substantial part of the heat transfer area. In this situation we must take special care for the evaluation of the average temperature difference. The calculation, using the effectiveness method, as presented in McQuiston e Parker (1994) are used here.

2.3. Frost formation.

When the temperature is bellow the freezing point a special provision for the frost formation is necessary. The frost thickness increases with time, and strictly speaking the process becomes transient, although the speed are very slow. In this work, we consider a quasi-steady state approach to handle this problem and as a consequence the frost influence is summed up in a thermal resistance written as $\dfrac{\delta_{fst}}{k_{fst} \cdot A_0 \cdot \Phi_0}$

The procedure outlined above was used by Oskarsson et al (1990).

2.4. Heat transfer to refrigerant

The convective heat transfer to the refrigerant, is treated as a three region problem, named as; two-phase, transition and superheated regions. The correlations used for each region and the transition criteria are the same as presented in Oskarsson, et al (1990).

2.5. External air heat transfer

The convective heat transfer coefficient at the air side is considered the same, regardless the wetness of the external surface, and we adopt the McQuiston/Rich scheme (1981).

Both the condensate film or the frost layer are considered as thermal resistances, the first was estimated by the classical Nusselt (1916) theory and the other by the Oskarsson method, *op cit.*

3. FIN TREATMENT

Geometrically the fin is usually made of a thin rectangular plate crossed by several tubes as can be seen at "Fig. 1". One way to simplify the fin influence in each tube is to make a virtual partition of the plate as shown in “Fig. 2”, e.g. McQuiston e Parker (1994). Obviously the partition is a function of the tube array.

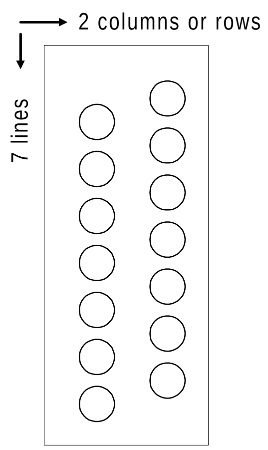

Figure 1- Lateral view of fin-plate

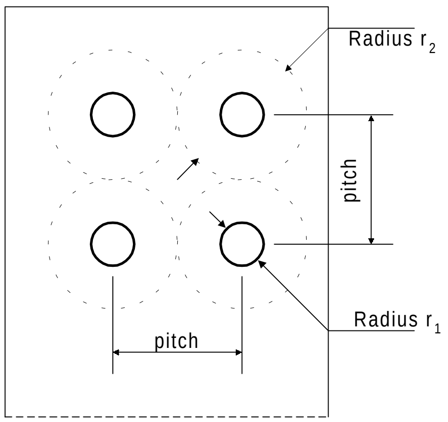

Figure.2 – 90° Tube-Array

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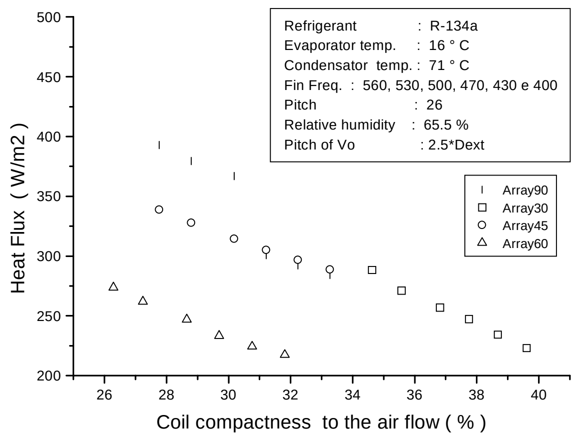
Following this procedure the fin efficiency will be taken as equal to one of uniform section and length equal to $\mathrm{L} = (\mathrm{r}_2 - \mathrm{r}_1)$.

4. RESULTS AND DISCUSSION

One parameter of great importance for the evaluation of a DX Evaporator is the degree of compactness of the heat exchanger. Although the usual definition of surface area density β (= heat exchange area / total volume) is the accepted indication of the compactness of a heat exchanger, Shah, R.K. (1981), we start our analysis with another definition for the compactness for a heat exchanger. We use one dimensionless number, named coil compactness for simplicity, which expresses the relative variation between the total volume, possible to be filled by air, and the volume available to the flow of air in a fictitious comparative heat exchanger with the same external diameter, length and array considering this time a fixed reference pitch, equal 2.5 times the used external diameter, without any fin.

The " Figure 3 " presents the heat flux absorbed by the evaporator as a function of the coil compactness for a specific evaporator (see box in figure for process conditions). The coil compactness is varying, in this situation, due the change in the fin frequency. It is possible to observe that a greater compactness will decrease the heat flux for all arrays considered. Although the internal velocities increase with larger fin frequencies, the growth in the total area is such that the heat flux decreases. This result, points out that special care must be taken when we increase the fin frequency, not only a greater pressure drop occur but a more costly heat exchanger may come up.



It can be observed in " Figure 4 ", a similar behavior considering now, the Reynolds number. The Reynolds number is also varying as a function of the change in the fin frequency.

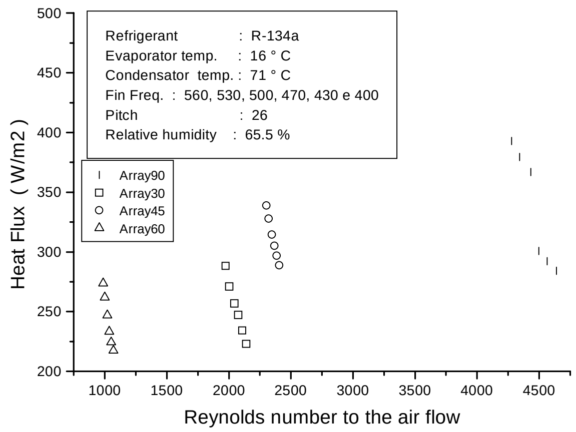


Figure 4 – Reynolds number influence

A very important parameter for the evaporator analysis is the so called by-pass factor. It can be interpreted as the ratio between the air flow, supposed unaffected by the evaporator external surface, and the total air flow. The use of this parameter as a design critical factor is presented in the classical book Carrier ( ) .

Here we want to study the dependence between the by-pass factor and the coil compactness. We may expect a strong dependence between these two variables, if we only think in the geometrical aspect, but as it can be observed in "Fig. 5" a weak correlation occur.

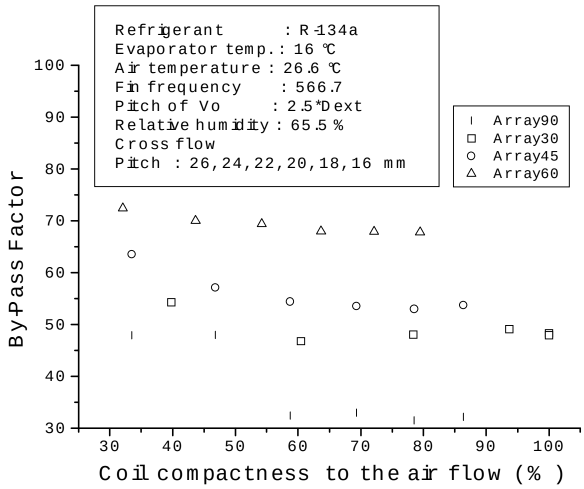


Figure 5 - The by-pass factor versus coil compactness having the tube array as parameter.

The change in the coil compactness, in the above figure, is obtained by varying the tube pitch. Due this result it seems that for a real change of the by-pass factor, the thermodynamic condition is more important than the geometrical compactness.

The next result shows an interesting use of this computational simulator .In "Fig. 6" the influence of additional rows are presented. We observe that there is no difference between the 45^o and 90^o arrays as well as the 30^o and 60^o, also after 5 rows we see no more economical gain for the service indicated.

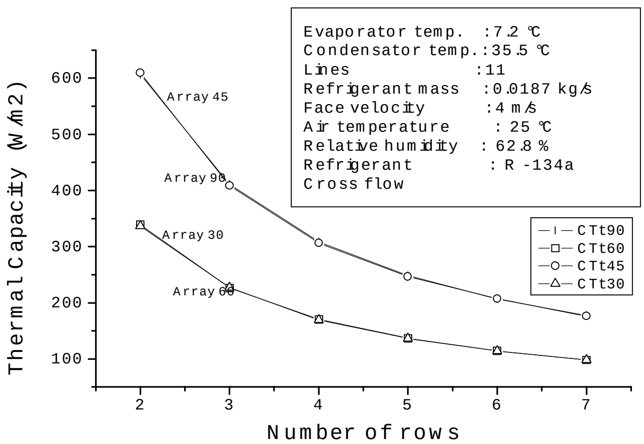


Figure 6 - Rate of heat removal as a function of the increment of the coil rows number

In relation to frost formation, it can be seen in " Fig. 7 " that at a lower row number the frost thickness is greater which is expected due the larger area to deposit the frost. considered and therefore to smallest numbers of Reynolds the growth of the frost is more efficient but all time that more rows are considered, decreasing. It can be noticed in " Fig. 8 " that all time that overheated happens in certain row the growth of the frost is reduced flagrantly. This doesn't happen when the phase is purely two-phase being observed a gradual reduction.

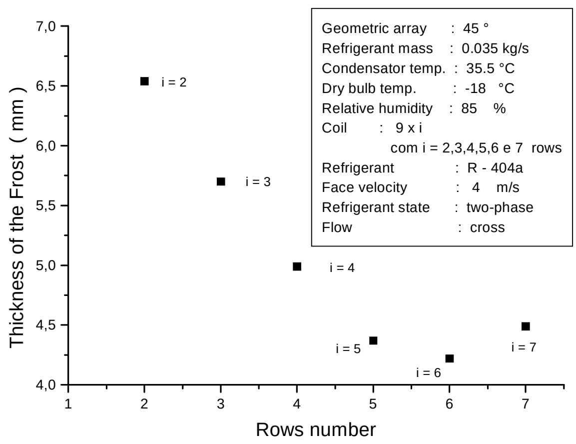


Figure 7 - Growth of the frost in function of N° of Reynolds that moves with the increment of the coil rows number of columns (i) (two-phase condition) Arrangement of 45° in cross flow

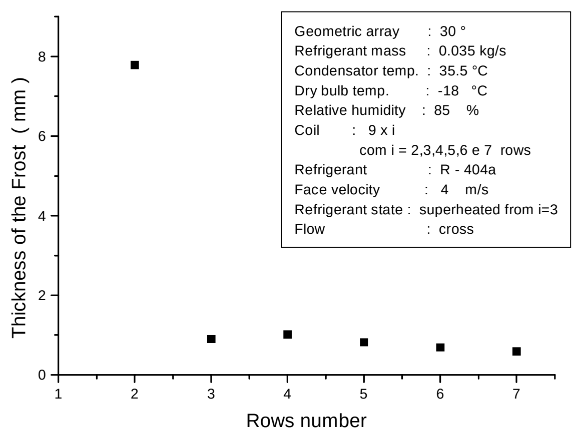


Figure 8 - Growth of the frost vs Re (it moves with increase in N° of rows " i ") in two-phase and superheated condition. Array of 30° in cross flow

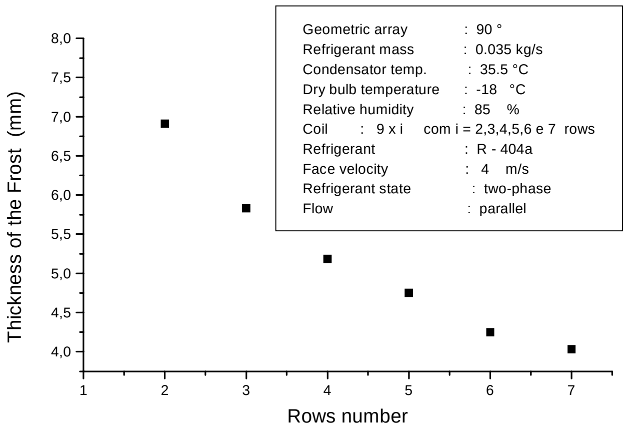


Figure 9 - Growth of the frost in function of N° of Reynolds that moves with the increment of the coil rows number (i) in condition two-phase. Arrangement of 90° in parallel flow.

An analysis of the thickness of the frost in function of the air Reynolds number is made for the arrangements 90°, 60°, 45° and 30° in " Fig. 9". The used refrigerant is R-404a. The flow is parallel. They are worth the same observations done for " Fig. 7 " and " Fig. 8 ".

5. CONCLUSIONS

A comprehensive code for direct expansion evaporator analysis was developed, and some unexpected results rose, when we analyze the influence of the geometrical compactness. Increasing this parameter may reduce the heat flux and also the by-pass factor can stay unaltered which is not an obvious result.

The program makes use of the routines for refrigerant properties calculations, developed by the National Institute of Standards and Technology-NIST/USA – McLinden, M.O. et al. (1998)

Symbols

A	:	superficial area (m^2)
b	:	enthalpy variation rate of saturation of the humid air for the temperature.
C	:	specific heat (W/°C)
Cp	:	specific heat to the constant pressure (J/kg°C)
D	:	diameter of the tube (m)
h	:	enthalpy (J/kg) or convection coefficient (W/m^2K)
h_a	:	specific enthalpy for dry air (J/kg)
$h_{g\_}$	;	specific enthalpy for saturated water vapor at temp. of the mixture (J/kg)
k	;	thermal conductivity (W/mK)
L	;	height or length of the fin (m).
m	;	mass flux (kg/s).
p	:	pitch
q	;	heat transfer (W)
t	;	temperature (°C).
W	;	air humidity ratio (kg_{agua}/k_{gas})

Φ	:	dry fin effectiveness (%)
Φ_w	:	fin effectiveness for wet tube (%)
δ_{fst}	:	frost layer thickness (mm)

Subíndices

1, i	in the entrance	o	total
2, f	in the exit	ow	wet total
ar	relative to the air	pi	internal wall
F	of fin	po	external wall
fst	of the frost	refr, r	of the refrigerant
m	average		

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REFERENCES

Carrier,

Gatica, J.A., 1999, Simulação de evaporadores de expansão direta. Tese de Mestrado, UFRJ Universidade Federal do Rio de Janeiro.

Mclinden, M.O., Klein, S.A.,Lemmon, E.W.,1998, NIST Thermodynamic and transport properties of refrigerants and refrigerant mixtures---REFPROP, NIST Standard Reference Database 23, version 6.0. (National Institute of Standards and Technology. U. S. Department of Commerce). Gaithersburg, Maryland 20899, USA.

McQuiston, F. C.,1978 , Correlation of heat, mass and momentum transport coefficients for plate-fin-tube heat transfer surfaces with staggered tubes", ASHRAE Transactions, v.84, part 1 , pp. 290-308.

Mcquiston, F. C.,1981 , Finned tube heat exchangers : state of the art for the air side, ASHRAE Transactions, v.87, part 1 .

Mcquiston, F. C.,Parker, J.D.,1994 , The NTU method, In: Cliff Robichaud (ed.), Heating Ventilating, and Air Conditioning. Analysis and Design, New York, USA. John Wiley & Sons, Inc.

Nusselt, W.,1916 , Surface condensation of water vapor, Part I e II

Oskarsson, S. P., Krakow, K. I., Lin, S.,1990, Evaporator models for operation with dry, wet, and frosted finned surfaces. Part I: heat transfer and fluid flow theory. ASHRAE Transactions, v. 96, part1

Oskarsson, S. P., Krakow, K. I., Lin, S.,1990. Evaporator models for operation with dry, wet, and frosted finned surfaces. Part II: evaporator models and verification. ASHRAE Transactions, v. 96, part1

Shah, R.K., 1981,

Threlkeld, J. L., 1962, Thermal Environmental Engineering, 1ª ed. Editora Prentice –Hall, Inc.Englewwod Cliffs, N.J..