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Abstract: This study attempts to establish a general method to analyze the performances of evaporator coils with complex tube circuitry using a distributed simulation model, which has three elements: branch, tube and control volume. The governing equations for a control volume are presented in the paper together with the computer simulation procedure for branches, tubes and control volumes of a coil. Using this model, the heat transfer and fluid flow characteristics of the coils are studied with R134a as a refrigerant. A guideline is proposed for the refrigerant circuitry arrangements in order to improve the coil comprehensive performance.

1. INTRODUCTION

The study of coil tube circuitry is complicated by a number of factors. One of the factors is that the flowing refrigerant inside the tube undergoes phase change, which results in significant variations in heat transfer and fluid flow characteristics along the coil tubes. Another factor is that there are numerous possible refrigerant circuitry arrangements for a coil, which contributes the complex for the general study. To study such a coil, a general simulation model which considers all the variations in the coils is required.

Fischer et al. (1983) proposed an evaporator model which divided coil into three regions. The average refrigerant-side HTC in two-phase region was obtained by integrating local values determined with correlation equation. Domanski (1991) used the tube-to-tube computation approach while developing an evaporator model to study the effect of nonuniform air distribution on the performance of the heat exchanger. Oskarsson et al. (1990) presented a finite element model to study the local behavior of heat transfer of fluids, as well as a three-region model and a parametric model. Recently, Judge et al. (1997), Liu (1996) and Ragazzi (1995) investigated the tube-finned evaporator coil using distributed-parameter evaporator models. However, all these existing evaporator simulation models only consider the coils with simple tube circuitry and do not study the effect of refrigerant circuitry. Ellison et al. (1981) presented a model for the condensers with specified refrigerant circuiting, which was based on a tube-to-tube computation approach. Up to now, to authors' knowledge, there is not an existing distributed simulation model for the study of tube circuitry of evaporator coils.

In the design of evaporator coils, increasing refrigerant mass velocity can enhance the heat transfer of an evaporator coil. However, this will increases the pressure drop in the refrigerant side and performances of the coil and system are adversely affected. An attempt is made to investigate the possibility of improving the coil performance by changing refrigerant mass velocity within a coil through proper refrigerant circuitry.

2. SIMULATION MODEL

In this paper, the tube circuitry without any refrigerant branch joining or splitting is referred to as a simple tube circuitry. The complex refrigerant circuitry has two branches joining to one or one branch splitting to two, which takes place in tube return bends. There are three elements in this model: branch, tube and control volume. The basic element for computation in the model is the control volume, shown in Figure 1.

2.1 Equations For A Control Volume

There are three heat transfer zones inside the tube corresponding to evaporation, transition and superheating of refrigerants, and there are two zones outside the tube due to unsaturated (dry) and saturated (wet) air. The driving potential for the simultaneous heat transfer in the air side is taken as the temperature difference between air around the tube and the saturated water film at the outer surface of the tube (Threlkeld ,1970).

2.1.1 Heat Balance On Air-Side

With the assumptions that the specific humidity fin efficiency is equal to the temperature fin efficiency and the condensed water temperature outside the tube is equal to the tube outer surface temperature, the heat balance equation for a control volume can be obtained:

$$\delta m_a h_{a2} = \delta m_a h_{a1} - \left[(\alpha_{sen} + \alpha_{lat}) \cdot (\delta A_t + \phi \cdot \delta A_f) \cdot (T_a - T_{t,o}) + C_{p,w} T_{t,o} \delta m_a (W_{a1} - W_{a2})\right] \tag{1a}$$

$$\alpha_{lat} = \frac{\alpha_{sen} \cdot i_{fg} \cdot C}{Le \cdot C_{p,a}} \tag{1b}$$

$$C = \frac{W_a - W_{s,t,o}}{T_a - T_{t,o}} \tag{1c}$$

where A is surface area, C_p isobaric specific heat, h enthalpy, i_{fg} latent heat of water condensation, Le Lewis number, m mass flow rate, T temperature, W air specific humidity, α_{lat} heat transfer coefficient due to moisture condensation, α_{sen} heat transfer coefficient due to temperature difference, δ incremental element for a control volume, and ϕ fin efficiency, subscripts a, f, o, s and t denote air, fin, outer tube, saturated moisture and tube respectively, subscripts 1 and 2 denote the inlet and outlet of a control volume respectively.

2.1.2 Mass Balance On Air-Side

The specific humidity at the exit is given by the following equation:

$$\delta m_a W_{a2} = \delta m_a W_{a1} - \frac{\alpha_{sen} \cdot C}{C_{p,a} \cdot Le} (\delta A_t + \phi \cdot \delta A_f) \cdot (T_a - T_{t,o}) \tag{2}$$

2.1.3 Heat Balance On Refrigerant-Side

Heat balance equation on the refrigerant-side is relatively simple, and it is:

$$\delta q = \delta m_r (h_{r2} - h_{r1}) \tag{3}$$

where q is heat transfer rate, and subscript r denotes refrigerant.

2.1.4 Heat Transfer Equation

To solve Equations (1) and (2), $T_{t,o}$ is determined firstly. The following heat transfer equation is used to determine α_c that is the heat transfer coefficient from the tube outer surface to the refrigerant:

$$\alpha_c \delta A_i (T_{t,o} - T_r) = \alpha_r \cdot \delta A_i (T_{t,i} - T_r) = \frac{\pi (d_o + d_i) \cdot k_t \cdot \delta L}{(d_o - d_i)} (T_{t,o} - T_{t,i}) \tag{4}$$

where d is diameter, k thermal conductivity, L tube length, and subscript i denotes inner tube. Then, the heat balance between the air side and the refrigerant side is used to determine $T_{t,o}$ with α_c:

$$(\alpha_{sen} + \alpha_{lat})(\delta A_t + \phi\ \delta A_f)(T_a - T_{t,o}) = \alpha_c \delta A_i (T_{t,o} - T_r) \tag{5a}$$

$$T_{t,o} = \frac{(\alpha_{sen} + \alpha_{lat})(\delta A_t + \phi \cdot \delta A_f) \cdot T_a + \alpha_c \delta A_i \cdot T_r}{(\alpha_{sen} + \alpha_{lat})(\delta A_t + \phi \cdot \delta A_f) + \alpha_c \delta A_i} \tag{5b}$$

2.1.5 Heat Transfer Coefficient

In the calculation of HTC in the air-side, correlation presented by McQuiston (1981) is used for the dry zone. The HTC on refrigerant-side depends on the flow conditions. In the region of evaporating two-phase flow, correlation proposed by Jung et al. (1991) is used.

2.1.6 Pressure Drop Of Refrigerant

The pressure drop on the refrigerant side for a control volume is computed as the sum of friction, momentum, and return bend pressure drops. For two phase flow, friction is calculated according to correlation recommended by Paliwada (1989) and the return bend pressure drop according to a two-phase multiplier provided by Paliwoda (1992).

2.2 Coil Simulation Procedure

In order to simulate a coil with complex tube circuitry, a special approach is proposed. Compared with the method by Ellison et al. (1981), this approach can be used to simulate a coil based on control volume.

A simplified branch network diagram is required first to trace the joining and branching of refrigerant flow. Although the tube circuitry may be complicated, the branch network usually is quite simple. In this way, the whole coil can be regarded as a network consisting of several separated branches. Six representative refrigerant circuitry configurations chosen for investigation are shown in Figure 2. Configuration A has only one simple branch. Configurations B, C, D, E and F use complex refrigerant circuitry, but the circuits are joined or branched in different points along the path. For example, configuration B has two branches with 7 tubes and one branch with 2 tubes. To simulate a coil, a hierarchical system is used, which follows the procedure given below:

a) First level: to determine the computation sequence for all branches in the coil and carry out computation branch-by-branch .
b) Second level: to determine the computation sequence for all tubes in the branch and carry out computation tube-by-tube .
c) Third level: to determine the computation sequence for all control volumes in the tube and carry out computation control volume-by-control volume.

A main problem in the first level is how to determine the inlet refrigerant properties of each branch. The approach by Ellison et al. (1981) is used where the refrigerant properties at the inlet of each branch are derived from those branches upstream. It is noted that the apportionment of refrigerant flow at a split is such that the downstream pressure drops of the two branches are balanced. Another problem is to specify the iteration sequence for all the branches. In order to develop a general program which can simulate the coils with all kinds of refrigerant branch networks, a two-direction adjacent matrix in the graphic theory is used here to describe the connection of refrigerant branches. So the program can automatically search the branch to be computed just according to the matrix.

In the second level, two arrays are introduced for each branch to indicate the address of each tube in the coil. For example, Nx1(I), Ny1(I) are introduced for Branch 1, where I indicates the tube sequence in which refrigerant reaches the tube in the branch. Nx1(I) is the count of the tube from one edge of the coil and Ny1(I) numbers the row of the tube in the coil. These two arrays can be capable of indicating the refrigerant flow sequence in the branch as well as the location of the tube in the coil. With these two arrays, the tubes in the branch are computed in the sequence of priority of refrigerant flow. In the third level, the computation sequence of control volumes in each tube is also along the refrigerant flow direction.

In the air side, air inlet properties are assumed initially. After every control volume is computed, the inlet air properties of each control volume are replaced by the upstream air properties. This procedure is repeated until the air-side properties of each control volume converge. For a coil with staggered tube arrangement, it is assumed that a given tube is exposed to the airstream that consists of 50% of the airstreams associated with the two closest neighbors in the proceeding row (Domanski 1991).

3. ANALYSIS

In this investigation, different refrigerant circuitry arrangements shown in Figure 2 are analyzed for coil design. Since changing refrigerant circuitry normally results in other changes, such as the change of coil pressure drop, and hence affect the whole refrigeration system. To isolate the effect to the evaporator coil, the outlet pressure of the coil (i.e. the inlet pressure of the compressor) and the inlet enthalpy (i.e. the outlet enthalpy of throttling device) are kept unchanged. The geometric parameters of the tubes and fins are given in Table 1. In this study, conditions at a humid environment is simulated. Design parameters used are: coil face velocity of 2.0 m/s, air inlet relative humidity of 90%, air inlet temperature of 30 °C, refrigerant inlet enthalpy is given by an assumed condenser outlet condition (condensing temperature of 40 °C; subcooling temperature difference of 5 °C); refrigerant superheating temperature difference of 5 °C, the refrigerant outlet pressure is given corresponding to a saturation temperature of 10 °C.

For a given refrigerant mass velocity, the refrigerant inlet temperature and coil length are iterated until the refrigerant outlet enthalpy and pressure converge. Then all the parameter distributions are finally obtained. This procedure is repeated with different refrigerant mass velocities for each configuration. Here, the coil average heat flux (total heat transfer rate divided by coil inner heat transfer area) is used as a criteria, whose value reflects the required heat transfer area for a design cooling load.

Geometric Parameters	Values	Geometric Parameters	Values
No. of tube rows in air direction	2	Inner tube diameter	8.53 mm
No. of tube rows perpendicular to air direction	8	Fin thickness	0.12 mm
Transverse tube spacing	25 mm	Fin density, fins/meter	394
Longitudinal tube spacing	21.6 mm	Fin type	plain plate
Outer tube diameter	9.53 mm		

Table 1 Coil geometric parameters

Figure 3 shows the coil average heat flux for different configurations. Firstly, it is seen that each configuration is appropriate for a specified range of refrigerant mass velocity. Here, the heat transfer mainly depends on two factors: the refrigerant-side HTC and temperature difference between refrigerant and air. With the increase of refrigerant mass velocity in the range, the coil average heat flux increases due to the increase of refrigerant-side HTC. When the mass velocity is relatively high, the refrigerant pressure drop is large and the refrigerant inlet pressure and temperature increase, which causes the significant decrease in temperature difference between refrigerant and air. Therefore, the coil average heat flux drops after it reaches a maximum. In coil design, in order to minimize the required heat transfer area, for each configuration the mass velocity corresponding to the maximum heat flux must be used.

Secondly, it is seen from Figure 3 that the maximum heat fluxes are different for each configuration. It means that using different refrigerant circuitry, the possible minimum heat transfer area of a coil for a design cooling load is different. In design of a bigger coil than the investigated, these refrigerant circuiting arrangements can be considered as the basic components of the coil and the above results may be used as a general guideline for optimizing coil by using refrigerant circuitry. When the coil cooling load is relatively small, the configuration (A) or (F) can be used. When the cooling load is large, the configuration (D) can be used and the required heat transfer area can be reduced by 5% compared to the simple configuration (A). Compared with configurations (A), (D) and (F), configuration (B), (C) and (E) are not recommended. Hence, this analysis provides a method to choose suitable refrigerant circuitry in coil design.

In order to highlight the general characteristics of air-cooling evaporator coils in heat transfer and fluid flow and better explain the above results, one of the computation cases for the coil with a simple tube circuitry shown in (A) of Figure 3 is chosen for analysis. Figure 4 shows the distribution of refrigerant-side heat transfer coefficient. It is to be noted that the relative length refers to the ratio of refrigerant flowing length measured from the inlet to the total path. It is seen that the refrigerant-side HTC varies greatly along the path. Figure 5 shows the distributions of thermal resistances of both sides. Although the air side HTC is much lower than the refrigerant side, the air side thermal resistance is greatly reduced due to the large finned area. It can be seen clearly that the thermal resistances are comparable for refrigerant side and air side, and the main thermal resistance exists in the refrigerant side in the superheating region. Figure 6 shows the distribution of pressure drop gradient. The sudden increases is due to return bends.

As far as refrigerant side is concerned, an ideal coil should have a high HTC with a low pressure drop. Both the HTC and pressure drop are closely related to the refrigerant mass velocity. It can be known from Figures 4 and 6 that the distribution shapes of pressure gradient and HTC of refrigerant side are basically similar. Both the pressure gradient and HTC are comparatively low in the low vapor quality two-phase region, where the HTC of refrigerant side can be increased by increasing refrigerant mass velocity with a limited increase of pressure drop. In the high vapor quality two-phase flow region, the HTC and pressure drop gradient are much higher than those in other regions. Thus, refrigerant mass velocity in this region should not be too high. Therefore, varying the refrigerant mass velocity in different regions is a reasonable approach to balance the increases of refrigerant-side HTC and pressure drop. This explains why using a suitable complex refrigerant circuitry can improve the coil performance.

4. CONCLUSIONS

To evaluate the performance of a coil, the coil average heat flux and the refrigerant-side pressure loss are two of the most important factors that need to be considered. The study shows that for a simple circuit the distribution shapes of pressure drop gradient and HTC of refrigerant side are basically similar. Both the pressure drop gradient and HTC are comparatively low in superheating region as well as in low vapor quality two-phase region. In the high vapor quality two-phase flow region, the HTC and pressure drop gradient are much higher than those in other regions. Therefore, varying the refrigerant mass velocity in different regions while the refrigerant goes through the path would be a reasonable approach to balance the increases of HTC and pressure drop. A convenient way to change the refrigerant mass velocity along the path is to join or split the refrigerant circuits in the proper return bends. Properly selecting a refrigerant circuitry is a necessary and effective approach to improve the evaporator coil comprehensive performance.

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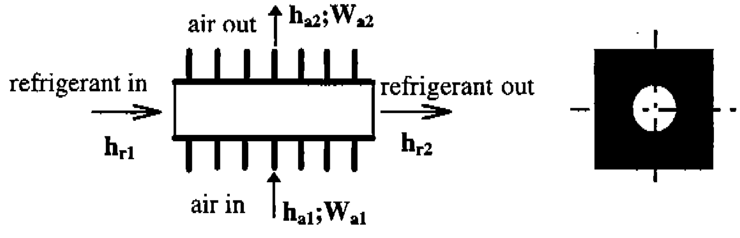

Figure 1 A control volume along a tube with fins

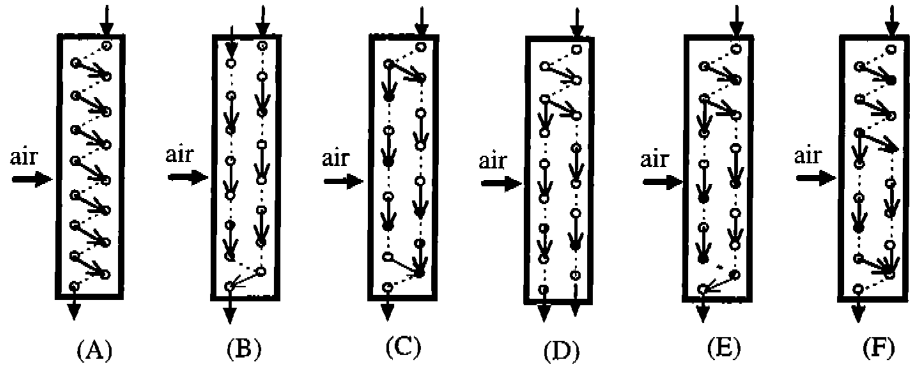

Figure 2 Schematics of return bends of six coil configurations with 16 tubes each

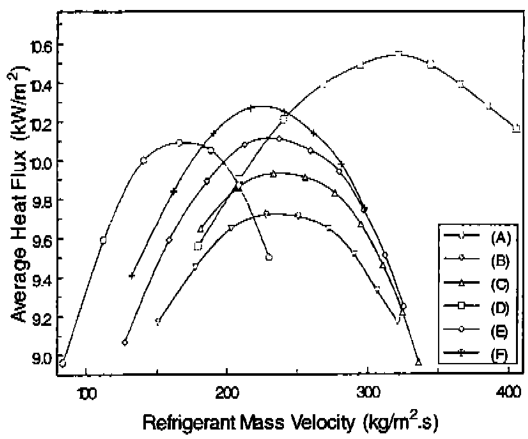

Figure 3 Average heat flux with refrigerant mass velocity for different configurations

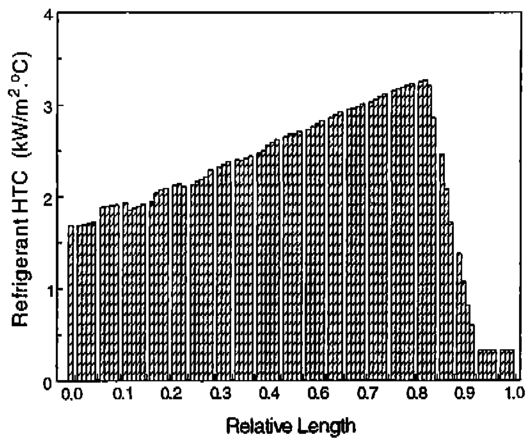

Figure 4 Refrigerant HTC distribution

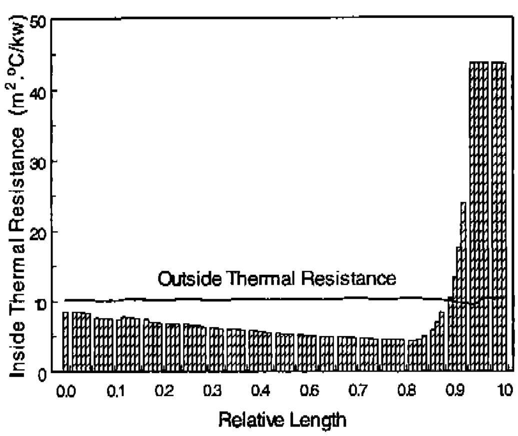

Figure 5 Inside and outside thermal resistance distributions

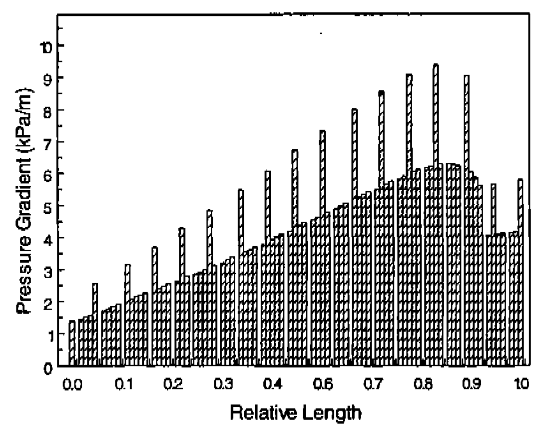

Figure 6 Refrigerant pressure gradient distribution