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A COMPUTER MODEL FOR AIR-COOLED REFRIGERANT CONDENSERS WITH SPECIFIED REFRIGERANT CIRCUITING

by

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ABSTRACT

A computer model for an air-cooled refrigerant condenser is presented; the model is intended for use in detailed design analyses or in simulation of the performance of existing heat exchangers that have complex refrigerant circuiting or unusual air-side geometries. The model relies on a tube-by-tube computational approach calculating the thermal and fluid-flow performance of each tube in the heat exchanger individually, using local temperatures and heat transfer coefficients. The refrigerant circuiting must be specified; the joining or branching of parallel circuits is accommodated using appropriate mixing expressions. Air-side heat exchange correlations may be specified so that various surface geometries can be investigated. Results of the analyses of two condensers are compared to experiment.

INTRODUCTION

The authors have been involved in the development of a heat pump computer model which is useful in the design and performance analysis of conventional electric heat pumps. The present version of this model(1) employs computational routines for the heat exchangers based on the effectiveness vs N_{tu} analytical approach of Kays and London(2). This treatment of the heat exchangers has proved to be adequate for most uses of the model.

The effectiveness vs N_{tu} approach, however, can rigorously accommodate only straight, equivalent, unbranched flow circuits. In contrast, the refrigerant-side circuiting of refrigerant condensers and evaporators is often neither straight or unbranched, nor are parallel circuits equivalent. In fact, the circuiting is often intricate. Accordingly, a decision was made to attempt to formulate a heat exchanger computer model that would rigorously treat complex refrigerant-side circuiting arrangements. The condenser version of the resulting model is described in this paper.

An evaporator version is also being developed by the authors; the evaporator model is necessarily more complicated and has not been adequately developed in all respects as of this writing.

The more rigorous models may replace simpler heat exchanger routines in a system simulation model. The cost of the greater detail and accuracy is, of course, increased

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computer running time. The models may also be used by themselves for the performance analysis of heat exchangers; the authors hope that they will prove to be a useful heat exchanger design tool.

GENERAL DESCRIPTION OF THE MODEL

This new model for an air-cooled refrigerant condenser is based on a tube-by-tube computation approach. The thermal and fluid-flow performance of each tube (the length of tube between two return bends in the conventional geometry) is computed individually, based on local temperatures and heat transfer coefficients. Equations due to Traviss, Baron, and Rohsenow(3) are used to calculate two-phase refrigerant-side heat transfer coefficients, and the methods of Lockhart and Martinelli(4) and Pierre(5) are used for pressure drop. Air-side heat transfer and pressure drop are calculated using correlations of the Colburn "j" factor and the Fanning friction factor with the Reynolds number. Heat flux is then found by applying the Kays and London(2) effectiveness vs N_{tu} (number of transfer units) formulation for crossflow heat exchangers with one fluid mixed for each individual tube. The use of existing computer routines for calculation of refrigerant properties requires that the model be coded in English units.

In order to accomodate complex refrigerant circuiting, a table of tube connections is required. The program uses this table to trace the flow of refrigerant in each circuit, taking proper account of the joining or branching of parallel circuits, and using appropriate mixing expressions. Thus the refrigerant properties at the inlet to each tube are those at the outlet of the tube immediately upstream from it, or derived from those of two tubes upstream. The outflow from a tube may be routed to a single tube downstream or split between two tubes. Apportionment of refrigerant flow at a split is such that the downstream pressure drops of the two branches will be balanced; it is assumed, but as yet unverified that the vapor quality is the same in both branches. Performance of the heat exchanger taken as a whole is calculated as the aggregate of the individual-tube performances.

Though not always necessary, it is often convenient to display the results of the calculations in graphical form. In order to make best use of the graphics capabilities of many computers, performance parameters for each tube can be saved for transmission to a plotting program. The graphs shown in a subsequent section of this paper were produced in this fashion. Summary tables of selected performance parameters for each tube may be printed. If desired, circuit tracing routines may be invoked so that tables may be produced for each refrigerant circuit, with the entries arranged in the order that refrigerant reaches them.

INDIVIDUAL-TUBE HEAT TRANSFER AND PRESSURE DROP CALCULATIONS

Computational routines were formulated for the purpose of modeling the thermal performance and pressure losses of individual tubes. These are:

- Refrigerant condensing heat transfer coefficient and pressure drop;
- Air-side heat transfer coefficient and pressure drop;
- Effectiveness and heat flux for a condenser tube.

In the following paragraphs, a description of the basic relationships used for the above routines is presented.

Condensing Heat Transfer Coefficient and Pressure Drop

The correlations developed by Traviss, Baron and Rohsenow(3) for condensing heat transfer coefficients are based on an analytical derivation assuming annular flow in a tube with the von Karman universal velocity distribution. Experimental data obtained using R-12 and R-22 were found to be in good agreement with the analytical prediction, except that it was necessary to introduce a correction for flow conditions under which droplet entrainment occurs. The computations proceed as follows.

An average vapor quality, x_m, is estimated and used to compute the Lockhart-Martinelli parameter(4) for the tube:*

*See Appendix A for explanation of nomenclature.

$$X_{tt} = \left(\frac{1 - x_m}{x_m}\right)^{0.9} \left(\frac{\rho_v}{\rho_\ell}\right)^{0.5} \left(\frac{\mu_\ell}{\mu_v}\right)^{0.1} . \tag{1}$$

A Reynolds number for the liquid phase is then computed:

$$N_{Re,tp,\ell} = G_r(1 - x_m)D_e/\mu_\ell . \tag{2}$$

A factor, F_2, is calculated by the appropriate one of the following expressions:

$$F_2 = 0.707N_{Pr,\ell}N_{Re,tp,\ell}^{0.5} ; \qquad (N_{Re,tp,\ell} < 50) \tag{3}$$

$$F_2 = 5N_{Pr,\ell} + 5\ln[1 + N_{Pr,\ell}(0.09636N_{Re,tp,\ell}^{0.585} - 1)] ; \qquad (50 < N_{Re,tp,\ell} < 1125) \tag{4}$$

$$F_2 = 5N_{Pr,\ell} + 5\ln(1 + 5N_{Pr,\ell}) + 2.5\ln(0.0031N_{Re,tp,\ell}^{0.812}) . \qquad (N_{Re,tp,\ell} > 1125) \tag{5}$$

Another factor, $F(X_{tt})$, is computed by:

$$F(X_{tt}) = \frac{0.15}{X_{tt}}(1 + 2.85X_{tt}^{0.524}) . \tag{6}$$

Finally, the condensing heat transfer coefficient is computed by the appropriate one of the following expressions:

$$h_{tp} = \frac{k_\ell F(X_{tt})N_{Pr,\ell}(N_{Re,tp,\ell})^{0.9}}{D_e F_2} . \qquad [F(X_{tt}) < 1] \tag{7}$$

$$h_{tp} = \frac{k_\ell [F(X_{tt})]^{1.15}N_{Pr,\ell}(N_{Re,tp,\ell})^{0.9}}{D_e F_2} . \qquad [1 < F(X_{tt}) < 15] \tag{8}$$

The above expressions are used for the vapor quality range $0.1 < x < 0.95$. Conventional single-phase heat transfer relationships for turbulent flow in circular tubes are used for the liquid and superheated vapor regions. Between these regions, (i.e., $0 \leq x \leq 0.1$ or $0.95 \leq x \leq 1.0$) the coefficient is computed by linear interpolation. In the superheated vapor region, the tube wall temperature may sometimes fall to or below the refrigerant saturation temperature while the bulk temperature of the refrigerant remains above the saturation temperature. In this case, the heat transfer coefficient is computed as though the refrigerant were in two-phase flow with a vapor quality of 0.99. This expedient accomodates the onset of two-phase flow for a region in which specific correlations are not available.

Friction pressure drop in the straight sections of the tubes is calculated by the method of Lockhart and Martinelli.(4) The friction factor is computed by the curve fit,

$$f = 0.049N_{Re,tp,\ell}^{-0.2} . \tag{9}$$

Pressure drop is calculated by the following expression,

$$\Delta p_{tp} = \phi_\ell^2 \Delta p_{x,\ell} \ , \tag{10}$$

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where ϕ_ℓ is defined by: $\phi_\ell = 1.467 - 0.51346 \ \ell n \ X_{tt} + 0.048789(\ell n X_{tt})^2$ and $\Delta p_{x,\ell}$ is computed by:

$$\Delta p_{x,\ell} = \frac{2fG_\ell^2}{\rho_e g_c} \cdot \frac{L}{D_e} \ . \tag{11}$$

In the above expression, the mass velocity of the liquid, G_ℓ, is given by:

$$G_\ell = (1 - x_m)G_r \ . \tag{12}$$

The pressure drop in the return bends is computed by the method of Pierre,(5) as follows:

$$\rho_m = 1/\left[\frac{x_m}{\rho_v} + \frac{1 - x_m}{\rho_\ell}\right] \ ; \tag{13}$$

$$\Delta p_{rb} = \frac{\zeta G_r^2}{2g_c\rho_m} \ ; \tag{14}$$

where ζ is a function of x_m, as follows:

x_m	ζ
0	0.7
0.2	0.8
0.4	1.0
0.6	1.1
0.8	1.0
1.0	0.6

Momentum pressure gain in the condenser is neglected; accordingly, the total pressure drop is taken as the sum of the straight-tube and return-bend pressure drops.

Air-Side Heat Transfer and Pressure Drop

For computing air-side heat transfer coefficient and pressure drop, curve fits for friction factor and "j" factor of the following form are used:

$$f = C_f(N_{re})^{n_f} \ ; \tag{15}$$

$$j = C_j(N_{Re})^{n_j} \ . \tag{16}$$

As necessary, the curve fits may be piecewise continuous. For specific air-side geometries the values for C_f, C_j, n_f, n_j, and the Reynolds number range over which the constants are valid, can be specified as input to the program; otherwise, default values, suitable for conventional tube-and-plate-fin geometry, are provided by the program.

Computations proceed as follows:

$$N_{Re} = 4G_a r_h/\mu_a \; ; \tag{17}$$

obtain C_f, n_f, C_j, n_j, for N_{Re} ;

$$j = C_j (N_{Re})^{n_j} \; ; \tag{18}$$

$$f = C_f (N_{Re})^{n_f} \; ; \tag{19}$$

$$h = jG_a c_{pa}/N_{Pr}^{2/3} \; ; \tag{20}$$

$$\Delta p = \frac{fG_a^2}{2g_c\rho_a} \cdot \frac{L}{r_h} \; . \tag{21}$$

While it is not done in the present version of the model, it may be appropriate in the future to account for variations in the values of f and h by tube row, as described by Rich(6).

Effectiveness and Heat Flux for a Condenser Tube

For computation of the thermal performance of individual condenser tubes, effectiveness-N_{tu} relationships for a crossflow heat exchanger with one fluid mixed, as presented by Kays and London,(2) are used. Values for air-side and refrigerant-side heat transfer coefficients and air-side finned-surface effectiveness are obtained from subroutines, and computation then proceeds as follows:

$$\frac{1}{UA} = \frac{1}{h_a A_{s,a} \eta_s} + \frac{1}{h_r A_{s,r}} \; ; \text{ (metal tube wall resistance is neglected)} \tag{22}$$

$$C_a = \dot{m}_a c_{p,a} ; \tag{23}$$

$$C_r = \dot{m}_r c_{p,r} \; ; \text{ (single-phase only)} \tag{24}$$

$$C_{min} = \text{smaller of } C_a \text{ and } C_r \; ; \tag{25}$$

$$N_{tu} = UA/C_{min} \; . \tag{26}$$

The tube effectivess is then computed for the two-phase region as:

$$E = 1 - e^{-N_{tu}} \tag{27}$$

For the single-phase regions the calculation depends on whether capacity rate C for the unmixed fluid is greater or less than C for the mixed fluid. In this application, air is the unmixed fluid and refrigerant is mixed. Thus for the case that $C_{unmixed} > C_{mixed}$ (i.e., $C_a > C_r$)

$$\Gamma = 1 - e^{-N_{tu}C_r/C_a} ; \tag{28}$$

$$E = 1 - e^{-\Gamma C_a/C_r} ; \tag{29}$$

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and if $C_{mixed} > C_{unmixed}$ $(C_r > C_a)$

$$\Gamma = 1 - e^{-N_{tu}} ; \tag{30}$$

$$E = (C_r/C_a)(1 - e^{-\Gamma C_a/C_r}) . \tag{31}$$

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The heat flux, air temperature change, and refrigerant property change are then computed by:

$$Q = EC_{min}(T_{r,in} - T_{a,in}) ; \tag{32}$$

$$T_{a,out} = T_{a,in} + Q/C_a ; \tag{33}$$

$$x_e = x_i - Q/\dot{m}_r h_{fg} ; \text{ (two-phase only)} \tag{34}$$

$$T_{r,out} = T_{r,in} - Q/C_r . \text{ (single-phase only)} \tag{35}$$

For the two-phase region, $T_{r,out}$ is taken as the saturation temperature at the exit pressure. In using the last two equations above, a test is performed each time to see whether the refrigerant is moving into or out of a single-phase region. Finally, the average wall temperature is calculated:

$$T_w = (T_{r,in} + T_{r,out})/2 - Q/h_r A_{s,r} . \tag{36}$$

Specification of Tube Circuits

In order to calculate the performance of tubes in the sequence in which refrigerant reaches them, a table of tube connections is used to trace the path of refrigerant flow. Usual construction methods for compact heat exchangers (i.e., circuits branch or split at preformed return bends) require that a tube receive refrigerant directly from no more than two tubes, and in turn deliver it to no more than two tubes downstream. It is, then, sufficient to specify at most four tubes in the heat exchanger that may connect to a tube in question. It is convenient to consider the assembly as consisting of layers or rows of tubes, with each row being perpendicular to the direction of airflow. A tube may then be referenced by its position in a row and the row number.

If each tube is considered to have two position indices (or coordinates), I and J, where I is the count of tubes from one edge of the heat exchanger and J numbers the row of tubes, we may associate elements of four 3-dimensional arrays with each tube in order to store the indices of tubes that feed or receive refrigerant to or from tube I,J. The arrays INI(I,J,K) and INJ(I,J,K), K = 1 or 2, will contain indices of tubes that supply refrigerant to tube I,J; OUTI(I,J,K) and OUTJ(I,J,K) contain the addresses of tubes that may receive the outflows. In order to keep the input of this table of tube connections straightforward, indices of all the tubes that connect to tube I,J are read on one data input card, in the order:

```
INI(I,J,1), INJ(I,J,1): INI(I,J,2), INJ(I,J,2);
OUTI(I,J,1), OUTJ(I,J,1); OUTI(I,J,2), OUTJ(I,J,2) .
```

The simple arrangement of tubes shown in Fig. 1, may be used to illustrate a table of tube connections. The diagram shows an end view of the heat exchanger with the tubes represented by circles, the return bends at the near end of the tubes by solid lines, and those at the far end by dashed lines. For this rather unlikely arrangement, the table of tube connections would be:

Table 1. Tube Connections

I	J	INI (I,J,1)	INJ (I,J,1)	INI (I,J,2)	INJ (I,J,2)	OUTI (I,J,1)	OUTJ (I,J,1)	OUTI (I,J,2)	OUTJ (I,J,2)
1	1	−1	−1	0	0	2	1	1	2
2	1	1	1	0	0	3	1	0	0
3	1	2	1	0	0	3	2	0	0
1	2	1	1	0	0	2	2	0	0
2	2	1	2	0	0	3	2	0	0
3	2	3	1	2	2	99	99	0	0

Connections to the inlet and outlet manifolds of the heat exchanger are denoted by −1,−1 and by 99,99 respectively; zero entries in Table 1 indicate no connection. There may be multiple entries or exits to or from the heat exchanger. Note that if there is only one connection at entry or exit to a particular tube, it must be entered in the table so that its third index is 1, not 2. "Missing" tubes, if any, must be represented in the table by all zero entries; all rows must be considered to have the same number of tubes, including the "missing" ones.

Examples of the table of tube connections for real condensers, such as those analyzed later in this paper, are too lengthy to include here. They may currently be obtained from the authors.

Estimation of Refrigerant Flow Distribution

Before calculations of performance can proceed, it is necessary to apportion the flow of refrigerant through each path through the heat exchanger. In general, the flow in any tube is the same as that in the tube upstream, or the sum of the flows in two upstream tubes. Exceptions are tubes connected to the inlet header of the heat exchanger or the two tubes immediately downstream of a branch point. For these cases, the incoming flow is apportioned inversely to the flow resistance seen in each circuit downstream of the header or branch point. The flow resistance for a single tube is calculated as $(\Delta p)^{0.5}/\dot{m}_r$; the resistance of a circuit is calculated by summing the resistances of those tubes connected in series, and the inverses for tubes with parallel flows. Initial estimates for Δp and $\dot{m}_r$, and the refinement procedure, are discussed below.

SEQUENCE OF CALCULATIONS

It may be expected that changes in thermodynamic properties of the refrigerant will be much greater than those for the air as both fluids move through the heat exchanger. Accordingly, the assembly of tubes into a heat exchanger is treated as viewed from the refrigerant side. Individual tubes are modeled in the sequence in which refrigerant, not air, reaches each tube so that calculational instabilites may be reduced. A consequence of this choice, since the two fluids are in crossflow, is that the air temperature at a particular tube may not be known accurately when the refrigerant-side heat exchange is calculated for that tube. It is, then, necessary to calculate performance of the heat exchanger within an iterative loop over the entire assembly of tubes so that the air (as well as the refrigerant) inlet conditions at each tube will eventually be known with sufficient accuracy as the tube is modeled.

The executive computer routine for the condenser model, subroutine COND, handles the task of refrigerant circuit tracing, using the tables of tube connections to select individual tubes to model in the order in which refrigerant reaches them. The bookkeeping chores and tests for convergence are also assigned to this routine. It is called from MAIN, the first routine of the program to be executed, which supplies, mostly through common statements, the thermodynamic properties and mass flow rates of the air and refrigerant at inlet to the heat exchanger. The geometric parameters and tables of tube connections are supplied, again by means of common statements, from data input statements in MAIN. The input parameters for running the program are described in Appendix B.

Subprograms available to the executive routine COND are listed below. They are straightforward applications of the principles stated earlier.

CTUBE	models thermal performance of one condenser tube
CRSID	calculates refrigerant-side heat transfer coefficient and pressure drop
AIRSID	calculates air-side heat transfer coefficient and pressure drop
FINEFC	calculates fin effectiveness
HTEXEF	calculates heat exchanger effectiveness
FLUPRP	calculates conductivity, viscosity, density, specific heat, and Prandtl number for liquid or vapor refrigerant
TLIQ	calculates temperature of saturated liquid refrigerant from enthalpy of saturated liquid
MIXER	performs mixing calculation of refrigerant enthalpy, temperature, and quality at confluence of streams
GEOM	calculates geometric factors (surface and free-flow areas, hydraulic radius)
SPLIT	calculates downsteam resistance seen from each tube and estimates refrigerant flow rate in tubes

Thermodynamic properties at saturation and for superheated vapor refrigerant are calculated using subprograms written by Hiller and Glicksman.(7) These routines are written for R-12, R-22, and R-502; however, the fluid transport properties are available for R-22 only.

Subroutine COND, the executive subprogram, first calls subroutine GEOM to calculate geometric factors such as free-flow and surface areas, and then proceeds to set the initial air and refrigerant conditions for each tube, as described below, making use of subroutines FLUPRP and CTUBE. Having accomplished these preliminary tasks, COND is ready to begin the calculation of heat exchanger performance, reiterating the process until the calculated performance of each tube of the assembly is stable. The sequence of calculations is shown in Fig. 2.

The loop seeking convergence of performance parameters (remote connections α in Fig. 2) begins with a call to subroutine SPLIT which calculates the refrigerant flow in each tube connected directly to the inlet header. This flow rate is established using the downstream resistance estimated from the pressure drop and mass flow rate calculated in a previous pass through the loop or, the first time, in the initializing routines. SPLIT also handles the task of apportioning the refrigerant flow at downstream branch points according to the resistance seen in each downstream path.

On return from SPLIT, a loop over all circuits in the heat exchanger (remote connections β) can begin: one of the several tubes connected directly to the inlet header is selected and its performance is modeled by a call to subroutine CTUBE. Performance parameters of this tube are stored, and a test is made to see if the tube just modeled was the last one

in a branch circuit, if a new branch point is encountered, or if flow progresses to a single tube downstream. The indices of the next tube to be modeled are found from the table of tube connections, and the refrigerant flow rate for the new tube is set accordingly. Next follows a test for a joining of circuits (confluence) at the inlet to the new tube. If there is no joining, refrigerant conditions are taken to be the same as those at the outlet of the preceeding tube; inlet air conditions are the outlet air conditions of the corresponding tube in the preceeding row.

If there is a confluence at the inlet to the new tube, the flow rate through it is calculated as the sum of flows in the (two) contributing tubes. The inlet refrigerant temperature and quality are calculated from the average of the exit enthalpies weighted by the mass flow rates in the contributing tubes. Inlet pressure is taken to be the average of the exit pressures of the two upstream tubes; if these pressures differ by more than 5%, an error message is printed.

Using this scheme for passing the calculated outlet refrigerant properties from one tube to serve as the inlet properties for the next (or a contributor to them), we may follow the path of refrigerant through a circuit, modeling the performance of each tube as it is encountered, by means of calls to CTUBE, until we reach the discharge manifold. At this point in the calculation, it is necessary to return to the first of any branch points in the circuit just traced and to model tubes from that point until the discharge manifold is again reached. After all branches have been modeled, the process is repeated starting with another tube that is connected directly to the inlet header. It should be noted that the tubes downstream of a confluence may be modeled more than once. This is inefficient, but avoids programming complications; it presents no other problems since the last pass through a tube is the desired calculation, and the only one that is saved. After all the tubes have been modeled, the convergence of the process is checked, by comparing the exit air and refrigerant temperatures at each tube with those from the previous iteration. A significant change in any of these temperatures calls for another iteration of the heat exchanger performance calculations. When convergence has been reached, the heating capacity of the condenser is found from the sum of the capacities of the individual tubes; thermodynamic properties of the air and refrigerant at exit from the heat exchanger are the averages of those of the several contributing streams weighted by the mass flow rates of those contributors.

Initial Values for Pressure Drops

Since each tube is being modeled separately, it is necessary to set initial conditions at each tube for the first calculational pass through the heat exchanger. These conditions of air and refrigerant temperature, pressure, quality, and mass flow rate must be estimated from those at the inlet to the assembly. For the first iteration, all circuits are treated as having equal refrigerant flow rates, and inlet air and refrigerant conditions for individual tubes are taken to be the same for all tubes in one row. The tube nearest the center of the first row to see the incoming air is modeled as though it were in two-phase flow with refrigerant pressure taken to be that at the inlet header, and temperature equal to the saturation temperature at the header. All other tubes in that row are then assigned performance parameters equal to those of the central tube. Inlet air and refrigerant conditions for tubes in the subsequent row in the direction of airflow are assigned values equal to the outlet conditions of the preceding row. The process of modeling just one tube in a row is repeated for each subsequent row in order to obtain somewhat realistic starting values for the pressure drop in each tube, and thus the resistance to flow. The first iteration can then proceed as though there has been a previous pass through the loops and all variables are defined.

Display of the Results

The calculated performance parameters for each tube are stored in an array so that they may be retained for later use by the program that produces graphs, such as those shown later in this paper. As desired, the plots may show selected values for tubes in one refrigerant circuit, or for all of the tubes. In the latter case, the points for tubes in one row may be connected by line segments; alternatively, refrigerant circuits may be traced. Quantities available for plotting or tabular display include the following for each tube:

- heat flux,
- tube wall temperature, both observed and calculated,
- refrigerant pressure drop,
- air and refrigerant mass flow rate,
- mass and volume of vapor and liquid,
- resistance to refrigerant flow expressed as $(\Delta p)^{0.5}/\dot{m}_r$;

and at the inlet and outlet of each tube (for the air side this means before and after passing over the tube):

- air and refrigerant temperatures,
- refrigerant vapor quality,
- refrigerant pressure.

VERIFICATION OF THE MODEL

The condenser model was run using the geometric descriptions and the tables of tube connections for two tube-and-sheet-fin heat exchangers which had been tested in our laboratory. The values predicted by the model for tube wall temperatures and heat exchanger capacity are compared to those measured in the laboratory in order to verify the accuracy of the model; other calculated quantities are shown in some of the figures to illustrate the model's usefulness.

Analysis of a Condenser with Simple Circuitry

Results for a heat exchanger with simple circuitry are presented first. This condenser has 72 tubes arranged in 3 rows of 24 tubes each, and 3 parallel refrigerant circuits. Refrigerant (R-22) entering the front row of tubes (the side where air enters the heat exchanger) is switched to the rear row at tube 13, and that from the rear row is brought to the front; one refrigerant circuit lies entirely in the middle row of tubes.

The points plotted in Fig. 3 represent the calculated temperature of air as it leaves each tube. The lines connecting these points have no physical significance, as viewed from airside considerations, but rather illustrate the way that tubes are connected by return bends. The region of the condenser containing superheated refrigerant is seen to be small, occupying no more than three tubes in any of the circuits. As expected, the air temperatures are nearly constant over the two-phase region beginning at about tube 3. The subcooled region (revealed by plots of the refrigerant temperature not shown here) begins at tube 12 for path 1, which begins in the front row, tube 15 for path 2, in the middle row, and tube 16 for path 3. The reduction in air temperatures in the subcooled region reflects the drop in temperature of the refrigerant as it proceeds through the subcooled region to the exit. For lack of experimental data, the velocity of air reaching the condenser was considered to be uniform across the entire heat exchanger face.

The refrigerant, tube wall, and air temperatures for tubes in path 2 (which lies in the center row of tubes) are shown in Fig. 4. The superheated, two-phase, and subcooled regions are clearly seen in the top curve, that for calculated refrigerant temperature. The calculated tube wall temperatures are compared to the measured values in the middle curve. Agreement is generally good, except near the end of the subcooled region. The calculated heat exchanger capacity of 30,800 Btu/h (9.02 kW) compares quite well with the 31,200 Btu/h (9.14 kW) observed.

The heat flux at each tube is shown in Fig. 5, where the points are connected so as to show the path of the refrigerant circuits. Again, the superheated and two phase regions are easily seen, and the heat transfer rate there varies in the expected manner. As could be seen in the previous figures, the subcooled region is large; the condenser is operating almost half "flooded". The consequences of this flooding are easily seen in the rapid decline of heat flux for tubes in the subcooled region.

Analysis of a Condenser with Complex Circuitry

A second tube-and-sheet-fin heat exchanger, one with complex refrigerant circuiting, was simulated to provide a more severe test of the condenser model. A diagram of the arrangement of return bends, from which the circuits can be traced, is shown in Fig. 6. The dashed lines represent the return bends at the far ends of the tubes, and the "x"'s show the location of "missing" tubes. There are 23 tubes in the first row (where air enters the heat exchanger), 25 in the second row, and 24 in the third. The refrigerant used in this heat exchanger is R-22.

In Fig. 7, the temperatures for air leaving each of the tubes in this more complex heat exchanger are shown. As in Fig. 3, the lines connecting the points have no physical significance as viewed from the air side, but show instead the refrigerant circuiting; the symbols indicate in which row a tube lies. The low temperatures for the first tubes in rows 2 and 3 show the effect of a "missing" tube in the first row. A similar but less pronounced effect can be seen at the other end of the heat exchanger.

The temperature of the air leaving the tubes where refrigerant enters the heat exchanger (tubes 9 through 13 in the third row, i.e., tubes 9,3 through 13,3) is high because these tubes contain superheated vapor. The variations in the air temperatures leaving these five tubes may be explained by a combination of causes. Each of the five circuits has a different refrigerant flow rate, with tubes 10,3 and 11,3 having the highest rates and tubes 9,3 and 13,3 having the lowest. On the air side, the velocity profile of air entering the heat exchanger is such that the mass flow rate of air at tube 13,3 is some 18% higher than that at tube 9,3.

The drop in air temperature that starts after tube 17,1 is due to the drop in refrigerant temperature as it enters the sub-cooled region. This temperature reduction is reflected in the corresponding tubes in rows 2 and 3 even though those tubes contain two-phase refrigerant. The effects of confluences in the refrigerant streams are clearly seen in the region of tube 20.

Refrigerant, tube wall, and air temperatures for tubes lying in the circuit that begins at tube 11,3 are shown in Fig. 8. The observed wall temperatures, measured at the return bends, are shown for comparison. Again, the superheated and two-phase regions are clearly discernable. The onset of the subcooled region is seen at tube 17,1; the effect of a confluence may be seen at tube 20,1.

The calculated and observed tube wall temperatures are in good agreement except in the subcooled region. The cause of this discrepancy [as much as 7 F (3.9°C)] has not been determined as of this writing; in some of the other circuits, the observed wall temperature in this region is as much as 8 F (4.4°C) higher than the calculated values. Calculated heat exchanger capacity is 34,700 Btu/h (10.2 kW), which agrees quite well with the 34,500 Btu/hr (10.1 kW) observed in the laboratory after correction for the heat added by the blower motor. The agreement with observed performance is judged sufficiently close to make the model useful in design and simulation applications.

CONCLUSIONS

The approach used in this model, that of calculating the performance of each tube individually in an iterative procedure, would be virtually unmanageable without the aid of a fast computer — even for a modest sized coil. In fact, it seemed uncertain, at the outset of the effort, that the process would prove to be practical even with large computers. However, the uses of the model, described herein demonstrate the feasibility of the approach. It works, and does so without using up much computer time or space. Reasonably tight convergence limits are typically achieved in 3 to 7 cpu sec on our IBM 3033 computer, using less than 35K words of memory.

Tests of the model have, thus far, revealed that calculated quantities agree quite well with condenser performance parameters for which we have values measured in the laboratory. Armed with the confidence thus gained, the authors feel that they can recommend the use of the model for its intended purposes: to calculate condenser performance at a level of detail that will be useful in the design of more efficient heat exchangers, or alternatively, in application studies of existing heat exchangers that employ complex refrigerant circuiting or unusual air-side geometries.

The work described herein is in the public domain, thus the computer program is currently available to prospective users for execution on their own computers.

APPENDIX A

Nomenclature

Variables

A_s	surface area
c_p	specific heat at constant pressure
C	heat capacity rate, $\dot{m}c_p$
C_f, C_j	correlation coefficients for friction factor and heat transfer coefficient respectively
D_e	equivalent diameter
E	effectiveness
f	friction factor
F_2	computed factor in condensing coefficient correlation, defined in text
$F(X_{tt})$	function of X_{tt}, defined in text
g_c	gravitational constant
G	mass velocity or mass flux (mass flow rate per unit of free-flow area)
h	heat transfer coefficient
h_{fg}	heat of vaporization
j	Colburn "j" factor: (Stanton number)·(Prandtl number)$^{2/3}$
k	thermal conductivity
L	tube length
$\dot{m}$	mass flow rate
n_f, n_j	correlation exponents for friction factor and heat transfer
N_{Pr}	Prandtl number
N_{Re}	Reynolds number
N_{tu}	number of transfer units
Q	heat flow rate
r_h	hydraulic radius
UA	product of effective overall heat transfer coefficient and surface area
W	specific humidity
x	vapor quality
X_{tt}	Martinelli-Lockhart parameter for two-phase flow with both phases turbulent, defined in text

Γ	factor in computation of effectiveness
Δp	pressure drop
Δp_{rb}	return-bend pressure drop
Δp_x	friction pressure drop in tube
Δx	change in vapor quality
ζ	return-bend loss coefficient
η	extended surface effectiveness
μ	viscosity
ρ	density
ϕ	function of X_{tt}, used in computation of two-phase pressure drop

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General Subscripts

a	air or air-side
e	exit
est	estimated
i	inlet
in	inlet
ℓ	liquid
m	mean
min	minimum
out	outlet
r	refrigerant or refrigerant-side
tp	two-phase
v	vapor
w	wall surface

APPENDIX B

Input Parameters

The tube-by-tube model of a condenser requires, as input, the refrigerant and air temperatures and pressures at entry to the heat exchanger, refrigerant mass flow rate and volumetric airflow rate. Provision is made for entry of a velocity profile of air reaching the heat exchanger if such is available; otherwise, the velocity is assumed uniform. A geometric description of the tubes and fins is required, as well as a table of tube connections. Details of the input parameters are given in Table B.1.

The computer program is written in FORTRAN IV and requires 140 K bytes (35 K words) of memory. Typical running times on an IBM 3033 computer have ranged from 3 to 7 CPU sec.

Table B.1. Operating and Geometric Input Parameters

Card 1	TAIIC	Entering air temp	F
	PA	Atmos. pressure	$lb/in.^2$
	DEAC	O.D. of tubes	ft
	DERC	I.D. of tubes	ft
	DELTAC	Fin thickness	ft
	XKFC	Fin conductivity	Btu/hr-ft-°F
Card 2	NTF	No. tubes in row	
	NTC	No. rows of tubes	
	NSECT	No. tubes to inlet	
Card 3	STC	Spacing of tubes	ft
	WTC	Spacing of rows	ft
	TUBLC	Tube length	ft
	FPC	Fin pitch	fins/ft
	CFMC	Air flow rate	ft^3/min
Card 4	XMRIC	Refrig. mass flow rate	lbm/hr
	TASC	Condensing sat temp	°F
	TRIC	Refrig. temp into cond	°F
	HIC	Enthalpy into cond	Btu/lbm
	PIC	Refrig. press entering	$lb/in.^2$
Card 5 through N+5		(N = NTF × NTC) The table of tube connections as described in Section 2.5	

If a velocity profile of the air passing through the condenser is to be specified, the following cards are needed:

Card N+6	NV	Number of observations in the velocity profile
Card N+7 through N+6+NV	IROW and VEL	Tube number in the front row of the heat exchanger (IROW = 1 at the top and NTF at the bottom) where the velocity, VEL (ft/min), was measured.

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