A Computer Program for Analyzing Moist Air in Fin and Tube Crossflow Heat Exchangers

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A COMPUTER PROGRAM FOR ANALYZING MOIST AIR IN FIN AND TUBE CROSSFLOW HEAT EXCHANGERS

BY

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B.S., Florida Technological University, 1973

RESEARCH REPORT
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ABSTRACT

A computer model of a fin and tube air-to-air heat exchanger is presented. The model incorporates a computational scheme to account for latent effects due to small amounts of moisture in one or both fluid streams. A testing program is described which was performed in order to mathematically characterize the heat transfer and pressure drop relationships of the tube with turbulator used in the heat exchanger. These relationships are included in the computer model. A comparison of the computer model to heat exchanger test data indicates that the computer model may be relied upon to provide design and analysis information. Finally, a parametric study is performed using the computer model in order to explore the characteristics of the heat exchanger and to demonstrate its usefulness to the heat exchanger designer. It is concluded that, in addition to presenting an analysis tool for heat exchanger design, there are several important secondary results. These include; verification of the modelling technique, the analytical description of the tube with turbulator and, identification of a problem area in the header design.
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<table>
<thead>
<tr>
<th>Symbol</th>
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</tr>
</thead>
<tbody>
<tr>
<td>Btu</td>
<td>British Thermal Unit</td>
</tr>
<tr>
<td>$c_{\text{max}}$</td>
<td>Heat Capacity of the Maximum Fluid</td>
</tr>
<tr>
<td>$c_{\text{min}}$</td>
<td>Heat Capacity of the Minimum Fluid</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific Heat</td>
</tr>
<tr>
<td>$D$</td>
<td>Diameter</td>
</tr>
<tr>
<td>$D_h$</td>
<td>Hydraulic Diameter</td>
</tr>
<tr>
<td>$f$</td>
<td>Fanning Friction Factor</td>
</tr>
<tr>
<td>$f_{\text{pi}}$</td>
<td>Fins per Inch</td>
</tr>
<tr>
<td>$ft$</td>
<td>Feet</td>
</tr>
<tr>
<td>$FV$</td>
<td>Face Velocity</td>
</tr>
<tr>
<td>$G$</td>
<td>Mass Flow Rate per Unit Area</td>
</tr>
<tr>
<td>$g_c$</td>
<td>Gravitational Constant, 32.2 ft lbm/lbf sec$^2$</td>
</tr>
<tr>
<td>$g_{\text{pm}}$</td>
<td>Gallons per Minute</td>
</tr>
<tr>
<td>$h$</td>
<td>Convection Film Coefficient</td>
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<tr>
<td>$h_f$</td>
<td>Enthalpy of Liquid</td>
</tr>
<tr>
<td>$h_g$</td>
<td>Enthalpy of Gas</td>
</tr>
<tr>
<td>$h_l$</td>
<td>Enthalpy of Solid</td>
</tr>
<tr>
<td>$hr$</td>
<td>Hour</td>
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<tr>
<td>I.B.</td>
<td>Inside Diameter</td>
</tr>
<tr>
<td>in.</td>
<td>Inch</td>
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<tr>
<td>in. Hg</td>
<td>Inches of Mercury</td>
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<td>Definition</td>
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<tr>
<td>( k )</td>
<td>Thermal Conductivity</td>
</tr>
<tr>
<td>( L )</td>
<td>Length</td>
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<tr>
<td>( \text{Ibf} )</td>
<td>Pounds Force</td>
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<tr>
<td>( \text{Ibm} )</td>
<td>Pounds Mass</td>
</tr>
<tr>
<td>( \text{LMTD} )</td>
<td>Log Mean Temperature Difference</td>
</tr>
<tr>
<td>( \ln )</td>
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</tr>
<tr>
<td>( \log )</td>
<td>Logarithm to base 10</td>
</tr>
<tr>
<td>( \dot{m} )</td>
<td>Mass Flow Rate</td>
</tr>
<tr>
<td>( \text{min} )</td>
<td>Minute</td>
</tr>
<tr>
<td>( N_{\text{tu}} )</td>
<td>Number of Transfer Units</td>
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<tr>
<td>( \text{Nu} )</td>
<td>Nusselt Number</td>
</tr>
<tr>
<td>( \text{O.D.} )</td>
<td>Outside Diameter</td>
</tr>
<tr>
<td>( P, p )</td>
<td>Pressure</td>
</tr>
<tr>
<td>( \text{Pr} )</td>
<td>Prandtl Number</td>
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<tr>
<td>( Q )</td>
<td>Heat Transferred</td>
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<tr>
<td>( R )</td>
<td>Ratio of Minimum to Maximum Heat Capacity</td>
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<tr>
<td>( \text{Re} )</td>
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<td>( T )</td>
<td>Temperature</td>
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<td>( U_0 )</td>
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<td>Humidity Ratio</td>
</tr>
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<td>Pressure Difference</td>
</tr>
<tr>
<td>( \varepsilon )</td>
<td>Effectiveness</td>
</tr>
<tr>
<td>( \eta )</td>
<td>Fin Efficiency</td>
</tr>
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</table>
\(\rho\)  \(\mu\)

**Density**

**Viscosity**
CHAPTER I
INTRODUCTION

Purpose of Project

The purpose of this project is to develop an analytical tool for use in the design and analysis of an air-to-air fin and tube heat exchanger. A primary requirement of this analytical tool is that it must be capable of analyzing the heat exchanger with moist air as the heat transfer fluid on either or both sides of the exchanger. The analytical tool is in the form of a FORTRAN computer program.

As part of the information gathering required for building the computer program, a testing sequence was performed on the heat transfer tube used in the fin and tube heat exchanger. The results of this test are empirical relationships describing the heat transfer and the pressure drop characteristics.

Background

The ROVAC Corporation, of Rockledge, Florida is developing an air cycle refrigeration system for truck transport use. The air cycle system offers many potential advantages over conventional vapor compression cycle (fluorocarbon) refrigeration. Among these advantages are potentially higher efficiency, lower maintenance costs and environmental compatibility.
However, heat exchanger selection is a problem area in the system. Due to the fact that the working fluid is air, a high performance or compact heat exchanger surface is required. Additionally, for commercial feasibility, the heat exchangers must be economical to build or purchase, a feature not usually associated with compact heat exchangers. The cold side heat exchanger (analogous to the evaporator in a vapor compression system) has one additional requirement. (See Figure 1-1 for a system schematic of the truck refrigeration system.) For a truck refrigeration system, frost buildup on the cold side heat exchanger requires occasional defrosting. The external surface of the heat exchanger must be constructed to minimize the incidence of frost buildup and to allow water to run off during defrost periods.

A number of heat exchanger types were examined for use in this system. A plate and fin unit was examined first.\(^1\) The "shelf-like" characteristics of the external plate-fin surface formed small reservoirs that allow frost to build up easily and, during defrost cycles, prevent the water from flowing off. Also, the cost of these exchangers is too high to be considered for a commercial product. Next, a flat tube with plate fin heat exchanger was tried.\(^2\) This unit has an extruded flat tube with multiple parallel passages for the internal surface, and has a plate-fin sandwiched between

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\(^1\)Purchased from the Hughes-Treitler Corp., Garden City, New Jersey.

\(^2\)Purchased from the Modine Co., Racine, Wisconsin.
Figure 1-1

Schematic of Refrigerant Flow for Truck Refrigeration System
the flat tubes for the external surface. This item was reasonably priced but retained the frost buildup problems of the plate and fin heat exchanger.

The heat exchanger finally selected by ROVAC for use in the truck refrigeration system is a fin and tube type.\textsuperscript{3} The fins are conventional plain fins and have 1" x 1" inline tube spacing. The tubes are 3/8" O.D. x .016" wall and have twisted tape turbulators inserted in them. The fins are oriented vertically and are easily defrosted. In addition, they are affordably priced. This heat exchanger and the two described above are shown in Figures 1-2 and 1-3.

The unit looks like a conventional fin and tube heat exchanger from the outside, but it is the addition of the turbulator that raises the performance to a level acceptable in air cycle machinery. "Acceptable" performance in this case is met by a heat exchanger that has an effectiveness of at least 90%, a pressure drop on the tube side of less than .75 psi, a pressure drop on the fin side of less than .25 inches of water, and yet will fit the physical envelope available.

The turbulator itself is just a thin strip of aluminum, 0.006" thick and as wide as the tube I.D., that has been twisted into a spiral shape. The function of the turbulator is to force the flow stream to follow a spiral path through the tube, thus promoting

\textsuperscript{3}Furnished by Sundstrand Heat Transfer, Inc., Dowagiac, Mich.
Figure 1-2
Plate and Fin Heat Exchanger, Top (Hughes-Treitler Corp.). Flat Tube and Plate Fin Heat Exchanger, Bottom (Modine Co.).
turbulence and thereby breaking up the boundary layer that tends to build up inside a smooth tube. The result is a higher film coefficient than a smooth tube.

Until now, selection of heat exchanger size has been based solely on "engineering judgement". The use of a computer program for heat exchanger analysis enables the choice to be made in a quantifiable manner. Also, the program lends itself well to performance of parametric analyses (i.e.: What are the effects of varying fin density, fan flow, or heat exchanger width, etc. on heat transferred?). A parametric analysis example is the subject of Chapter IV.

The "moist air heat transfer fluid" requirement is dictated by a very promising offshoot of air cycle technology, the air/vapor cycle. This is a combination of the reverse Brayton cycle (air cycle and the reverse Rankine cycle (vapor compression cycle). In practice, the primary refrigerant, air, is combined with a small amount of a secondary refrigerant. The initial analyses and experiments are being conducted using water as the secondary refrigerant, although future work will certainly include investigations of other secondary refrigerants. An in-depth discussion of the cycle is available in references (4) and (5).

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The impact of the secondary refrigerant upon heat exchanger analysis can be very significant, as evidenced by the studies in Chapter IV. This is due to the latent heat transfer which occurs with the phase change of the secondary refrigerant. The combination of sensible and latent heat transfer requires that the heat exchanger be analyzed as a large number of very small interactive heat exchangers. The technique is discussed in Chapter III.

Scope of Project

It has been the purpose of this project to build a computer program that analyzes a fin and tube heat exchanger with turbulators in the tubes and with moist air as the heat transfer fluid. This involved searching the available literature for data upon which to base the analyses, and testing where required to fill voids in the literature. The computer model was verified by comparing to a test of a heat exchanger. After the computer model verification, the program was used to perform parametric analyses of a sample heat exchanger.
CHAPTER II

TUBE HEAT TRANSFER TESTS

Tube Description

As part of the information and data gathering process required in the analysis of a heat exchanger, the empirical or the theoretical relationships describing heat transfer and pressure drop on both surfaces is required. In this instance, the two surfaces are the fin surface and the tube surface. Data on heat transfer and pressure drop for the fin surfaces is readily available.\(^1\),\(^2\)

However, the literature and vendor search failed to produce this data for the tube with turbulator insert. Therefore, it was necessary to test a representative tube to produce the data. A heat exchanger built by Sundstrand was supplied to ROVAC with an incorrect header design. A single tube was cut out of this core and the fin stock was removed from the tube.

The tube is a 3/8 in. O.D. x .016 in. wall copper tube. Due to the fact that the tube had been expanded inside the fins, the outside surface had a very slight ripple, corresponding to the nesting

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of the fins, and the actual O.D. was measured as .390 in. Throughout the active length of the tube, an aluminum twisted tape turbulator had been installed. The turbulator itself is a long thin strip of aluminum, .006 in. thick and as wide as the I.D. of the tube. The tape is twisted into a spiral form before being inserted into the tube. A sketch of the tube is shown in Figure 11-1.

Test Plan

At the outset of this testing effort, the assumption is made that a relationship describing the heat transfer characteristics of this tube will be of the form:

\[ Nu_d = C \cdot Re^m \cdot Pr^n \]

where \( C, m \) and \( n \) are experimentally determined constants. In order to fit data to an equation of this nature, the tube must be tested as a heat exchanger over a wide range of Reynolds numbers, and for more than one fluid to get the Prandtl number variation.

The tube used in this test is 28 in. long, and has been built into a double tube heat exchanger. The test section is the center tube and it is surrounded by a 3/4 in. copper tube. The annulus between the test section and the 3/4 in. tube serves as a passage through which water is passed over the outer surface of the test section.

The test section tube has 6 split-junction copper-constantan thermocouples soldered to the outer surface. These are placed at each end of the active test section, and also throughout the center
Figure 11-1

3/8" Tube with Twisted Tape Insert
of the test section. The temperature measurements at each end were the only ones used in calculations; the others were used just to observe the temperature distribution throughout the tube length. Other temperature instrumentation included thermocouple probes inserted at the entrance to and exit from the test section and the outer annulus section.

Pressure taps were located at the entrance to and exit from the test section. These were connected across a differential manometer and this was used to measure test section pressure drop. The water flow rate to the outer annulus section was measured by connecting a rotameter into the water supply line. Figure 11-2 shows the assembled test section with thermocouples and pressure taps.

There were two fluids used in the test section, heated air and heated water. The first portion of the testing was with the heated air. For this portion, a turbine wheel velocity probe was used to measure air velocity.

The turbine wheel probe was being used in a slightly different manner than designed. The instrument manufacturer suggested that it be calibrated in the system using a pitot-static tube as the reference. This instrumentation can be seen in Figures 11-3 and 11-4. The end result of this calibration was the curve shown in

---

3 The suggestion is from a telephone conversation with a Sales Engineer at Flow Technology, Inc., Phoenix, Az.
Figure 11-3

Schematic of Tube Test Heat Exchanger with Instrumentation
Figure 11-4

Figure 11-5. The data was curve-fitted with a third-order polynomial to within 4%. The reason for using the turbine-wheel probe as the basic test instrument, instead of the pitot-static tube, is the increased resolution of the turbine-wheel output, especially at low flow rates.

Additional data required when using air as the test fluid are the ambient temperature, barometric pressure, and relative humidity.

For the second portion of the testing, using heated water, a rotameter is used to measure the water flow rate.

During the conduct of the test, the test section flow rate was adjusted after each set of data was taken. This provides a large quantity of Reynolds number variations. The use of two fluids provides the necessary Prandtl number variation.

Sample Data Set and Reduction

After the data taking was completed, the next step was reducing the data into more meaningful expressions. In this case, the data were reduced to Reynolds Number, Prandtl Number, Nusselt Number, and Fanning Friction Factor.

One of the data points taken with air as the test fluid is analyzed below as a sample.

Ambient Temperature: 76.0 F  
Air Temperature into Test Section: 211.3 F  
Air Temperature out of Test Section: 76.0 F  
Tube Wall Temperature at Air Exit: 69.8 F  
Turbo-Probe Reading: 5.05
Pressure Drop across Test Section: 0.1 in. Hg
Relative Humidity: 73%
Barometric Pressure: 30.43 in. Hg
Rotameter: 1.0 gpm

Calculations

Air Velocity:

\[
V_{air} = 7.64200 + 17.87325 \times 5.05 - .12244 \times 5.05^2 \\
+ .00095 \times 5.05^3
\]

\[V_{air} = 94.9 \text{ ft/min}\]

Air Volume through test section:

\[
V_{air} = 94.9 \dfrac{\text{ft}}{\text{min}} \times .008728 \dfrac{\text{ft}^2}{\text{in.}^2}
\]

\[V_{air} = .828 \dfrac{\text{ft}^3}{\text{min}}\]

Humidity Ratio: (from psychrometric chart)

\[W = .0141 \text{ lbm moisture/lbm dry air}\]

Air Density:

\[
\rho_{air} = 30.43 \frac{\text{in. Hg}}{144 \frac{\text{in.}}{\text{ft}^2}} \times .4912 \frac{\text{lbf}}{\text{in.}^2} \times \frac{\text{lbm R}}{53.342 \text{ ft} \times \text{lbf}}
\]

\[
\times \frac{1}{(76 + 460)R} \times 144 \frac{\text{in.}^2}{\text{ft}^2} \times (1 + .0141)
\]

\[\rho_{air} = .076 \dfrac{\text{lbm}}{\text{ft}^3}\]

Air Mass Flow Rate:

\[
\dot{m}_{air} = .828 \dfrac{\text{ft}^3}{\text{min}} \times .076 \dfrac{\text{lbm}}{\text{ft}^3} \times 60 \dfrac{\text{min}}{\text{hr}}
\]

\[\dot{m}_{air} = 3.794 \text{ lbm/hr}\]

Specific Heat of Air:

\[
c_{p,air} = \frac{1}{1 + .0141} \times .240 \frac{\text{Btu}}{\text{lbm} \cdot \text{F}} + .0141 \times \frac{.505 \text{ Btu}}{\text{lbm} \cdot \text{F}}
\]
\[ c_{p, \text{air}} = 0.244 \text{ Btu/lbm F} \]

Heat Transferred from Air:
\[
Q = 3.794 \text{ lbm/hr} \times 0.244 \text{ Btu/lbm F} \times (211.3 \text{ F} - 76.0 \text{ F})
\]
\[
Q = 125 \text{ Btu/hr}
\]

Effectiveness:
\[
\varepsilon = \frac{Q}{Q_{\text{max}}} = \frac{C_{h}}{C_{\text{min}}} \left( t_{h, \text{in}} - t_{h, \text{out}} \right)
\]
\[
C_{h} = C_{\text{min}}
\]
\[
\varepsilon = \frac{211.3 \text{ F} - 76.0 \text{ F}}{211.3 \text{ F} - 69.8 \text{ F}}
\]
\[
\varepsilon = 0.956
\]

Heat Capacity:
\[
C_{\text{air}} = 3.794 \text{ lbm/hr} \times 0.244 \text{ Btu/lbm F}
\]
\[
C_{\text{air}} = 0.924 \text{ Btu/hr F}
\]
\[
C_{\text{water}} = 1.0 \text{ gal/min} \times 0.1337 \text{ ft}^3 \text{ gal} \times 60 \text{ min/hr} \times 62.32 \text{ lbm/ft}^3 \times 0.999 \text{ Btu/lbm F}
\]
\[
C_{\text{water}} = 499.23 \text{ Btu/hr F}
\]

Number of Transfer Units \((N_{tu})\):
\[
N_{tu} (\text{counterflow}) = \frac{1}{R - 1} \ln \frac{1 - \varepsilon}{1 - R\varepsilon}
\]
\[
R = \frac{C_{\text{min}}}{C_{\text{max}}} = 0.00185
\]
\[
N_{tu} = \frac{1}{0.00185 - 1} \ln \frac{1 - 0.956}{1 - (0.00185 \times 0.956)}
\]
\[
N_{tu} = 5.152
\]

Film Coefficient:
\[ U_0 = \frac{N_t u C_{\text{min}}}{A_0} \]

\[ U_0 = 3.132 \times 0.924 \text{ Btu} \times \frac{1}{\text{hr F}} \times \frac{1}{.24036 \text{ ft}^2} \]

\[ U_0 = 12.044 \text{ Btu/hr ft}^2 \text{ F} \]

**Nusselt Number:**

\[ \text{Nu}_d = \frac{h D}{k} \]

\[ h = U_0 \]

\[ \text{Nu}_d = \frac{12.044 \text{ Btu} \times 0.02983 \text{ ft} \times \text{hr ft} \times \text{F}}{\text{hr ft}^2 \text{ F}} \]

\[ \text{Nu}_d = 22.22 \]

**Reynolds Number:**

\[ \text{Re}_d = \frac{D G}{\mu} \]

\[ G = 3.794 \text{ lbm} \times \frac{1}{\text{hr}} \times \frac{1}{6.9887 \times 10^{-4} \text{ ft}^2} \]

\[ \text{Re}_d = 0.0298 \text{ ft} \times 5428.8 \text{ lbm} \times \frac{\text{ft} \times \text{hr}}{\text{ft}^2 \text{ hr}} \times \text{.0477 lbm} \]

\[ \text{Re}_d = 3392.8 \]

**Fanning Friction Factor:**

\[ f = \frac{\Delta P D \rho g_c}{2 \text{ L G}^2} \]

\[ f = \frac{0.1 \text{ in. Hg} \times 0.0298 \text{ ft} \times 0.076 \text{ lbm} \times \text{ft}^4 \times \text{hr}^2}{2 \times 2.354 \text{ ft} \times \text{ft}^3 \times (4250)^2 \text{ lbm}^2} \]

---

The data reduction for the cases with water as the test fluid follow a similar pattern. To simplify the data reduction, two programs were written for use with a Texas Instruments TI-59 programmable calculator. One of them performs the data reduction for the air data, and the other performs the data reduction for the water data. Copies of the raw data taken in the tube testing and the data reduction programs are included as appendix B.

With the raw data reduced to groups of dimensionless numbers, the next step was to correlate them into equation form. As previously stated, the assumption was made that the heat transfer relationship is of the form:

\[ Nu_d = C \, Re^m \, Pr^n. \]

The first step towards finding the constants \( C, m \) and \( n \) in this equation was to make a log-log plot of \( Nu_d \) vs. \( Re_d \) for one fluid \( (Pr = \text{constant}) \). This gives a first estimate for the exponent \( m \). This plot is shown in Figure II-6 for the air data. The resulting value of \( m \) is 0.598.

Next, the data for both fluids were plotted as \( \log Nu_d/Re_d^{598} \) vs. \( \log Pr \) to give a value for the exponent \( n \). The resultant is: \( n = .118 \). This is shown in Figure II-7. Finally, all the data were plotted once again as \( \log Nu_d/Pr^{.118} \) vs. \( \log Re_d \), as shown in Figure II-8. This yields a final value for the exponent \( m \) and a value
for the constant $C$. The final equation is now:
\[ \text{Nu}_d = 0.14192 \text{Re}^{0.619} \text{Pr}^{0.118} \]

The relationship describing the pressure drop characteristics of this tube is assumed to be of the form:
\[ f = D \text{Re}^p \]
where $D$ and $p$ are constants. A log-log plot of $f$ vs $\text{Re}_d$, shown in Figure 11-9, yields the equation:
\[ f = 0.2344 \text{Re}^{-0.19} \]

Results

The testing program has yielded a pair of equations describing the heat transfer and pressure drop characteristics of the tube with twisted tape turbulator.

For comparison purposes, the heat transfer and pressure drop characteristics of a smooth tube are given by:\(^6\)
\[ \text{Nu}_d = 0.023 \text{Re}_d^{0.8} \text{Pr}^{0.4} \]
and
\[ f = 0.316 \text{Re}^{-0.25} \]

In the first equation, the heat transfer relationship, it can be seen that the tube with turbulator yields a higher Nusselt number than the smooth tube at the lower Reynolds numbers. This is a very

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\(^6\) Ibid., pp. 160 and 176.
Figure II - 9

F vs. Re
pronounced difference in the range of Reynolds numbers expected to be used in air-to-air heat exchangers for truck refrigeration systems (Re < 10,000). For the case using air (Pr = .7), the tube with turbulator yields higher Nusselt numbers than the smooth tube at Reynolds numbers up to 40,000.

Examining the pressure drop equations, it can be seen that, as expected, the tube with turbulator will have a higher pressure drop than the smooth tube. However, the pressure drop increase is not excessive and still well within the desirable range.
CHAPTER III
COMPUTER PROGRAM

Overall Approach

One of the first steps taken in the creation of a computer program is defining the limitations and the input and output parameters. At the outset of this project it was determined that the computer program to be written would analyze the Sundstrand tube and fin heat exchanger, 1" x 1" inline pattern, with 3/8" tubes and turbulators. This heat exchanger can be analyzed as crossflow in the single-pass case, or as countercrossflow in the multiple-pass case. The problem input parameters will include inlet temperatures, pressures and flow rates along with the physical dimensions of the core. The program output will include the heat transferred, the outlet temperatures, the core pressure drop, and the effectiveness.

In selecting which parameters are to be used as inputs to the program and which are to be calculated, the general uses of the program are defined. In heat exchanger analysis, there are two main problem solving paths. The first path is; given heat exchanger core size, what is its heat exchanging ability at the specified flow conditions. The second path is; given the heat transfer required, what must be the physical size of the heat exchanger core. The first path is referred to as the "direct" problem and the second
path as the "inverse" problem.¹

The solution undertaken with the program presented here is the direct problem. This is in spite of the fact that a majority of cases require solution of the inverse problem. In order to solve these inverse problem cases, where performance is specified and the dimensions are to be determined, the dimensions are estimated and the calculated performance is compared to the specified performance. The dimensions are re-estimated as necessary until the calculated and specified performances are sufficiently close.

The reason for this iterative approach to the inverse problem solution lies in the methodology which must be used in accounting for the heat transfer fluid phase changes. As can be seen in the discussion below, this phase changing problem requires that dimensions be given and that performance be calculated.

In heat exchanger analysis, there are two widely accepted analysis methods, the "Log Mean Temperature Difference" (LMTD) method and the "Effectiveness - Ntu" method. In the LMTD method, the calculations are based on knowledge of the inlet and outlet temperatures. In the instances where there is moisture undergoing phase changing, this method can be cumbersome to use.

The Effectiveness - Ntu method has been chosen for use in this computer program. This method relies primarily on the computation

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of two dimensionless numbers, the effectiveness ($\varepsilon$) and the Number of Transfer Units ($N_{tu}$). The effectiveness is defined as:

$$\varepsilon = \frac{Q}{Q_{\text{max}}}$$

where $Q$ is the actual heat transferred and $Q_{\text{max}}$ is the maximum thermodynamically possible heat transfer. The $N_{tu}$ is defined as:

$$N_{tu} = \frac{A}{C_{\text{min}}} U_{av}$$

where $U_{av}$ is the average heat transfer coefficient through the wall separating the two heat transfer fluid streams, $A$ is the area of the wall, and $C_{\text{min}}$ is the heat capacity of the minimum fluid. For a single-pass crossflow heat exchanger, the effectiveness and $N_{tu}$ are related by the following equation:

$$\varepsilon = R \frac{1}{N_{tu}} \sum_{n=0}^{\infty} \frac{(1 - \exp(-N_{tu}) \sum_{k=0}^{n} \frac{N_{tu}^k}{k!} (1 - \exp(-R N_{tu}))^k)}{k!}$$

where $R$ is the ratio of heat capacities of the minimum fluid to the maximum fluid. The above three equations are sufficient to describe the heat transfer of a crossflow heat exchanger assuming that both heat transfer fluids are single phase.

---


An added dimension is brought into the analysis with the addition of a phase-changing component in one or both fluid streams. An assumption of primary importance to this analysis is that the mass of the secondary component, water, is very small compared to the mass of the primary component, air. This assumption is necessary in order to lend credence to a further assumption, that the phase changing does not affect the film coefficient. That is, there are no boiling or condensation factors raising the film coefficient. The amount of water present in the fluid stream is limited to that amount present in saturated air at approximately 30°F.

The usual analysis method with single phase fluids is to find the $N_{tu}$ and from this the effectiveness can be calculated. Knowing effectiveness, the heat transferred and the exit temperatures can be calculated. However, with a phase-changing component in the fluid stream, the exit temperature is not the same as with a single phase fluid. Part of the energy transferred has been a latent transfer and part has been a sensible transfer. This means that the temperature distribution throughout the core is different than the single-phase case, and therefore, the heat transferred must be different.

The approach taken in the solution of this problem is to divide the heat exchanger up into a large number of very small single-pass heat exchangers. Each section is analyzed to find the heat transferred. Then the exit temperature, humidity and condensate is used as the inlet condition for the next section. The division
of a typical heat exchanger core into sections is shown schematically in Figure III-1. In the limit, as the section length becomes infinitesimally small, the summation of energy transfers from each section approaches the actual energy transfer. In the computer solution, an estimate must be made of the temperature distribution at the entrances to each section. Then, an analysis is made for each of the sections, thus updating the temperature distribution estimate. The process of analyzing all the sections continues until successive sets of analysis have the same temperature distribution (within 0.001F).

Description of Program

The computer program written for this project is a FORTRAN V program and is being used on a Univac 1108 computer system. The program name is RFS-FINTUBE.

The program operation begins by reading the input variables. A NAMELIST format is used. The variable names and their descriptions are shown in Table III-1. The input variables are stored in a data file which must be assigned as unit 8. After the READ statement, the input is checked for omissions. Any omissions will cause the program to call subroutine ERROR which prints the name of the offending variable and stops further execution.

Next, the information on moisture content is stored in an array W(I,J,K,L); where I = 1 for moisture in vapor form, I = 2 for moisture in liquid form, and I = 3 for moisture in solid form; J = 1
Figure III-1

Division of Heat Exchanger Core into Multiple Single Pass Sections for Two-Phase Flow Analysis
<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>HEIGHT</td>
<td>The height of the heat exchanger in inches. This dimension is perpendicular to the fan flow direction and perpendicular to the tubes.</td>
</tr>
<tr>
<td>WIDTH</td>
<td>The width of the heat exchanger in inches. This is the dimension parallel to the tubes.</td>
</tr>
<tr>
<td>DEPTH</td>
<td>The depth of the heat exchanger in inches. This is the dimension parallel to the fan flow.</td>
</tr>
<tr>
<td>NFPI</td>
<td>The number of fins per inch used on the core. This variable is input as an integer.</td>
</tr>
<tr>
<td>TTIN</td>
<td>The inlet temperature of the fluid on the tube side of the heat exchanger. The temperature is input in °F.</td>
</tr>
<tr>
<td>TPIN</td>
<td>The inlet pressure of the fluid on the tube side in psia.</td>
</tr>
<tr>
<td>TRATE</td>
<td>The tube side fluid mass flow rate. Its units are lbm/min.</td>
</tr>
<tr>
<td>TTYPE</td>
<td>An identifier of up to 6 hollerith characters, used to identify the type of fluid on the tube side. This variable has no influence on the operation of the program and is merely printed in the output listing.</td>
</tr>
<tr>
<td>FTIN</td>
<td>The fin side inlet temperature in °F.</td>
</tr>
<tr>
<td>FPIN</td>
<td>The inlet pressure, in psia, on the fin side of the core.</td>
</tr>
<tr>
<td>FVGL</td>
<td>The volume rate of fluid passing through the fin side of the heat exchanger. The units are Actual Cubic Feet per Minute (ACFM).</td>
</tr>
<tr>
<td>FTYPE</td>
<td>An identifier for the fin side fluid. Its type and purpose are identical to TTYPE.</td>
</tr>
<tr>
<td>TITLE</td>
<td>A 78 character string of Hollerith characters used as an identifier for the individual set of analysis. This identifier is printed at the beginning of the output listing.</td>
</tr>
<tr>
<td>TRH</td>
<td>The relative humidity of the inlet tube side fluid, in percent.</td>
</tr>
</tbody>
</table>
TABLE III-1 (contd.)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>TWLIN</td>
<td>The ratio of inlet liquid moisture to air on the tube side.</td>
<td>lbm moisture/lbm dry air</td>
</tr>
<tr>
<td>TWIN</td>
<td>The ratio of inlet solid moisture (ice) to air on the tube side.</td>
<td>lbm moisture/lbm dry air</td>
</tr>
<tr>
<td>FRH</td>
<td>The fin side relative humidity in percent.</td>
<td></td>
</tr>
<tr>
<td>FWLIN</td>
<td>The ratio of inlet liquid moisture to air on the fin side.</td>
<td>lbm moisture/lbm dry air</td>
</tr>
<tr>
<td>FWSIN</td>
<td>The ratio of inlet solid moisture (ice) to air on the fin side.</td>
<td>lbm moisture/lbm dry air</td>
</tr>
</tbody>
</table>
for data pertaining to the entrance of a section, and $J = 2$ for the exit from the section; $K = 1$ for tube side data, and $K = 2$ for fin side; $L$ is the section number within the heat exchanger. As an example, the moisture in vapor form at the tube side exit from section 5 would be stored in $W(1,2,1,5)$. In order that consistent units are used for all the humidity data, the input relative humidities are converted to humidity ratios. The vapor, liquid and solid water contents are all used as lbm moisture/lbm dry air.

The first step toward finding the temperature distribution is to predict an exit temperature for each side. This is done by a call to subroutine HXFER. This makes a single set of calculations for the entire heat exchanger, as if all fluids were single phase. The predicted exit temperatures are used to estimate the temperature distribution throughout the unit.

Next the heat exchanger is divided up into sections approximately .75 inch wide. A large number of sample executions was run to help find the best section width. In Figure III-2, these sample execution results have been plotted as a function of section width. By drawing a curve through the data, the limiting case can be estimated as the section width approaches zero. In this sample case, the error at .75 inch section width is only 2.5%. Three-fourths inch was chosen as a good balance between accuracy of result and cost of program execution. The total number of sections is $(\text{width} / .75 \text{ inch}) \times \text{number of passes}$. Each section is given an estimated inlet and outlet temperature using the inlet temperatures read in
the input data and the outlet temperatures predicted above. As a crude approximation, a linear temperature distribution is assumed between the inlets and outlets of the heat exchanger.

Now, an iterative process is begun where each section is individually examined as a single pass heat exchanger. The outlet temperature and moisture data for each section is used as the input to the following sections. As each section is examined, the output temperature is compared with the previous estimation. If the difference between the output temperature calculation and its predecessor is greater than 0.001°F, a flag is set which indicates that the entire set of calculations must be repeated. Once all the sections have passed the temperature difference criteria, the problem is finished. The program then calls on subroutine QMAX to find the maximum thermodynamically possible heat transfer. This is compared against the actual heat transfer calculated, to determine overall heat exchanger effectiveness. Finally the input and output data are printed in a data table. The data table format was chosen to present a cohesive view of the calculated heat exchanger performance and the conditions leading to that result, rather than printing a block of data labelled "Input" and another block of data labelled "Output".

In the previous paragraphs, the operation of the program in general, and the main routine in particular, have been discussed. Now, the operation of the subroutines, including special techniques, will be discussed in greater detail.
In the main routine, note that all of the heat transfer calculations are made by calling subroutine HXFER. This subroutine coordinates the calculation of film coefficients on both the fin and tube sides of the heat exchanger. It gathers the pressure drop information and it finds the heat transferred along with the exit temperatures and moisture contents.

In operation, the average of the inlet tube side temperature and the predicted outlet temperature is figured. This average temperature is an input in the call to subroutine TCOEF, which calculates a film coefficient for the tube surface. Next, subroutine FFPD is called to calculate the Fanning friction factor and the pressure drop through the tube.

An average fin side temperature is computed in the same manner as above for the tube side, and is used in the call statement to subroutine FCOEF. This routine returns a film coefficient for the fin surface. Then, subroutine PDO is called on to figure the fin side pressure drop.

Next, the fin efficiency is calculated. The fin efficiency, \( \eta_0 \), is given as:

\[
\eta_0 = 1 - \frac{A_f}{A} (1 - \eta_f)
\]

where:

\[
\eta_f = \frac{\tanh \left( \frac{ml}{m_f} \right)}{ml}
\]
\[ m = \frac{2 h}{k \delta} \]

\( A_f \) = fin surface area

\( A = \) total surface area (fin + tube)

\( l = \) fin length

\( \delta = \) fin thickness

\( h = \) film coefficient on fin

\( k = \) conductivity of fin material

The fin efficiency, along with the tube and fin film coefficients are now used to determine the overall coefficient, \( U_0 \). The equation for \( U_0 \) is:

\[ U_0 = \left( \frac{1}{h_f} + \frac{1}{(h_t/A_t/A_0) h_t} \right)^{-1} \]

where:

\( h_f = \) fin side film coefficient

\( h_t = \) tube side film coefficient

\( A_t = \) tube side surface area

\( A_0 = \) fin side total surface area

The overall heat transfer coefficient is used, along with the outside surface area and the minimum fluid \( C_{\min} \) (\( C_{\min} \) is the smaller of \( h \cdot c_p \) for the fin side and the tube side) to determine the Number of Transfer Units, \( N_{tu} \). This is used, together with \( R \) (\( R \) is the ratio of \( C_{\min}/C_{\max} \)) to find the effectiveness. The effectiveness is computed by routine EFFNTU.

\[ ^5 \text{Ibid, p. 13.} \]
The heat transferred is given by the equation:

\[ Q = C_{\text{min}} \varepsilon (T_{\text{h,in}} - T_{\text{c,in}}) \]

where: 
- \( Q \) = effectiveness
- \( T_{\text{h,in}} \) = hot side inlet temperature
- \( T_{\text{c,in}} \) = cold side inlet temperature

Finally subroutine TMOUT is called to use the inlet temperature, moisture data and the heat transferred to determine the outlet temperature and moisture data.

Subroutine HXFER calls upon a number of subroutines to perform specialized calculations. The first is subroutine TCOEF which has the purpose of calculating a film coefficient on the tube side of the heat exchanger. The initial step towards finding the film coefficient is determining the physical properties of the fluid. Subroutine PPROP is used to find the density, viscosity, conductivity, and Prandtl number of the fluid stream. The specific heat is found by calling Subroutine SPECHT.

Next, the flow area of the tube surface is determined. This is used to find the flow per unit area, \( G \), which is used along with the hydraulic diameter and the viscosity to find the Reynolds number. This is in turn used to calculate the Nusselt number. The equation is:

\[ Nu = 0.1419_2 Re^{0.613} Pr^{0.118}. \]

The source of this equation is the test data of Chapter II. The Nusselt number is used to calculate the tube side film coefficient.

The next subroutine to be examined is subroutine FFPD which,
like the preceding routine has its basis in the experimental data of Chapter II. The purpose of this routine is to calculate the Fanning friction factor and the pressure drop through the tube side of the heat exchanger. The friction factor is given by the equation:

\( f = 0.2344 \text{Re}^{19} \)

Subroutines FCOEF AND PDO serve the same purposes as the two previously examined subroutines, except that they are used to find the fin side film coefficient and pressure drop, respectively. In the operation of subroutine FCOEF, the fluid properties are first obtained by calling subroutines PPROP and SPECHT. Next, the geometric parameters of the fin surface are obtained by calling subroutine FINGEO.

Then, the flow per unit area, \( G \), is calculated and used along with the hydraulic diameter and the viscosity to calculate the Reynolds number. The Reynolds number is used in the determination of the Stanton number.\(^6\)

\[
St = \frac{A}{\text{Re}^B \text{Pr}^{2/3}}
\]

where \( A \) and \( B \) are constants.

Finally, the Stanton number, along with the specific heat and the flow per unit area, is used to calculate the fin side film coefficient.

Subroutine PDO is used to calculate the pressure drop on the

\(^6\)Ibid., p. 13.
heat exchanger fin side. The method used for finding pressure drop is a curve fit through the data supplied in the Sundstrand Heat Transfer Manual. The data supplied is a log-log plot of pressure drop vs. face velocity. Since the data plots as a straight line, then it is possible to fit a logarithmic equation to the data. The equation is:

$$\ln(P) = A + B \ln(FV)$$

where A and B are constants depending upon the number of rows of tubes.

The pressure drop equation is based on a fin density of 12 fpi. For fin densities other than 12 fpi, a correction factor is included. This correction factor (CF) is given as a second order polynomial.\(^7\)

$$CF = -.24643 + .15384 \ (\text{FPI}) - .00406 \ (\text{FPI})^2$$

The next routine to be examined is subroutine EFFNTU. This routine calculates crosscounterflow heat exchanger effectiveness when \(N_{tu}\) is specified. First, the single pass effectiveness is calculated. This equation is:\(^8\)

$$\varepsilon = \frac{1}{R} \sum_{n=0}^{\infty} \left( 1 - \exp(-N_{tu}) \right) \sum_{k=0}^{n} \frac{N_{tu}^k}{k!} (1 - \exp(-R \ N_{tu}))$$

---


Next, the overall effectiveness is found by the following equation.\(^9\)

\[
\epsilon_0 = \frac{(1 - \epsilon R)^n - 1}{(1 - \epsilon)^n - R}
\]

In the article by Mason (reference 8), he states that "the true value of effectiveness can be approximated to within 0.1 percent by at most five terms" of the series solution for effectiveness. This subroutine uses six terms in the solution.

One of the very important subroutines insofar as the phase change aspects of the program are concerned, is subroutine TWOUT. This routine determines the exit temperature and humidity data from a heat exchanger section. This is done by simulating a psychrometric chart.

The first step is to determine whether heating or cooling is taking place. For the cooling mode, the following sequence of steps is followed to find the outlet temperature and humidity data.

The input is checked for saturation. If the input is not saturated, then calculate the exit temperature assuming constant properties. Once the outlet temperature is calculated in this fashion, check to see if the output is saturated. For the case

of non-saturated input and output, this problem is now complete.

If the output is saturated, then the temperature must be recalculated. The input humidity ratio is used to find the temperature at which saturation occurs. The energy released to reach this temperature is then calculated. This energy is subtracted from the overall energy being transferred. The difference is the heat energy available after saturation has been reached.

Next, the energy released to condense liquid water from the moist air and lower the temperature of the air to 32°F is calculated. If this energy is greater than the available energy then the temperature lies between the saturation temperature and 32 degrees. An iterative process is then used to find the exit temperature.

If the energy released to reach 32°F is less than the available energy, then the water will change phase from liquid to solid. The routine calculates the available energy after 32°F has been reached and then finds the amount of energy released for a phase change of all the liquid to solid. If this energy is greater than the amount of available energy, then the output will have part liquid and part solid.

If the amount of available energy is greater than the phase change energy, then the output will be in the region below 32°F. The routine uses an iterative scheme to determine the final output temperature and humidity data.

For the case of the heating mode, the method is identical to, but the reverse of, the method used in the cooling mode.
A routine similar to, and in fact derived from, subroutine TWCUT is subroutine QMAX. This routine calculates the maximum thermodynamically possible heat transfer, equivalent to 100% effectiveness. Recall that the definition of effectiveness is the actual heat transfer divided by the maximum heat transfer possible. Due to the fact that the heat exchanger is being examined as many small heat exchangers, the effectiveness calculated for each of the small sections has no relationship to the overall effectiveness. Therefore, the effectiveness must be separately derived.

Subroutine QMAX, as with subroutine TWCUT, simulates a psychrometric chart. The difference here is that instead of being supplied with a beginning temperature and an enthalpy change, the beginning and ending temperatures are specified and the enthalpy change is to be calculated. The procedure is identical to that used in subroutine TWCUT.

The fluid stream physical properties are derived within the routines SPECHT and PPROP. Function SPECHT calculates the specific heat of the moist air and condensate mixture. The first step is to call subroutine PPROP to obtain the specific heats for air and each of the three water phases. Then the overall specific heat of the mixture is calculated according to the following equation:

$$c_{p,\text{mix}} = c_{p,\text{air}} + c_{p,\text{vap}} \frac{W_{\text{vap}}}{W} + c_{p,\text{liq}} \frac{W_{\text{liq}}}{W} + c_{p,\text{ice}} \frac{W_{\text{ice}}}{W}$$

The resultant output is given in terms of Btu/lbm dry air - F.

Subroutine PPROP returns the physical properties of air and water. For air, the properties calculated are density, viscosity,
specific heat, thermal conductivity, and Prandtl number. For water, the properties are the specific heats of each of the phases; vapor, liquid and solid.

All of the physical properties except Prandtl number and the density of air are given by polynomial equations relating the desired property to an input temperature. The polynomial equations represent a least-squares curve fit through tables of physical property vs. temperature data.\(^\text{10}\)

In order to obtain the curve fit equations, a series of programs were written for a Texas Instruments TI-59 programmable calculator. These programs fit polynomial equations up to sixth order, exponential equations, logarithmic equations, and power equations through the data, all using the method of least squares.\(^\text{11, 12}\) Copies of these programs and instructions for their use are given in Appendix C.

Although the four types of equations given above were fitted through the data, in all cases the best results were with the polynomials. The equations and their range of validity are:

**Viscosity of Air** \((-148^\circ F < T < 482^\circ F)\)

\[
\mu (\text{lbm/ft hr}) = 0.03896 + 6.455 \times 10^{-5} T - 2.15 \times 10^{-8} T^2
\]


Specific Heat of Air (-148F < T < 752F)

\[ c_p (\text{Btu/lbm F}) = 0.23942 + 5.18519 \times 10^{-6} T + 2.11640 \times 10^{-8} T^2 \]

Thermal Conductivity of Air (-58F < T < 320F)

\[ k (\text{Btu/hr ft F}) = 0.013065 + 2.15196 \times 10^{-5} T + 1.8436 \times 10^{-10} T^2 \]

Specific Heat of Water Vapor (212F < T < 644F)

\[ c_p (\text{Btu/lbm F}) = 0.52246 - 2.5448 \times 10^{-4} T + 3.0371 \times 10^{-7} T^2 \]

Specific Heat of Liquid Water (32F < T < 212F)

\[ c_p (\text{Btu/lbm F}) = 1.0140 - 3.5991 \times 10^{-4} T + 1.9540 \times 10^{-6} T^2 \]

Specific Heat of Ice (-112F < T < 32F)

\[ c_p (\text{Btu/lbm F}) = 0.46752 + 7.1027 \times 10^{-4} T - 3.0313 \times 10^{-6} T^2 \]

In all cases above, the temperature is in degrees F. An equation describing the density of air is obtained by assuming that, within the range of temperatures and pressures encountered in these heat exchangers, the air acts as a perfect gas. From the perfect gas law then:

\[ \rho (\text{lbm/ft}^3) = \frac{P \text{ lbf} \times \text{lbm R} \times 1 \times 144 \text{ in}^2}{\text{in}^2 \times 53.342 \text{ ft lbf} \times T(R) \times \text{ft}^2} \]

Finally, the equation describing the Prandtl number for air is taken from the definition of Prandtl number:

\[ Pr = \frac{\mu c_p}{k} \]

One of the input variables to the program is the fin density. This means that the geometric properties of the fin surface are not constants, but must be calculated for each case. This is done in the subroutine FINGEO. The parameters calculated are the hydraulic diameter, the ratio of mean free flow area to gross face area,
the outside surface area of the heat exchanger (tube area + fin area), and the outside area of the fin surfaces. Since the geometry of the tube spacing is non-variable, these parameters are functions of fin spacing only.

Three subroutines that are very important in calculating the latent heat transfer in the moist air are ENTHV, ENTHL and ENTHS. These routines calculate the enthalpy of water vapor, liquid water, and ice, respectively, at a given temperature. The enthalpy of water vapor is given by

\[ h_g = F_0 + 1 \frac{F_1}{2} p^2 + 1 \frac{F_3}{4} p^4 + 1 \frac{F_{12}}{13} p^{13} + F' \]

where \( F_0, F_1, F_3, F_{12} \) and \( F' \) are constants.\(^{13}\)

For the enthalpy of the liquid and solid phases, the following equations are used.

\[ h_f = 17.661 (T - 273) - 0.1479 \left( T^2 - 273^2 \right) \]
\[ + 6.0862 \times 10^{-4} \left( T^3 - 273^3 \right) - 1.1187 \times 10^{-6} \left( T^4 - 273^4 \right) \]
\[ + 7.80297 \times 10^{-10} \left( T^5 - 273^5 \right) \]
\[ h_i = -158.9286 + 0.47125 T + 4.883 \times 10^{-4} T^2 \]

In the equation for \( h_f \), the temperature is given in degrees K and in the equation for \( h_i \), temperature is in degrees F.

Another routine which is important in the moist air calculations is subroutine PSAT. This returns the vapor pressure of water at

saturation for the given temperature. The routine covers the vapor pressure of water over the range of -40°F to +70°F. Within this temperature span are six equations, each of which cover a portion of the range. Above 50°F, two equations of the form

$$\log\left(\frac{p}{p_c}\right) = \frac{x(a + bx + cx^3 + ex^4)}{1 + dx}$$

are used. The various constants are:

\begin{align*}
p &= \text{vapor pressure (atm)} \\
p_c &= \text{critical pressure, 218.167 atm.} \\
T &= \text{temperature (K)} \\
x &= T - T_c, \text{ where } T_c \text{ is the critical temperature 647.27K.}
\end{align*}

The coefficients a through e depend on the temperature and are given as:

**200°F < T < 705°F**
\begin{align*}
a &= 3.3463130 \\
b &= 4.14113 \times 10^{-2} \\
c &= 7.515484 \times 10^{-9} \\
d &= 1.3794481 \times 10^{-2} \\
e &= 6.5644 \times 10^{-11}
\end{align*}

**and 50°F < T < 200°F**
\begin{align*}
a &= 3.2437814 \\
b &= 5.86826 \times 10^{-8} \\
c &= 1.1702379 \times 10^{-8} \\
d &= 2.1873462 \times 10^{-3} \\
e &= 0.0
\end{align*}

Below 50°F, a logarithmic curve of the form

$$\ln(P_{\text{vapor}}) = B - \frac{A}{T}$$

\[\text{ibid., p. 14.}\]
is used. The vapor pressure is in psia and the temperature in degrees K. The constants $A$ and $B$ and their range of validity are:

- $40^\circ F < T < 50^\circ F$
  - $A = 5388.60$
  - $B = 17.30488$

- $32^\circ F < T < 40^\circ F$
  - $A = 5427.45$
  - $B = 17.3383$

- $0^\circ F < T < 32^\circ F$
  - $A = 6143.76$
  - $B = 20.06714$

- $-40^\circ F < T < 0^\circ F$
  - $A = 6098.37$
  - $B = 19.88939$

The humidity ratio at saturation for a given temperature is computed by function $WSAT$. The computations are based on the definition of humidity ratio.\(^{16}\)

\[ W = \frac{M_w}{M_a} = \frac{P_w}{P_a} \left( \frac{v}{R_w} \right) \frac{T}{T_a} \]

\[ W = 0.622 \left( \frac{P_w}{P - P_w} \right) \]

In addition to the subroutines and functions discussed above, there are a number of minor routines. Each has a comment statement at its beginning defining the purpose.


Test Data and Program Comparison

In any project such as this one, the computer program output needs to be compared against test data to establish credibility. A heat exchanger of the type defined herein was purchased from Sundstrand. It was installed in a ROVAC air cycle test loop and measurements of its performance were taken. (The reason it was installed in a test loop rather than directly into a truck refrigeration unit is because of the additional instrumentation available in the test loop.) The heat exchanger core is 15 inches high, 25 inches wide and 6 inches deep. It is single pass and has 6 fins per inch.

From the test data, a representative set is shown here.

Temperature In, Tube Side = -16F
Temperature Out, Tube Side = 61F
Temperature In, Fin Side = 82F
Temperature Out, Fin Side = 68.5F
Tube Side Pressure = 0.0psig
Tube Side Pressure Drop = 33 in. W
Humidity In, Tube Side = 1.77
(Translates to 100% Relative Humidity)
Circulator Speed = 1590 rpm
(Translates to 7.28 lbm/min)
Fin Side Relative Humidity = 67%
Barometric Pressure = 30.375 in. Hg
Fan Flow = 615 cfm
Using this test data, the heat transfer can be calculated.

The heat transferred on the tube side is:

\[ Q_s = \dot{m} c_p \Delta T \] (sensible heat transfer)

\[ Q_s = 436.8 \text{ lbm} \times \frac{.24 \text{ Btu}}{\text{hr} \cdot \text{lbm} \cdot \text{F}} \times (61 \text{F} + 16\text{F}) = 8072 \text{ Btu/hr} \]

\[ Q_l = \dot{m} h_{fg} \] (latent heat transfer)

\[ Q_l = .0057 \text{ lbm W} \times 436.8 \text{ lbm d. a.} \times 1075 \text{ Btu/hr} \cdot \text{lbm} \]

\[ = 2676 \text{ Btu/hr} \]

\[ Q_T = Q_s + Q_l \] (total heat transfer)

\[ Q_T = (8072 + 2676) \text{ Btu/hr} = 10748 \text{ Btu/hr} \]

On the fin side, the heat transferred is:

\[ Q_s = 2796.81 \text{ lbm} \times \frac{.24 \text{ Btu}}{\text{hr} \cdot \text{lbm} \cdot \text{F}} \times 13.5\text{F} = 9062 \text{ Btu/hr} \]

\[ Q_l = 2796.81 \text{ lbm} \times \frac{.00005 \text{ lbm W}}{\text{hr} \cdot \text{lbm} \cdot \text{d. a.}} \times 1050 \text{ Btu/hr} \cdot \text{lbm} \]

\[ = 1468 \text{ Btu/hr} \]

\[ Q_T = (9062 + 1468) \text{ Btu/hr} = 10530 \text{ Btu/hr} \]

The calculations based on fin side heat transfer agree very well with those based on tube side heat transfer.

Now, the test data is input to the computer program, and these results are to be compared against the test data. The input data file is shown in Figure III-3. The computer program printout is shown in Figure III-4.

A comparison of the significant output parameters between the test data and the program printout is shown in Table III-2. The temperature and heat transfer calculations show reasonably
INPUT
HEIGHT  =  15.0
WIDTH   =  25.0
DEPTH   =  6.0
HEPT    =  6
HPASS   =  1
TMIN    =  -46.0
TPIN    =  14.9
TATE    =  7.28
TTYPE   =  614 AIR
TINL    =  82.0
TPIN    =  14.9
PVOL    =  615.
P TYPE   =  614 AIR
TTLCE   =  7311
TOL    =  100.
INLIN   =  0.
TASIN   =  0.0152
EAT    =  67.
ENDLIN  =  0.
ENDSTN  =  0.

Figure III-3
Input Data File for Test Case Heat Exchanger
**Hvac Fin-and-Tube Heat Exchanger Analysis**

402 Secondary Heat Exchanger

<table>
<thead>
<tr>
<th></th>
<th>FIN SIDE</th>
<th>TUBE SIDE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Height (in)</td>
<td>15.00</td>
<td></td>
</tr>
<tr>
<td>Width (in)</td>
<td>25.00</td>
<td></td>
</tr>
<tr>
<td>Depth (in)</td>
<td>6.00</td>
<td></td>
</tr>
<tr>
<td>Number of Passes</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>Number of Fins per Inch</td>
<td>6</td>
<td></td>
</tr>
<tr>
<td>Inlet Temperature (F)</td>
<td>82.00</td>
<td>-16.00</td>
</tr>
<tr>
<td>Outlet Temperature (F)</td>
<td>63.99</td>
<td>46.95</td>
</tr>
<tr>
<td>Inlet Pressure (PSIA)</td>
<td>14.90</td>
<td>14.90</td>
</tr>
<tr>
<td>Flow Rate (Lb/Hr)</td>
<td>2765.37</td>
<td>436.80</td>
</tr>
<tr>
<td>Flow Rate (CFM)</td>
<td>615.8</td>
<td></td>
</tr>
<tr>
<td>Heat Transferred (Btu/HR)</td>
<td></td>
<td>10768.41</td>
</tr>
<tr>
<td>Effectiveness</td>
<td></td>
<td>57.52</td>
</tr>
</tbody>
</table>

**Fluid**

<table>
<thead>
<tr>
<th></th>
<th></th>
<th>AIR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure Drop (PSI)</td>
<td>0.002</td>
<td>0.062</td>
</tr>
<tr>
<td>Pressure Drop (IN H2O)</td>
<td>0.05</td>
<td>1.72</td>
</tr>
<tr>
<td>Inlet Relative Humidity</td>
<td>67.00</td>
<td>100.00</td>
</tr>
<tr>
<td>Inlet Humidity Ratio (Lb w/Lb D.A.)</td>
<td>0.01551</td>
<td>0.00033</td>
</tr>
<tr>
<td>Inlet Water (Lb/Lb D.A.)</td>
<td>0.00000</td>
<td>0.00000</td>
</tr>
<tr>
<td>Inlet Ice (Lb/Lb D.A.)</td>
<td>0.00000</td>
<td>0.01520</td>
</tr>
<tr>
<td>Outlet Relative Humidity</td>
<td>99.11</td>
<td>100.00</td>
</tr>
<tr>
<td>Outlet Humidity Ratio (Lb w/Lb D.A.)</td>
<td>0.0149</td>
<td>0.0067</td>
</tr>
<tr>
<td>Outlet Water (Lb/Lb D.A.)</td>
<td>0.0007</td>
<td>0.0088</td>
</tr>
<tr>
<td>Outlet Ice (Lb/Lb D.A.)</td>
<td>0.0000</td>
<td>0.0000</td>
</tr>
</tbody>
</table>

**Figure III-4**

Computer Program Printout for Test Case Heat Exchanger
<table>
<thead>
<tr>
<th></th>
<th>Test Data</th>
<th>Program Output</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature Out, Tube Side</td>
<td>61°F</td>
<td>47°F</td>
</tr>
<tr>
<td>Temperature Out, Fin Side</td>
<td>68.5°F</td>
<td>69°F</td>
</tr>
<tr>
<td>Heat Transferred</td>
<td>10639 Btu/hr</td>
<td>10829 Btu/hr</td>
</tr>
<tr>
<td>Pressure Drop, Tube Side</td>
<td>33 in. W</td>
<td>1.72 in. W</td>
</tr>
<tr>
<td>Relative Humidity Out, Tube Side</td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>Condensate Out, Tube Side</td>
<td>.0044 lbm/lbm</td>
<td>.0088 lbm/lbm</td>
</tr>
</tbody>
</table>
good agreement, indicating that the program may be relied upon to provide design information.

There is one area, however, that has a very large disagreement. The program predicts a tube side pressure drop in the core of 1.7 in. W. The measured test data shows a pressure drop of 33 in. W across the heat exchangers (including headers). A probable explanation is that the test loop was using a type of lubricant unsuited for the extreme low temperatures involved. The lubricant was thickening and collecting in the entrance header. This theory was borne out by a later test. With the heat exchangers mounted in a truck refrigeration unit and with the correct lubricant in the system, the pressure drop was measured at 5.5 in. W. Other conditions in the system were similar to the test loop conditions. This data comparison has pointed out an area where further design work needs to be done. The actual pressure drop through the heat exchanger is much higher than that calculated for the core. In a properly designed heat exchanger, the manifold losses should be small in comparison to core pressure losses. The conclusion to be reached is that the header design needs improvement. At this time, alternate header approaches are being pursued.
CHAPTER IV
HEAT EXCHANGER PARAMETRIC STUDIES

The heat exchanger analysis program in Chapter III opens up a path to better understanding of heat exchanger characteristics and limitations. Without an effective and easy to use analysis tool, many of the heat exchanger designer's questions go unanswered. The computer model of the heat exchanger offers an easy to use, quick, and inexpensive answer to any parameter change questions.

In this chapter, a parametric study will be conducted upon an arbitrarily chosen heat exchanger. The results will be graphed so that parametric changes can be easily seen.

The study will focus upon changes in fin density, mass flow rates, moisture content of heat transfer fluids, inlet temperature difference, physical size and number of passes.

The heat exchanger chosen for study is 16 in. high, 30 in. wide and 4 in. deep. It has 12 fins per inch and is single pass. The tube side inlet temperature is -20F and the fin side inlet is 85F. The air on each side is saturated and in addition has 0.01 lbm condensate per lbm dry air. The program output shows the heat transfer as 16172 Btu/hr and 64% effectiveness.

The conditions were purposely chosen for a low effectiveness so that the results of parametric changes can be more easily seen.
When operating at very high effectiveness, sometimes very large changes in a parameter are required to produce a significant change in effectiveness, especially in the direction of higher effectiveness.

The first parameter to be examined is fin density. The graph of Figure IV-1 shows that the fin density was varied from 4 fins/inch (fpi) to 14 fpi. Recall that the base design was 12 fpi.

An increase from 12 fpi to 14 fpi adds approximately 15% to the fin surface area. Yet the effectiveness only increased 0.1%. Similarly, a decrease from 12 fpi to 10 fpi, a 15% fin area change also, decreases the effectiveness by only 0.15%. Looking at the entire range of densities, a change from 4 fpi to 14 fpi, a 350% area change, increases effectiveness by less than 2%.

The reason that such large changes have such a minute effect on the heat exchanger performance can best be explained through a simple quantitative analysis. Recall the equation for overall conductance in a heat exchanger.

\[ \frac{1}{U_0} = \frac{1}{n_0, f h_f} + \frac{1}{(A_t/A_f) n_0, t h_t} \]

In this equation, the wall conductance has been ignored, which is usually a safe assumption with heat exchanger analysis. Assume the following as typical values for fin efficiency and film coef-

---

For this example, use the fin areas for 4 fpi and 14 fpi.

\[
\begin{aligned}
A_f, 4 &= 0.68913 \text{ ft}^2/\text{ft} \\
A_f, 14 &= 2.1567 \text{ ft}^2/\text{ft} \\
A_t &= 0.10210 \text{ ft}^2/\text{ft}
\end{aligned}
\]

Using these values, the conductances for 4 fpi and 14 fpi are:

\[
\begin{aligned}
U_{0,4} &= 1.34 \text{ Btu/hr ft}^2 \text{ F} \\
U_{0,14} &= 0.51 \text{ Btu/hr ft}^2 \text{ F}
\end{aligned}
\]

Then, in the equation for \(N_{tu}\) the following are obtained:

\[
\begin{aligned}
N_{tu} &= U_0 A_0 / C_{\text{min}} \\
N_{tu,4} &= 1.34 \times 0.68913 / C_{\text{min}} = 0.923 / C_{\text{min}} \\
N_{tu,14} &= 0.51 \times 2.1567 / C_{\text{min}} = 1.10 / C_{\text{min}}
\end{aligned}
\]

Since \(C_{\text{min}}\) is the same for both cases, \(N_{tu,14}\) is only 20% larger than \(N_{tu,4}\). Then, the change in effectiveness will depend on the values of \(C_{\text{min}}\) and \(R\). In a typical problem, \(N_{tu}\) might be from 2 to 5 which would mean that the 20% change in \(N_{tu}\) is about a 2% to 5% change in effectiveness.

Included in Figure IV-1 is a graph of fin side pressure drop vs. fin density. As would normally be expected, the pressure drop is highly dependent upon the fin density. Because of this, the pressure drop, rather than effectiveness, may dictate the fin dens-
ity. In the lower pressure drop range, less power is required to maintain a given flow rate. In this case, the lower fin densities appear to be a logical choice for the designer.

The next parameter to be examined is the tube side inlet temperature. More significantly, the temperature difference between the hot and cold outlets is examined. Figure IV-2 displays a graph of effectiveness and heat transfer vs. tube side inlet temperature and overall inlet temperature difference.

Inspection of Figure IV-2 reveals several interesting things. First, the heat transfer is an almost linear function of the temperature difference. There is, however, one discontinuity in the curve, at an inlet temperature of 32°F. This discontinuity is caused by the fact that on the left side of 32°F, the air stream condensate is ice, while on the right side it is liquid water. The magnitude of the discontinuity is equal to the energy required to melt the ice.

The effectiveness follows the same general trend as the heat transfer, but is more non-linear. The definition of effectiveness is:

\[ e = \frac{Q}{Q_{\text{max}}} \]

From the graph, \( Q \) is nearly linear, therefore, for effectiveness to be non-linear, then \( Q_{\text{max}} \) must be non-linear with respect to inlet temperature (or temperature difference). The relative humidity at the tube side inlet has remained at 100%, and the amount of condensate is 0.01 lbm/lbm dry air. This moisture in the fluid
stream is causing the non-linearity of $Q_{\text{max}}$. Examination of a psychrometric chart confirms the theory. This example points out that fluid stream latent effects must be accounted for.

Probing further into the latent effects, the moisture on each side of the heat exchanger was varied and the results graphed in Figures IV-3 and IV-4. Varying the moisture on the fin side, shown in Figure IV-3, produces an almost negligible change in either effectiveness or heat transfer. This case, along with the case previously shown in Figure IV-1, point out that the heat exchanger performance is dominated by the tube side.

On the tube side, however, the moisture changes have a definite effect upon performance. Figure IV-4 shows that effectiveness is an almost linear function of the moisture content. The curve relating heat transfer to moisture content has a discontinuity at 0.00015 lbm/lbm dry air. This is the saturation point for the air stream. With a non-saturated condition, the only heat transfer effects available are sensible, and therefore the curve is flat in this region.

Note that the effectiveness is highest at the low moisture condition. As previously stated, the overall film coefficient remains constant over the range of moisture contents. Now, assume that the latent effects can be grouped with sensible effects in a pseudo-specific heat. Therefore, the high moisture content air would have a higher pseudo-specific heat than the low moisture content air.
Recall the definition of Number of Transfer Units:

\[ N_{tu} = \frac{U A}{C_{\text{min}}} \]

In this case, the tube side fluid is the minimum fluid. As the moisture content gets smaller, \( C_{\text{min}} \) gets smaller and \( U A \) remains constant, therefore \( N_{tu} \) must get larger. For a crossflow heat exchanger, as \( N_{tu} \) increases, effectiveness increases. And, the graph of Figure IV-4 behaves as predicted.

The next pair of variables being studied are the mass flow rates of the fin side and the tube side, Figures IV-5 and IV-6, respectively. Examination of the graphs reveals an interesting phenomena. On the fin side, as mass flow rate increases, the effectiveness increases. While, on the tube side, as mass flow rate increases, the effectiveness decreases.

As in the previous problem, the tube side fluid is the minimum fluid. This means that the only way the fin side fluid can enter into the equation for \( N_{tu} \) is in the overall film coefficient. The higher mass flow rates will increase the fin side film coefficient which will in turn raise (slightly) the overall film coefficient and therefore raise the \( N_{tu} \) and effectiveness.

On the other hand, the tube side mass flow rate enters the \( N_{tu} \) equation in two places. First, the overall film coefficient is affected as above. And second, the heat capacity \( (C_{\text{min}}) \) term is increased with increasing mass flow. Since one term in the numerator and one term in the denominator are affected, what happens
Figure IV-5

Effectiveness and Pressure Drop vs Fin Side Mass Flow Rate
to $N_{tu}$? This is answered by looking at the magnitude of the change in each term for a given change in mass flow. For example, if the mass flow rate were doubled, the overall film coefficient would increase, but not double. And, the heat capacity would be doubled. Therefore, the $N_{tu}$ would decrease with an increase in tube side mass flow, resulting in a drop of effectiveness.

Also shown on these two graphs is the pressure drop vs. flow rate curve. Both graphs show a smooth, almost linear increase in pressure drop with increasing mass flow. This is as would be expected.

The previous examples have all used a constant size heat exchanger core. Now, the effect of dimensional changes will be examined. In Figure IV-7, the effectiveness and tube side pressure drop are graphed as a function of core height. The effect on the tube side of changing the height is to change the number of tubes being used. As the number of tubes increases, the tube side flow velocity and Reynolds number decreases. But, in spite of the lower Reynolds number, there is enough additional surface area that the net result is an increase in effectiveness. Also, note that the pressure drop through the tube side increases rapidly as the height decreases.

In Figure IV-8, the overall core width has been varied. The effectiveness can be seen to behave in a smooth, nearly linear fashion with respect to core length. As expected, the effectiveness and pressure drop both increase with increasing core length.
The final parameter to be examined is the number of passes. This example was run as a 2 pass heat exchanger and compared to the results of the single pass case. As shown on Figure IV-9, the effectiveness increased quite substantially, from 64% for the single pass to 88% for the 2 pass. Unfortunately, the pressure loss also increases very considerably. The additional effectiveness with the 2 pass case is due to the very long flow length in the tubes. The tube side flow travels through a 60 in. active length in the core.

The various studies detailed above are a good example of the kinds of information available using the computer model of the heat exchanger. These studies are useful in the design of systems, such as the ROVAC Truck Refrigeration System. They also are very useful in the optimization of the heat exchanger itself. For example, the above studies showed the heat exchanger to be tube side limited. With this information, the heat exchanger designer can spend his time searching for ways to increase the tube performance, rather than blindly grasping for the right key to higher heat exchanger effectiveness.
CHAPTER V
CONCLUSIONS

At the inception of this project, the goal was to create an analysis tool for use in designing a fin and tube heat exchanger with moist air as the heat transfer fluid on one or both sides. The particular heat exchanger modelled is a unit manufactured by Sundstrand Heat Transfer and uses tubes with turbulators in them.

The primary output of this project is, therefore, a computer model of a heat exchanger. However, there are a number of secondary, yet very important results. First, the tube testing of Chapter II has provided an analytical description of the tube with turbulator. The value of this analytical description is that it quantifies the contribution of the turbulence enhancer and it provides an opportunity to model other types of heat exchangers using this tube, e.g. tube and shell heat exchangers.

Secondly, the test results of a heat exchanger and their comparison to the computer model output verified a method of modelling heat transfer to a flow stream containing moist air. The technique of dividing the heat exchanger into small sections and summing the results from each section (analogous to the mathematical technique of integration), was shown to be reasonably accurate. Also, this technique is easy to use, simple to model, and uses the basics of conventional one-phase heat transfer fluid analysis methods.
Another important result of this project is that it has pointed out two problem areas. The first problem area, which surfaced in the comparison between computer output and test results in Chapter III, is the pressure drop in the headers. The present design is apparently very inadequate with respect to pressure drop. Without having the tube test results and the computer analysis for the heat exchanger core, the true magnitude of the header deficiency would have gone unnoticed. But, as a result of this project, new and more efficient header designs are being developed as replacements.

The other problem area uncovered by this project is that the heat exchanger is, in many instances, tube side limited. This was pointed out by the parametric studies of Chapter IV. This limitation becomes more acute as the heat capacity of the tube side fluid stream becomes large, or more precisely, as the specific heat or pseudo-specific heat gets large. The value of this revelation lies in providing the heat exchanger designer information on the weak areas of the heat exchanger. It is not necessary to guess at the areas which need improvement. In all likelihood, the next generation of heat exchangers to evolve from this design will have tubes with a higher film coefficient, or larger surface area, or both.

The computer model is particularly well adapted to providing insight into cause and effect relationships as evidenced by the parametric studies of Chapter IV. The designer can easily and inexpensively find out the parameters to be adjusted to raise effectiveness, lower pressure drop, etc. In addition to answering
specific questions about the effect of parameter changes, these studies also serve to increase the overall understanding and "feel" for the heat transfer process inside the heat exchanger.

Mention has been made several times that the computer model is inexpensive to execute. A quantitative example is available from the parametric studies. In one instance, 19 separate cases were run. The total computer time was 280 system seconds at a cost of $50.33. This means the average cost for each case was only $2.65.

In conclusion, this project has succeeded in not only delivering a very useful and easy to use analysis tool, but has also contributed to better understanding of the heat exchange process when using moist air, and has contributed to the general data base available for heat exchanger design.
APPENDIX A

COMPUTER PROGRAM LISTING
PROGRAM HFS-FINTUBE.

COMMON/ONE/FVOL,FPIN,FMU,FK,FCP,FPR,FRH0,FDH,GFMFA,FAO,FAF,
1 FG,FRE,FH,FST
COMMON/TWO/TPIN,TMU,TK,TCP,TPR,TRH0,TDH,TRATE,INU,TRE,TG
COMMON/THREE/HEIGHT,DEPTH,NFPI
COMMON/FOUR/FRATE

DATA TDH/.02933/,PSIH20/27.684/
DATA ALK/111./,FL/0.35/
DIMENSION TITLE(13),WW(3,2,2),TTI(100),TTO(100),FTI(100),FTO(100)
DIMENSION W(3,2,2,100)

NAMELIST/INPUT/HEIGHT,WIDTH,DEPTH,NFPI,NPASS,TTIN,TPIN,TRATE,
1 TTYPE,FTIN,FPIN,FVOL,FTYPE,TITLE,TRH,TLIN,TWIN,FRH,
2 FWLIN,FWIN
KEYI=0

C
SET ALL NAMELIST VARIABLES TO ZERO PRIOR TO THE READ STATEMENT.

6 HEIGHT=0.
WIDTH=0.
DEPTH=0.
NFPI=0
NPASS=0
TTIN=0.
TPIN=0.
TRATE=0.
TTYPE=''
FTIN=0.
FPIN=0.
FVOL=0.
FTYPE=''

DO I=1,13
1 TITLE(I)=

34: C READ NAMELIST INPUT
35: C READ(8,INPUT,END=2)
36: C CHECK THE INPUT FOR OMISSIONS.
37: C IF(HEIGHT.EQ.0.) CALL ERROR('HEIGHT')
38: C IF(WIDTH.EQ.0.) CALL ERROR('WIDTH')
39: C IF(DEPTH.EQ.0.) CALL ERROR('DEPTH')
40: C IF(NPASS.EQ.0.) CALL ERROR('NPASS')
41: C IF(NFPI.EQ.0.) CALL ERROR('NFPI')
42: C IF(TP.IN.EQ.0.) CALL ERROR('TPIN')
43: C IF(TRATE.EQ.0.) CALL ERROR('TRATE')
44: C IF(TYPE.EQ.' ') CALL ERROR('TYPE')
45: C IF(FTIN.EQ.0.) CALL ERROR('FTIN')
46: C IF(FVOL.EQ.0.) CALL ERROR('FVOL')
47: C IF(TYPE.EQ.' ') CALL ERROR('TYPE')
48: C TRATE=TRATE*60.
49: C STORE THE HUMIDITY RATIOS IN AN ARRAY, RH(I,J,K,L), WHERE
50: C I = 1, VAPOR
51: C = 2, LIQUID
52: C = 3, SOLID
53: C J = 1, ENTRANCE
54: C = 2, EXIT
55: C K = 1, TUBE SIDE
56: C = 2, FIN SIDE
57: C L = HX SECTION NO.
58: C TWIN=RHIN(FTIN,TPIN,TRH)
59: C FWIN=RHIN(FTIN,TPIN,FRH)
60: DO 12 L=1,100
67:     W(1,1,1,L)=TWIN
68:     W(2,1,1,L)=TWLIN
69:     W(3,1,1,L)=WSIN
70:     W(1,1,2,L)=FWWIN
71:     W(2,1,2,L)=FWLIN
72:     W(3,1,2,L)=FSWIN

73:     C
74:     PREDICT THE EXIT TEMPERATURES.
75:     C
76:     C
77:     C
78:     C
79:     C
80:     TTOUT=FTIN-0.75*(TTIN-FTIN)
81:     FTOUT=FTIN-0.05*(TTIN-FTIN)
82:     OK1=1.0
83:     IF(TTIN.GT.FTIN)OK1=-1.0
84:     DO 13 I=1,3
85:     DO 13 J=1,2
86:     DO 13 K=1,2
87:     13     W(I,J,K)=W(I,J,K,1)
88:     CALL HXFXH(TTIN,TTOUT,FTIN,FTOUT,WW,WIDTH,0,TPD,FPDW,NPASS,OK1,
89:              *     FRATE,EFF)
90:     C
91:     C
92:     C
93:     C
94:     SECS=INT(WIDTH/0.75+0.5)
95:     KSECS=INT(SECS)
96:     SFCW=WIDTH/SECS
97:     NSecs=KSECS*NPASS
98:     FDI=FTOUT-FTIN
99:     FDTS=FDT/FLOAT(NPASS)
100:   IDI=ITOUT-ITTIN
101:   TDTS=TDI/FLOAT(NSECS)
102:   DO 8 L=1,NSECS
103:   DIS=FLOAT(L-1)
104:   TTI(L)=TTIN+TDIS*DIS
105:   8 TTO(L)=TTI(L)+TDIS
106:   L=NSECS+1
107:   DO 9 I=1,NPASS
108:   DO 9 J=1,KSECS
109:   L=L-1
110:   DIS=FLOAT(I-1)
111:   FTTI(L)=FTIN+FTDIS*DIS
112: 9 FTTO(L)=FTTI(L)+FTDIS
113: C ANALYZE EACH SECTION OF THE HX, USING THE PREVIOUS SECTION OUTPUT
114: C AS THE NEW INPUT.
115: C
116: NCOUNT=0
117: FS_RATE=FRATE/SECS
118: 5 Q=0.0
119: KEY1=0
120: FTEMP=0.0
121: FWVOUT=0.0
122: FWLOUT=0.0
123: FWOUT=0.0
124: DO 3 L=1,NSECS
125: ITOUT=TTI(L)
126: FTOUT=FTT(L)
127: DO 14 I=1,3
128: DO 14 K=1,2
129: 14 WNE(I,1,K)=W(I,1,K,L)
130: CALL HXFER(TTI(L),TTOUT,FTI(L),FTOUT,WN,SECH,QS,D1,D1,1,0K1,
131:   * FS_RAT,EFF)
IF(ABS(TTOUT-FTO(L)).GT.0.001)KEY1=1
IF(ABS(FTOUT-FTO(L)).GT.0.001)KEY1=1
0=Q+QS

TT0(L)=TTOUT
FT0(L)=FTOUT
TTI(L+1)=TT0(L)
DO 16 I=1,3
I(W(I,1,1,L+1)=W(I,2,1)
DO 16 K=1,2
W(I,2,K,L)=W(I,2,K)
NXTRW=(L-1)/KSECS
IF(NXTRW.LE.0)GO TO 11
NXSEC=(NXTRW*KSECS)-(L-1-NXTRW*KSECS)
FTI(NXTSEC)=FT0(L)
DO 15 I=1,3
W(I,1,2,NXTSEC)=W(I,2,2)
GO TO 3
IF(FTP=FTP+FT0(L)
FWOUT=FWOUT+W(I,2,2)/SECS
FWLOUT=FWLOUT+W(I,2,2)/SECS
FWSOUT=FWSOUT+W(I,2,2)/SECS
3 CONTINUE
NCOUNT=NCOUNT+1
IF(NCOUNT.GE.50)CALL ERROR('NCOUNT')
IF(KEY1.EQ.1)GO TO 5
TPD=TPD*PSI20
FPD=FPD/PSI20
FTOUT=FTOUT/SECS
TTOUT=TT0(NSEC)
TWOUT=W(1,2,1,NSEC)
TWLOUT=W(2,2,1,NSEC)
WSOUT=W(3,2,1,NSEC)
TRHOUT=W(TTOUT,TPIN,TWOUT)
I
I
I
I
I
I
I
I
I
I
I
I
I
I
I
I
I
I
WRITE(6,108)FPIN,TPIN

108 FORMAT(TIO,'INLET PRESSURE (PSIA)',15(','),2(F10.2,5X))

WRITE(6,109)FRATE,TRATE

109 FORMAT(TIO,'FLOW RATE (LBM/HR)',18(','),2(F10.2,5X))

WRITE(6,110)EVOL

110 FORMAT(TIO,'FLOW RATE (CFM)',21(','),F10.0)

WRITE(6,111)Q

111 FORMAT(TIO,'HEAT TRANSFERRED (BTU/HR)',11(','),T54,F10.2)

WRITE(6,115)EFF

115 FORMAT(TIO,'EFFECTIVENESS',23(','),T54,F10.2)

WRITE(6,112)FSTYPE,TSTYPE

112 FORMAT(TIO,'FLUID',31(','),T50,2(A6,9X))

WRITE(6,113)FPD,TPD

113 FORMAT(TIO,'PRESSURE DROP (PSI)',17(','),2(F10.3,5X))

WRITE(6,114)FPDW,TPDW

114 FORMAT(TIO,'PRESSURE DROP (IN H2O)',14(','),2(F10.2,5X))

WRITE(6,134)FRH,TRH

134 FORMAT(TIO,'INLET RELATIVE HUMIDITY',13(','),2(F10.2,5X))

WRITE(6,128)FWVIN,TWVIN

128 FORMAT(TIO,'INLET HUMIDITY RATIO (LB W/LB D.A.)',',',2(F10.5,5X))

WRITE(6,129)FWLIN,TWLIN

129 FORMAT(TIO,'INLET WATER (LB/LB D.A.)',12(','),2(F10.5,5X))

WRITE(6,130)FWSIN,TWSIN

130 FORMAT(TIO,'INLET ICE (LB/LB D.A.)',14(','),2(F10.5,5X))

WRITE(6,135)FRHOUT,TRHOUT

135 FORMAT(TIO,'OUTLET RELATIVE HUMIDITY',12(','),2(F10.2,5X))

WRITE(6,131)FWVOUT,TWVOUT

131 FORMAT(TIO,'OUTLET HUMIDITY RATIO (LB W/LB D.A.)',2(F10.4,5X))

WRITE(6,132)FWLOUT,TWLOUT

132 FORMAT(TIO,'OUTLET WATER (LB/LB D.A.)',11(','),2(F10.4,5X))

WRITE(6,133)FWSOUT,TWSOUT

133 FORMAT(TIO,'OUTLET ICE (LB/LB D.A.)',13(','),2(F10.4,5X))

GO TO 6
FUNCTION CK2(X)
CONVERG A FROM KJ/KG TO JU/LBM
CK2=A*.430002
RETURN
END
FUNCTION CK3(A)
C
C
C
C
CK3=A*.10132
RETURN
END
SUBROUTINE EFFNTU(XNTU, R, NPASS, EFF)

THE PURPOSE OF THIS ROUTINE IS TO CALCULATE CROSS-COUNTERFLOW
HEAT EXCHANGER EFFECTIVENESS WHEN NTU IS SPECIFIED.

FIRST, FIND THE EFFECTIVENESS PER PASS.

KEY5=0

EFF1=0.

EFF2=1.

EFF3=1.

DO 2 N=1,6

EFF1=(1.-EXP(-XNTU)*EFF2)*(1.-EXP(-R*XNTU)*EFF3)*EFF1

IF(N.EQ.3) GO TO 4

EFF2=1.

FAC=1.

DO 3 K=1,N

FAC=FAC*FLOAT(K)

3 EFF2=XNTU*FLOAT(K)/FAC+EFF2

EFF3=1.

FAC=1.

DO 2 K=1,N

FAC=FAC*FLOAT(K)

2 EFF3=(R*XNTU)*FLOAT(K)/FAC+EFF3

4 EFF0=(1./(R-A-XTU))**EFF1

NEXT, FIND THE OVERALL EFFECTIVENESS.

PASS=FLOAT(NPASS)

IF(R.GT.0.99) GO TO 4

EFF=((1.-EFF0*R)/(1.-EFF0))**PASS

EFF=(EFF-1.)/(EFF-R)

RETURN
34: \[ \text{EFF} = \text{PASS} \times \text{EFF0} / (1. + \text{PASS} - 1) \times \text{EFF0} \]
35:     \text{RETURN}
36:     \text{END}
SUBROUTINE ENTHL(TT,HSL)
C THIS CALCULATES ENTHALPY OF LIQUID PHASE OF WATER
C REF: ASHRAE
C
DATA A/17.6011/,B/-147914/,C/6.08619E-04/
DATA D/-1.11867E-06/,E/7.80297E-10/
DATA TA/273.16/,THI/350.0/
C IF (TI .GT. THI) GO TO 10
TTK=FK(TT)
C HSL=A*(TTK-TA)+(B*(TTK*TTK-TA*TA)/2.0)
HSL=HSL+C*((TTK**3.)-(TA**3.))/3.0
HSL=HSL+D*((TTK**4.)-(TA**4.))/4.0
HSL=HSL+E*((TTK**5.)-(TA**5.))/5.0
HSL=CK2(HSL)
GO TO 20
10 CONTINUE
C WRITE(6,100) TT
100 FORMAT(' ',I,10X,'TI=',E8.2,10X,'TEMPERATURE OUT OF RANGE FOR', 'LIQUID ENTHALPY EVALUATION - EXECUTION TERMINATED')
HSL=-9999.99
20 CONTINUE
RETURN
END
SUBROUTINE ENTHS(TT,HSS)
C THIS CALCULATES ENTHALPY OF SOLID PHASE
C OF WATER
C REF: ASHRAE
C DATA AO/-158.9285936/,A1/0.471247296/,A2/0.0004882881/
C DATA TLOW/-40.001/,THI/32.001/
C IF ((TT .LT. TLOW) .OR. (TT .GT. THI)) GO TO 10
C HSS=AO+A1*TT+A2*TT**2
GO TO 20
CONTINUE
C WRITE(6,100) TT
C 100 FORMAT('O',10X,'TT=',F8.2,10X,'TEMPERATURE OUT OF RANGE FOR', 2
1 'SOLID ENTHALPY EVALUATION - EXECUTION TERMINATING')
C HSS=-9999.99
CONTINUE
RETURN
END
SUBROUTINE ENTHV(TT,PS,HSV)

CALCULATE ENTHALPY OF VAPOUR PHASE OF WATER

REF: EOH 14, PG IS KEENAN AND KEYES

DOUBLE PRECISION DLOG,T2,A9

DATA A2/1.89/,A3/-2641.62/,A4/82.545/,A5/-1.6246E05/

DATA A6/-1.21828/,A7/-1.2697E05/,A8/3.635E-04/

DATA A1LOG/186210.0565/

DATA A9/-6.768064/

DATA C1/1.4720/,C2/1.5566E-04/,C3/47.8365/,C4/2502.36/

DATA TAA/273.1600/

DATA TII/645.0/

IF (TT ,GT. TII) GO TO 10

TTK=FK(TT)

CALCULATE F COEFFICIENTS (VAPOUR PHASE)

T=1./TTK

T2=T*T

T3=T**3.

T4=T**4.

T5=T**5.

T6=T**6.

T12=T**12.

T13=T**13.

T14=T**14.

A1T2=80870.0*T2

A1T2=10.0**A1T2

FO=A2+(2.0*A3*T*A1T2)

FO=F0+(2.0*A3*A1LOG*T3*A1T2)
34: \[ FA = (A_2 + A_3 \cdot A_1 T_2 \cdot T) \]
35: \[ FB = (1.0 + 2.0 \cdot T_2 \cdot A_{1 \log}) \]
36: \[ F_1A = FA \cdot FA \cdot (3.0 \cdot A_4 \cdot T_2 + 4.0 \cdot A_5 \cdot T_3) \]
37: \[ F_1B = FB \cdot (A_4 \cdot T_3 + A_5 \cdot T_4) \cdot FA \]
38: \[ F_1B = 2.0 \cdot A_3 \cdot A_1 T_2 \cdot F_1B \]
39: \[ F_1 = F_1A + F_1B \]
40: \[ C \quad F_3A = (FA \cdot FA) \cdot 2. \]
41: \[ F_3A = F_3A \cdot (4.0 \cdot A_6 \cdot T_3 + 6.0 \cdot A_7 \cdot T_5) \]
42: \[ F_3B = 4.0 \cdot A_3 \cdot A_1 T_2 \cdot FA \cdot FA \cdot FA \]
43: \[ F_3B = F_3B \cdot (A_6 \cdot T_4 + A_7 \cdot T_6) \cdot FB \]
44: \[ F_3 = F_3A + F_3B \]
45: \[ C \quad F_12A = (((FA \cdot FA) \cdot 6.) \cdot FA \]
46: \[ F_12A = F_12A \cdot (13.0 \cdot A_8 \cdot T_12 + 37.0 \cdot A_9 \cdot T_18 \cdot T_18) \]
47: \[ F_12B = F_12B \cdot (FA \cdot FA) \cdot 6. \]
48: \[ F_12B = 13.0 \cdot A_3 \cdot A_1 T_2 \cdot F_12B \]
49: \[ F_12B = F_12B \cdot (A_8 \cdot T_13 + A_9 \cdot T_18 \cdot T_18 \cdot T) \cdot FB \]
50: \[ F_12 = - (F_12A + F_12B) \]
51: \[ C \quad PSATM = PATM (PS) \]
52: \[ HSV = F_0 \cdot PSATM + (F_1 \cdot PSATM \cdot PSATM / 2.0) \]
53: \[ HSV = HSV + (F_3 \cdot (PSATM \cdot 4.) / 4.0) \]
54: \[ HSV = HSV + (F_12 \cdot (PSATM \cdot 13.) / 13.0) \]
55: \[ HSV = CK_3 (HSV) \]
56: \[ C \quad FPRIME = C_1 \cdot (T1K - TA) + (C_2 \cdot ((T1K \cdot 2.0) - (TA \cdot 2.0)) / 2.) \]
57: \[ FPRIME = FPRIME + C_3 \cdot DLOG (T1K / TA) + C_4 \]
58: \[ HSV = HSV + FPRIME \]
59: \[ HSV = CK_2 (HSV) \]
60: \[ C \quad GO TO 20 \]
61: \[ IO \quad CONTINUE \]
67: WRITE(6,100) TI
68: 100 FORMAT('0',10X,'TI=' ,F8.2,10X,'TEMPERATURE OUT OF RANGE FOR',
69: ' VAPOR ENTHALPY EVALUATION - EXECUTION TERMINATING')
70: HSV=-9999.99
71: 20 CONTINUE
72: RETURN
73: END
SUBROUTINE ERROR(NAME)

THIS ROUTINE STOPS THE PROGRAM WHEN AN ERROR HAS BEEN DETECTED.

IF(NAME.NE.'NCOUNT')GO TO 2
WRITE(6,3)
3 FORMAT(/,10X,'* * MAX ITERATIONS EXCEEDED IN MAIN ROUTINE * *',
      /,14X,'PROGRAM EXECUTION STOPPED')
STOP
2 IF(NAME.NE.'NCL')GO TO 4
WRITE(6,5)
5 FORMAT(/,10X,'* * MAX ITERATIONS EXCEEDED IN TWOUT ROUTINE * *',
      /,14X,'PROGRAM EXECUTION STOPPED')
STOP
4 IF(NAME.NE.'NSAT')GO TO 6
WRITE(6,7)
7 FORMAT(/,10X,'* * MAX ITERATIONS EXCEEDED IN TSAT ROUTINE * *',
      /,14X,'PROGRAM EXECUTION STOPPED')
STOP
6 WRITE(6,1)NAME
1 FORMAT(/,10X,'* * INPUT ERROR * *',/10X,'CHECK VARIABLE ',A6)
STOP
END
FUNCTION F(Z1,Z2,TIK)

C
C
C
C

F=Z2-(Z1/TIK)
F=EXP(F)
RETURN
END
FUNCTION FC(T) CONVERT FAHRENHEIT TO CENTIGRADE
FC=(T-32.0)/1.8
RETURN
END
SUBROUTINE FCOEF(FTAVG, WIDTH, N)

C THE PURPOSE OF THIS ROUTINE IS TO CALCULATE THE FAN SIDE FILM
C COEFFICIENT OF A FIN AND TUBE HE.

COMMON/ONE/EVOL, EPIN, F1U, FK, FCP, FPR, FRHO, FDH, GFAMFA, FAO, FAF,
    FG, FREQ, F1, FST
COMMON/THREE/HEIGHT, DEPTH, NFPI
COMMON/FOUR/FRATE
DATA A/0.18756/, B/0.41504/

C FIND THE FLUID PROPERTIES.
CALL PPROP(FTAVG, EPIN, FRHO, FCP, F1U, FK, FPR, 1)
FCPI = SPCHET(FTAVG, EPIN, A, 1, 2)
FCPO = SPCHET(FTAVG, EPIN, A, 2, 2)
FCP = (FCPI + FCPO) / 2.

C GET THE GEOMETRIC PARAMETERS OF THE FIN SURFACE.
CALL FINGEO(NFPI, FDH, GFAMFA, FAO, FAF)

C CALCULATE THE FLOW PER UNIT AREA, FG (LRM/HR•FT•2)
FRATE = EVOL•FRHO•60.
FG = FRATE / (GFAMFA•HEIGHT•WIDTH)•144.

C THE REYNOLDS NUMBER IS:
FRE = FG•FDH/F1U
WRITE(6, 10) FG, FDH, F1U, FRE
10 FORMAT(5X, 'FG = ', F10.2, 5X, 'FDH = ', F10.5, 5X, 'F1U = ', F10.4,
     *5X, 'FRE = ', F10.2)
34: C
35: C THE STANTON NUMBER IS:
36: C
37: C FST=A/(FRE**B*FPR**(2./3.))
38: C WRITE(6,30)FPR
39: C 30 FORMAT(5X,'FPR = ',F10.5)
40: C
41: C THE FILM COEFFICIENT IS: H=ST*CP*G
42: C
43: C FH=FST*FCP*FG
44: C WRITE(6,20)FST,FCP,FH
45: C 20 FORMAT(5X,'FST = ',F10.5,5X,'FCP = ',F10.5,5X,'FH = ',F10.2)
46: RETURN
47: END
SUBROUTINE FPDP(RE,WIDTH,FF,PD,G,RHO,DH)

C THE PURPOSE OF THIS ROUTINE IS TO CALCULATE THE FANNING FRICTION
C FACTOR AND THE PRESSURE DROP THROUGH THE TUBE SIDE OF A HEAT
C EXCHANGER USING SUNDSTRAND 3/8 DIA TUBES WITH TURBULATORS.

C CALCULATE FRICTION FACTOR

3 FF = 16./RE

10: IF(RE.LE.1500.)GO TO 1

11: FF = .2344/(RE**0.19)

C CALCULATE PRESSURE DROP

15: PD=4.*FF*(WIDTH/DH)*G**2/(2.*RHO*32.*2.*2.239488E10)

16: RETURN

17: END
SUBROUTINE FINGEO(NFPI,FDH,GFAMFA,FAO,FAF)
C THE PURPOSE OF THIS ROUTINE IS TO CALCULATE FIN GEOMETRIC
C PARAMETERS FOR SUNDSTRAND FINS. THE PARAMETERS ARE:
C NFPI = NUMBER OF FINS PER INCH (INPUT VARIABLE)
C FDH = HYDRAULIC DIAMETER (FT)
C GFAMFA = MEAN FREE FLOW AREA / GROSS FACE AREA
C FAO = OUTSIDE SURFACE AREA OF TUBE AND FIN (FT**2/FT)
C FAF = OUTSIDE SURFACE AREA OF FIN (FT**2/FT)
C
DATA TUBEO0/.390/,FINTHK/.0055/,TUBSEP/.610/,PI/3.14159/
FPI=FLOAT(NFPI)
C FIN AREA, FAF
FAF=2.*1.00*1.00*FPI/12.
FAF=FAF-2.*PI/4.*TUBEO0**2*FPI/12.
C TOTAL OUTSIDE AREA, FAO
FAO=FAF+PI*TUBEO0*(1.-FINTHK*FPI)/12.
C MEAN FREE FLOW AREA / GROSS FACE AREA, GFAMFA
GFAMFA=((1./FPI)-FINTHK)*TUBSEP*FPI/1.00
C HYDRAULIC DIAMETER
FDH=4.*((1./FPI-FINTHK)*TUBSEP/12.
FDH=FDH/(2.*(1./FPI-FINTHK+TUBSEP))
RETURN
END
FUNCTION FK(T)
CONVERT FARRENHEIT TO KELVIN
FK=F(T)+273.16
RETURN
END
FUNCTION FR(T)
CONVERT FARRENHEIT TO RANKINE
FR=T*459.67
RETURN
END
FUNCTION HFG(T,P,W,J,K)

C                          CALCULATE THE HEAT OF VAPORIZATION.

C                          DIMENSION W(3,2,2)

PH2O=PW(P,W(1,J,K))

CALL ENTHV(T,PH2O,HG)

CALL ENTHL(T,HF)

HFG=HG-HF

RETURN

END
FUNCTION HR(PTOT, PW)

C FIND HUMIDITY RATIO GIVEN TOTAL PRESSURE AND VAPOR PRESSURE

C

HR=0.622*(PW/(PTOT-PW))

RETURN

END
FUNCTION HSG(T, P, W, J, K)

CALCULATE THE HEAT OF SUBLIMATION.

DIMENSION W(3,2,2)

PH2O = PM(W(1,1,1), J, K))

CALL ENTHY(T, PH2O, HG)

CALL ENTHS(T, HS)

HSG = HG - HS

RETURN

END
SUBROUTINE HXFER (FIN, TTOUT, FTIN, FTOUT, WIDTH, Q, TPDP, FPDP, HPASS, 
*     OKI, RATE, EFF)

THE PURPOSE OF THIS ROUTINE IS TO CALCULATE THE HEAT TRANSFER 
AND THE EXIT TEMPERATURES AND MOISTURE CONTENTS OF A HEAT 
EXCHANGER SECTION.

COMMON/ONE/FVOL, FPIN, FMU, FK, FCP, FPR, FRHO, FDH, GFA, FAO, FAP, 
    FC, FME, FH, EST
COMMON/TWO/IPIN, TMU, TK, TCP, TRP, TRHO, TDH, TRATE, TNU, TNE, TG
COMMON/THREE/HEIGHT, DEPTH, NFPI
DIMENSION W(3,2,2)
DATA ALK/111./,FL/0.50/, FAT/.1021/
TIAVG=0.5*(TTIN+TTOUT)
CALL TCOEF(TIAVG, TH, WIDTH, W, K, HPASS)

CALCULATE THE FRICTION FACTOR AND PRESSURE DROP.

CALL FFPD(TRE, WIDTH, TF, TPDP, TG, TRHO, TDH)

CALCULATE THE FAN SIDE FILM COEFFICIENT.

FIRST, ASSUME AN EXIT TEMPERATURE (FTOUT)

FIAVG=0.5*(FTIN+FTOUT)
CALL FCOEF(FIAVG, WIDTH, W)

FIND THE PRESSURE DROP ON THE FAN SIDE.

CALL PDO(FVOL, FPDP, WIDTH)

CALCULATE FIN EFFICIENCY (ETAO).
DELTA=.0055/12.
XM=SQRT((2.*FH)/(ALK*DELTA))
Xn=XM*FL/12.

ETAf=TANH(Xn)/Xn
ETAo=1.-((FAo/FAo)*(1.-ETAf))

C CALCULATE THE OVERALL COEFFICIENT, UO
UO=1./((1./(ETAo*FH))+(1./((FAo/FAo)*TH)))
C CALCULATE FIN AREA PER PASS, FAP
FAP=FAo*(WIDTH/12.)*(HEIGHT/1.0)*(DEPTH/1.0)/FLOAT(NPASS)

C FIND THE MINIMUM FLUID
FC=RATE*FCP
TC=THATE*TCP
CMIN=MIN(MIN(FC,TC))

C THE NUMBER OF TRANSFER UNITS PER PASS, NTU, IS UO*AO/CMIN.
XNTU=UO*A0*FAP/CMIN

C DETERMINE THE EFFECTIVENESS.
R=CMIN/CMAX
CALL EFFNTU(XNTU,R,NPASS,EFF)

C THE HEAT TRANSFERRED IS:
Q=CMIN*EFF*ABS(FIN-TTIN)*OKI
CALCULATE THE EXIT TEMPERATURES AND MOISTURE CONTENTS.

CALL TWOUT(TTIN,TTOUT,Q,W,TPIN,TRATE,1.,1)
CALL TVHOUT(TTIN,TTOUT,Q,W,FPIN,FRATE,-1.,2)

KY=1

IF (FC.LT.TC) KY=2

OM=0

GO TO (1,2),KY

P=FPIN
RATE=FRATE
GO TO 3

P=FPIN
RATE=FRATE

CALL QMAX(TTIN,FTIN,QM,W,P,RATE,KY)
RETURN
END
FUNCTION PATM(P)
CONVERT PSI TO ATMOSPHERES
PATM=P/14.696
RETURN
END
FUNCTION PSI(P)

C CONVERT ATMOSPHERES TO PSI

C

PSI=P*14.696

RETURN

END
SUBROUTINE PDO(FVOL,FPDN,WIDTH)
C
THE PURPOSE OF THIS ROUTINE IS TO CALCULATE THE PRESSURE DROP ON
THE FAN SIDE OF A CORE USING SUNDSTRAND FINS.
C
THE METHOD USED IS A CURVE FIT THROUGH THE SUNDSTRAND SUPPLIED
DATA.
C
DIMENSION A(6),B(6),C(3)
COMMON/THREE/HEIGHT,DEPTH,NFPI
DATA A/-5.57493,-5.28085,-5.21758,-5.09944,-4.96011,-4.76008/, 
* B/1.67811,1.62323,1.63914,1.62586,1.61364,1.57462/, 
* C/-0.24643,0.15384,-0.00406/
C
FIND THE GROSS FACE AREA (GFA). (INCHES**2)
C
GFA=HEIGHT*WIDTH
C
CALCULATE THE FACE VELOCITY (FV). (FT/MIN)
C
FV=FVOL*144./GFA
C
CALCULATE THE NUMBER OF ROWS OF TUBES IN THE CORE.
C
ROWS=DEPTH/1.00
ROWS=AINT(ROWS)
IF((ROWS * 1.00).EQ.DEPTH)GO TO 1
C
ERROR TERMINATION - INCORRECT FIN DEPTH
C
WRITE(6,2)DEPTH
2 FORMAT(10X,'** ERROR TERMINATION **',/10X,'THE FIN DEPTH GIVEN
*IS ','F6.3,' THIS IS NOT A MULTIPLE OF 1.00')
STOP

THE PRESSURE DROP IS GIVEN BY THE EQUATION:

\[
\log(\text{FPDW}) = A \cdot B \cdot \log(\text{FV})
\]

1 NIROW=IFIX(ROWS)

FPDW = 10.**A(NIROW) \cdot 10.**(B(NIROW)*ALOG10(FV))

INCLUDE THE FIN DENSITY CORRECTION FACTOR (CF) FOR FIN DENSITIES OTHER THAN 12 FPI.

\[
\text{IF(NFPI,LT.4.,OR.NFPI,GT.14)GO TO 4}
\]

FPI=NFPI

CF=0.

DO 3 I=1,3

J=I-1

3 CF=CF+C(I)*FPI**J

FPDW=FPDW*CF

RETURN

ERROR TERMINATION - INCORRECT FIN DENSITY

WRITE(6,5)NFPI

FORMAT(IOX,"** ERROR TERMINATION **",/IOX,"FIN DENSITY OF \( \cdot \),I2,"** IS OUT OF RANGE")

STOP

END
SUBROUTINE PPROP(T,P,RHO,CP,VIS,COND,PR,ITYPE)

THE PURPOSE OF THIS ROUTINE IS TO CALCULATE THE FLUID PROPERTIES FOR AIR AND WATER.

THE VARIABLES ARE:
RHO = DENSITY
CP = SPECIFIC HEAT
VIS = VISCOSITY
COND = THERMAL CONDUCTIVITY
PR = PRANDTL NUMBER

ITYPE = 1:  AIR
2: WATER VAPOR
3: LIQUID WATER
4: ICE

GO TO 10 (1,5,6,7),ITYPE

********************

FLUID PROPERTIES FOR AIR

DENSITY

1 RHO=(P/((T+459.67)*53.342))*144.

VISCOSITY

T= T-167.
VIS=0.049089

1F(T,T,E0.0)GO TO 2

VIS = 0.049089 + 0.53546E-4*T*T - 0.17133E-7*T**2 + 0.51093E-10 * T**3 - 0.42132E-14*T**4

SPECIFIC HEAT
2 T=T-392.
CP=0.24471

IF (T.T.EQ.0.160 10 3
CP = 0.24471 + 0.21778E-4*T + 0.21166E-7*T**2

C THERMAL CONDUCTIVITY

3 T=T-137.
COND=0.15994E-1

IF (T.T.EQ.0.160 10 4
COND = 0.15994E-1 + 0.22325E-4*T + 0.75668E-7*T**2
* 0.32434E-10*T**3 - 0.22635E-12*T**4

C PRANDTL NUMBER

4 PR=VIS*CP/COND
RETURN

C ************************************************************

C FLUID PROPERTIES FOR WATER VAPOR

C SPECIFIC HEAT

5 CP=0.2246E-1

IF (T.T.EQ.0.160 10 8
CP=CP-2.5446E-4*T+3.0371E-7*T**2
RETURN

C ************************************************************

C FLUID PROPERTIES FOR LIQUID WATER
C SPECIFIC HEAT

   C CP=1.0140
   IF(T.EQ.0.0)GO TO 9
   CP=CP-3.5991E-4*T+1.9540E-6*T**2
   9 RETURN

C**************************************************************

C FLUID PROPERTIES FOR ICE

C SPECIFIC HEAT

   C CP=4.6752E-1
   IF(T.EQ.0.0)GO TO 10
   CP=CP+7.1027E-4*T-3.0313E-6*T**2
   10 RETURN

END
SUBROUTINE PSAT(IT,PT,PP,PS)
C THIS CALCULATES Vapor Pressure of water,Ps, and air,PP.
C REF: (1) EON 11 AND 12, PG 14 KEENAN AND KEYES
C (2) CURVE FIT FROM STEAM TABLE DATA
C
DATA A1,B1,B1,D1/3.2437814,5.86826E-03,1.1702379E-08,
     1 2.1876462E-03/
DATA A2,D2,D2,D2,E2/3.3463130,4.14113E-02,7.515483E-02,
     1 1.3794481F-02,6.56444E-11/
DATA TCR,PCI/647.27,218.167/
DATA TLOW,THI,TA/-40.001,705.001,273.16/
DATA T1,T2,T3,T4,T5/0.0,32.0,40.0,50.0,200.0/
C
IF ((IT,GT, THI) .OR. (IT,LT, TLOW)) GO TO 60
IF (IT,LT, T5) GO TO 10
C
ITK=F(K(TT)
X=TCR-ITK
PS=((((E2*X)*X+C2*X)*X)+B2)*X+A2
PS=PS*(1.0+D2*X)
PS=PS*X/ITK
PS=1.0*X**PS
PS=PCI/PS
PS=PSI(PS)
GO TO 70
C
10 CONTINUE
C
IF (IT,LT, T4) GO TO 20
C
T1K=F(K(TT)
X=TCR-ITK
PS=((C1*X*X)+B1)*X+A1
PS=PS*(1.0+D1*X)
34: \[ PS \times X / TTK \]
35: \[ PS = 1.0 \times PS \]
36: \[ PS = PCR / PS \]
37: \[ PS = PSI(PS) \]
38: \[ GO TO 70 \]
39: \[ C \]
40: \[ 20 \] \[ CONTINUE \]
41: \[ IF (TT \ LT. T3) GO TO 30 \]
42: \[ TTK = FK(TT) \]
43: \[ Z1 = 5388.6012 \]
44: \[ Z2 = 17.30488 \]
45: \[ PS = F(Z1, Z2, TTK) \]
46: \[ GO TO 70 \]
47: \[ C \]
48: \[ 30 \] \[ CONTINUE \]
49: \[ IF (TT \ LT. T2) GO TO 40 \]
50: \[ TTK = FK(TT) \]
51: \[ Z1 = 5427.4515 \]
52: \[ Z2 = 17.444829 \]
53: \[ PS = F(Z1, Z2, TTK) \]
54: \[ GO TO 70 \]
55: \[ C \]
56: \[ 40 \] \[ CONTINUE \]
57: \[ IF (TT \ LT. T1) GO TO 50 \]
58: \[ TTK = FK(TT) \]
59: \[ Z1 = 6143.761123 \]
60: \[ Z2 = 20.06713685 \]
61: \[ PS = F(Z1, Z2, TTK) \]
62: \[ GO TO 70 \]
63: \[ C \]
64: \[ 50 \] \[ CONTINUE \]
65: \[ TTK = FK(TT) \]
66: \[ Z1 = 6098.367784 \]
Z2=19.88939019
PS=F(Z1,Z2,TK)
GO TO 70
C
60 CONTINUE
WRITE(6,100) TT
100 FORMAT('0',10X,'Ti=',F8.2,10X,'TEMPERATURE OUT OF RANGE FOR', 'SATURATION PRESSURE EVALUATION - EXECUTION TERMINATING')
PS=-9999.99
70 CONTINUE
PP=PT-PS
RETURN
END
FUNCTION PW(P, W)
FIND VAPOR PRESSURE OF WATER GIVEN HUMIDITY RATIO.

PH = P*W/(W + 0.622)
RETURN
END
SUBROUTINE OMAX(TI,TO,0,N,P,HAT,K)
C THE PURPOSE OF THIS ROUTINE IS TO DETERMINE THE MAXIMUM HEAT
C TRANSFER FROM A HEAT EXCHANGER.
C THE INPUTS ARE:
C TI - ENTERING TEMPERATURE TO THE HX SECTION (ºF)
C Q - HEAT TRANSFERRED IN THE HX SECTION (BTU/HR)
C W(1,1,K) - ENTERING HUMIDITY RATIO (LB H2O VAPOR/LB DRY AIR)
C W(2,1,K) - ENTERING LIQUID CONDENSATE (LB H2O LIQUID/LB DRY AIR)
C W(3,1,K) - ENTERING SOLID CONDENSATE (LB ICE/LB DRY AIR)
C P - THE TOTAL PRESSURE (PSIA)
C THE OUTPUTS ARE:
C TO - EXIT TEMPERATURE FROM THE HX SECTION (ºF)
C W(1,2,K) - EXIT HUMIDITY RATIO (LB H2O VAPOR/LB DRY AIR)
C W(2,2,K) - EXIT LIQUID CONDENSATE (LB H2O LIQUID/LB DRY AIR)
C W(3,2,K) - EXIT SOLID CONDENSATE (LB ICE/LB DRY AIR)
C WHERE K = 1, TUBE SIDE
C 2, FIN SIDE
DIMENSION W(3,2,2)
DATA HSF/143.333623/
C DETERMINE WHETHER HEATING OR COOLING IS TAKING PLACE.
C IF 0 IS POSITIVE, AND K IS 1, THE FLUID IS BEING HEATED.
C IF 0 IS POSITIVE, AND K IS 2, THE FLUID IS BEING COOLED.
C IF 0 IS NEGATIVE, AND K IS 1, THE FLUID IS BEING COOLED.
C IF 0 IS NEGATIVE, AND K IS 2, THE FLUID IS BEING HEATED.
C
C TH=AMAX(TI,TO)
C TL=AMAXI(TI,TO)
C IF(TH.EQ.TO)ITH=1
C ITL=2
C IF(TL.EQ.TI)ITL=1
IF((Q.GE.0..AND.K.EQ.1).OR.(Q.LT.0..AND.K.EQ.2))GO TO 1
COOLING MODE
SEE IF THE INPUT IS SATURATED.
Q=0.
IF(W(2,ITH,K).LE.0.).OR.(W(3,ITH,K).LE.0.))GO TO 3
SEE IF THE OUTPUT IS SATURATED.
WO=HSAT(TL,P)
IF(W.LT.W(1,ITH,K))GO TO 2
NON-SATURATED INPUT AND OUTPUT
TAVG=(TH+TO)/2.
CP=SPECHT(TAVG,P,W,1,K)
O=RATE*CP*(TH-TL)
RETURN
CALCULATE THE ENERGY REQUIRED TO REACH SATURATION.
FIND THE TEMPERATURE AT WHICH SATURATION OCCURS.
TS=TSAT(P,W(1,ITH,K))
TAVG=(TH+TS)/2.
CP=SPECHT(TAVG,P,W,1,K)
O=RATE*CP*(TH-TS)
GO TO 4
CALCULATE THE ENERGY REQUIRED TO CONDENSE LIQUID WATER.
O=0.
TS=TH
4 IF(TH.LT.32.00)GO TO 11
T=TL
IF(TL.LT.32.00)T=32.00
TAVG=(TS+T)/2.
CP=SPECHT(TAVG,P,W,1,K)
HIVAP=HFG(TAVG,P,W,1,K)
K0=KSAT(T,P)
COND=M(1,ITH,K)-K0
Q=RATE*CP*(TS-T)
QL=RATE*COND*HIVAP
Q=QL
0=0+Q
IF(TL.LE.32.00)GO TO 5
RETURN
C CALCULATE THE ENERGY REQUIRED FOR PHASE CHANGE FROM LIQUID TO
C SOLID.
C
5 W(1,ITH,K)=K0
W(2,ITH,K)=W(2,ITH,K)+COND
QPC=RATE*W(2,ITH,K)*HSP
Q=Q+QPC
IF(TL.EQ.32.00)RETURN
W(3,ITH,K)=W(3,ITH,K)+W(2,ITH,K)
W(2,ITH,K)=0.0
C CALCULATE THE ENERGY REQUIRED TO CONDENSE TO ICE.
C
11 TH=AMIN1(32.00,TH)
TAVG=(TH+TL)/2.
CP=SPECHT(TAVG,P,W,1,K)
HSUB=HSG(TAVG,P,W,0,K)
100: \( W_0 = W_{SAT}(T_L, P) \)
101: \( C_{HD} = W(1, I_{HL, K}) - W_0 \)
102: \( Q_0 = \text{RAT} \times C \times (T_H - T_L) \)
103: \( Q_L = \text{RAT} \times C \times H_{SUB} \)
104: \( Q_T = Q_0 + Q_L \)
105: \( Q = Q_T + Q_0 \)
106: RETURN
107: C HEATING MODE
108: C SEE IF THE INPUT IS SATURATED.
109: C
110: C
111: C
112: I \( Q = 0. \)
113: IF(\( N(2, I_{HL, K}) \),EQ.0.,AND.W(3, I_{HL, K}),EQ.0.,)GO TO 17
114: C
115: C CALCULATE THE ENERGY REQUIRED TO SUBLIME ICE.
116: C
117: C FIND THE TEMPERATURE AT WHICH SATURATION OCCURS.
118: C
119: IF(\( T_L \cdot G T \cdot 32.00 \))GO TO 18
120: \( W_T = W(1, I_{HL, K}) + W(3, I_{HL, K}) \)
121: \( T_S = I_{SAT}(P, W_T) \)
122: \( T = \text{AMIN}(32.00, T_S) \)
123: \( T_{AVG} = (T + T_L) / 2 \)
124: \( C_P = \text{SPECTR}(T_{AVG}, P, W, 1, K) \)
125: \( H_{SUB} = I_{SG}(T_{AVG}, P, W, 1, K) \)
126: \( W_0 = W_{SAT}(T, P) \)
127: \( E_{VAP} = W_0 - W(1, I_{HL, K}) \)
128: \( Q_0 = \text{RAT} \times C \times (T - T_L) \)
129: \( Q_L = \text{RAT} \times E_{VAP} \times H_{SUB} \)
130: \( Q = Q_0 + Q_L \)
131: IF(\( T_S \cdot L T \cdot 32.00 \))GO TO 6
132: C
C CALCULATE THE ENERGY REQUIRED FOR PHASE CHANGE FROM SOLID TO LIQUID.

\[ \text{QPC} = \text{RATE} \times \text{W}(3, \text{ITL}, \text{K}) \times \text{HSF} \]

\[ \text{Q} = \text{Q} + \text{QPC} \]

\[ \text{W}(2, \text{ITL}, \text{K}) = \text{W}(2, \text{ITL}, \text{K}) + \text{W}(3, \text{ITL}, \text{K}) \]

\[ \text{W}(3, \text{ITL}, \text{K}) = 0.0 \]

IF(TH.EQ.32.00) RETURN

C CALCULATE THE ENERGY REQUIRED TO EVAPORATE LIQUID WATER.

C FIND THE TEMPERATURE AT WHICH SATURATION OCCURS.

\[ \text{W}_T = \text{W}(1, \text{ITL}, \text{K}) + \text{W}(2, \text{ITL}, \text{K}) \]

\[ \text{TS} = \text{TSAT}(P, \text{W}_T) \]

\[ \text{T} = \text{AMIN}1(\text{TS}, \text{TH}) \]

\[ \text{TL} = \text{AMAX}1(32.00, \text{TL}) \]

\[ \text{TAVG} = (\text{TL} + \text{T}) / 2. \]

\[ \text{CP} = \text{SPECHT}(\text{TAVG}, P, \text{W}, 1, \text{K}) \]

\[ \text{HVAP} = \text{HFG}(\text{TAVG}, P, \text{W}, 1, \text{K}) \]

\[ \text{WO} = \text{WSAT}(P, \text{T}) \]

\[ \text{EVAP} = \text{WO} - \text{W}(1, \text{ITL}, \text{K}) \]

\[ \text{OS} = \text{RATE} \times \text{CP} \times (\text{T} - \text{TL}) \]

\[ \text{QL} = \text{RATE} \times \text{EVAP} \times \text{HVAP} \]

\[ \text{QT} = \text{OS} + \text{QL} \]

\[ \text{Q} = \text{Q} + \text{QT} \]

IF(TS, GE, TH) RETURN

C CALCULATE THE OUTLET TEMPERATURE AT NON-SATURATED CONDITIONS.

6 TL=TS

\[ \text{W}(1, \text{ITL}, \text{K}) = \text{WO} \]
17 \( w(2, \text{ITL}, k) = 0.0 \)

\( CP = \text{SPEECH}(TAVG, P, \text{N}, k) \)

\( CP = \text{RATET}(TAVG, P, \text{N}, k) \)

\( Q_0 = Q + Q_0 \)

RETURN

END
FUNCTION RH(PW, PSW)

C FIND RELATIVE HUMIDITY GIVEN VAPOR PRESSURE AND SATURATION VAPOR PRESSURE

C

RH=PW/PSW*100.

RETURN

END
FUNCTION RHW(T,P,RH)
C
C FIND HUMIDITY RATIO GIVEN RELATIVE HUMIDITY
C
CALL PSAT(T,P,PP,PSW)
PWN=RH*PSW/100.
RHW=HR(P,PWN)
RETURN
END
FUNCTION SPECHT(T,P,W,J,K)

THE PURPOSE OF THIS FUNCTION IS TO CALCULATE THE SPECIFIC HEAT
OF A MOIST AIR AND CONDENSATE MIXTURE.

OBTAIN THE SPECIFIC HEATS FOR AIR AND THE WATER PHASES.

DIMENSION W(3,2,2)

CALL PPROP(T,P,RHO,CPA,VIS,COND,PR,1)
CALL PPROP(T,P,RHO,CPV,VIS,COND,PR,2)
CALL PPROP(T,P,RHO,CPL,VIS,COND,PR,3)
CALL PPROP(T,P,RHO,CPS,VIS,COND,PR,4)

CALCULATE THE COMPOSITE SPECIFIC HEAT FOR THE MIXTURE

(BTU/LB DRY AIR - F)

SPECHT=CPA+CPV*W(1,J,K)+CPL*W(2,J,K)+CPS*W(3,J,K)

RETURN
END
SUBROUTINE TCOEF(ITAVG, TH, WIDTH, W, K, NPASS)

THE PURPOSE OF THIS ROUTINE IS TO FIND THE FILM COEFFICIENT ON THE
TUBE SIDE OF A HEAT EXCHANGER USING SUNDSTRAND 3/8 DIA TUBES
WITH TURBULATIONS.

COMMON/TWO/TPIN, TAU, TK, TCP, TPR, TRHO, TDH, TRATE, INU, TRE, TG
COMMON/THREE/HEIGHT, DEPTH, NPPI
DATA THFA/6.9887E-4/

CALCULATE THE PHYSICAL PROPERTIES OF THE FLUID STREAM.

CALL PPROP(ITAVG, TPIN, TRHO, TCP, TMU, TK, TPR, 1)
TCP1 = SPECHT(ITAVG, TPIN, w, 1, 1)
TCP0 = SPECHT(ITAVG, TPIN, w, 2, 1)
TCP = (TCP1 + TCP0) / 2.

CALCULATE FLOW PER UNIT AREA (G). (LBM/FT**2-HR)

TUBES = (HEIGHT/1.00)*(DEPTH/1.00)/FLOAT(NPASS)
TUBES = AINT(TUBES)
AREA = THFA*TUBES
TG = TRATE/AREA
WRITE(6,10)TUBES, TRATE, AREA
10 FORMAT(5X,'TUBES = ',F10.1,5X,'TRATE = ',F10.2,5X,'AREA = ',F10.4)

CALCULATE THE REYNOLDS NUMBER (TRE)

TRE = TG*TDH/TMU
WRITE(6,11)TG, TDH, TRE
11 FORMAT(5X,'TG = ',F10.2,5X,'TDH = ',F10.6,5X,'TRE = ',F10.1)

CALCULATE NUSSELT NUMBER
34: C
35: C  TNU=0.14192*TR**0.619*TPR**0.118
36: C
37: C  CALCULATE THE FILM COEFFICIENT
38: C
39: C  TH=TNU*TK/TH
40: C  WRITE(6,12)TNU,TH
41: 12 FORMAT(5X,'TNU = ',F10.3,5X,'TH = ',F10.2)
42: RETURN
43: END
FUNCTION TSAT(P,W)

C CALCULATE THE TEMPERATURE AT WHICH SATURATION OCCURS FOR A GIVEN HUMIDITY RATIO.

TL=-40.
TH=705.
NSAT=0
3 T=(TL+TH)/2.
įWS=W SAT(T,P)
11 IF(WS.LT.0.)TH=T
12 IF(WS.LT.0.)GO TO 3
13 IF(ABS(WS-W).LE.0.0000001)GO TO 4
14 NSAT=NSAT+1
15 IF(NSAT.GE.50)CALL ERROR('NSAT')
16 IF(WS-W)1,1,2
17 TL=T
18 GO TO 3
19 TH=T
20 GO TO 3
21 TSAT=T
22 RETURN
23 END
SUBROUTINE TOUT(I,T,O,Q,W,P,RATE,QKEY,K)
C THE PURPOSE OF THIS ROUTINE IS TO DETERMINE THE EXIT TEMPERATURE
C AND HUMIDITY RATIO FROM A HEAT EXCHANGER SECTION.
C THE INPUTS ARE:
C T1        - ENTERING TEMPERATURE TO THE HX SECTION (F)
C Q        - HEAT TRANSFERRED IN THE HX SECTION (BTU/HR)
C W(1,1,K) - ENTERING HUMIDITY RATIO (LB H2O VAPOR/LB DRY AIR)
C W(2,1,K) - ENTERING LIQUID CONDENSATE (LB H2O LIQUID/LB DRY AIR)
C W(3,1,K) - ENTERING SOLID CONDENSATE (LB ICE/LB DRY AIR)
C P        - THE TOTAL PRESSURE (PSIA)
C THE OUTPUTS ARE:
C TO        - EXIT TEMPERATURE FROM THE HX SECTION (F)
C W(1,2,K) - EXIT HUMIDITY RATIO (LB H2O VAPOR/LB DRY AIR)
C W(2,2,K) - EXIT LIQUID CONDENSATE (LB H2O LIQUID/LB DRY AIR)
C W(3,2,K) - EXIT SOLID CONDENSATE (LB ICE/LB DRY AIR)
WHERE K = 1, TUBE SIDE
        2, FIN SIDE
C DIMENSION W(3,2,2)
DATA HSF/143.338623/
C DETERMINE WHETHER HEATING OR COOLING IS TAKING PLACE.
C IF Q IS POSITIVE, AND QKEY IS +1, THE FLUID IS BEING HEATED.
C IF Q IS POSITIVE, AND QKEY IS -1, THE FLUID IS BEING COOLED.
C IF Q IS NEGATIVE, AND QKEY IS +1, THE FLUID IS BEING COOLED.
C IF Q IS NEGATIVE, AND QKEY IS -1, THE FLUID IS BEING HEATED.
C IF((Q.GE.0..AND.QKEY.EQ.1.) .OR. (Q.LT.0..AND.QKEY.EQ.-1.))GO TO 1
C COOLING MODE
C SEE IF THE INPUT IS SATURATED.
C
CALCULATE THE EXIT TEMPERATURE ASSUMING CONSTANT PROPERTIES.

IF \( (W(2, 1, K), N.E. = 0.) \), \( (W(3, 1, K), N.E. = 0.) \), GOTO 3

\[ \text{CP = \text{SPECHE}(T, P, W(1, 1, K)),} \]
\[ \text{TO = \text{TI} + \text{OKEY} * \text{Q}/(\text{RATE} * \text{CP})}. \]

SEE IF THE OUTPUT IS SATURATED.

\[ \text{WO = \text{MSAI}(T0, P)}. \]
\[ \text{IF (WO, N.E. = 0.)} \text{GOTO 2} \]

NON-SATURATED INPUT AND OUTPUT

\[ \text{TAVG} = (\text{TI} + \text{TO})/2. \]
\[ \text{CP = \text{SPECHE}(\text{TAVG}, P, W(1, 1, K))}, \]
\[ \text{TO = \text{TI} + \text{OKEY} * \text{Q}/(\text{RATE} * \text{CP})}. \]
\[ \text{IF (WO, N.E. = 0.)} \text{GOTO 2} \]
\[ \text{WO} = \text{MSAI}(T0, P) \]
\[ \text{W(1, 2, K)} = \text{W(1, 1, K)} = 0. \]
\[ \text{W(2, 2, K)} = \text{W(3, 2, K)} = 0. \]
RETURN

CALCULATE THE ENERGY REQUIRED TO REACH SATURATION.

\[ \text{TS} = \text{TSAT}(\text{P}, W(1, 1, K)) \]
\[ \text{CP} = \text{SPECHE}(\text{TAVG}, P, W(1, 1, K)) \]
\[ \text{OSAI} = \text{OKEY} * \text{Q}/(\text{RATE} * \text{CP}) \cdot (\text{TS} - \text{TI}) \]
\[ \text{OA} \text{IS THE HEAT ENERGY AVAILABLE AFTER SATURATION IS REACHED.} \]

CALCULATE THE EXIT TEMPERATURE AT WHICH SATURATION OCCURS.

\[ \text{TS} = \text{TSAT}(\text{P}, W(1, 1, K)) \]
\[ \text{CP} = \text{SPECHE}(\text{TAVG}, P, W(1, 1, K)) \]
\[ \text{OSAI} = \text{OKEY} * \text{Q}/(\text{RATE} * \text{CP}) \cdot (\text{TS} - \text{TI}) \]
\[ \text{OA} \text{IS THE HEAT ENERGY AVAILABLE AFTER SATURATION IS REACHED.} \]
C
QA=0-QUAT
GO TO 4

C
CALCULATE THE ENERGY REQUIRED TO CONDENSE LIQUID WATER.

C
3 QA=O
IF(TI.LT.32.00)GO TO 11
TS=TI

4 TH=TS
K1=O

TL=32.00
NCL=O

9 T=(TH+TL)/2.
TAVG=(TS+T)/2.
IF(KI.EQ.0)TAVG=T
IF(KI.EQ.0)T=TL
CP=SPECH(TAVG,P,W,1,K)
HVAP=HFG(TAVG,P,W,1,K)
WO=WSAT(T,P)
COND=W(1,1,K)-WO
OS=QKEY*RATE*CP*(T-TS)
QL=-QKEY*RATE*COND*HVAP
OT=OS+QL
IF((KI.EQ.0).AND.(ABS(QA).GT.ABS(QT)))GO TO 5
IF(KI.EQ.0)GO TO 6
IF(ABS(QA-QT).LE.0.05)GO TO 10
NCL=NCL+1
IF(NCL.GE.50)CALL ERROR('NCL')
IF(ABS(QA)-ABS(QT))7,7,8
7 TL=T
GO TO 9
8 TH=T
GO TO 9
3 K1=1
GO TO 9
C
C
DEFIN THE OUTLET HUMIDITY RATIOS.
C
10 W(1,2,K)=NO
10 W(2,2,K)=W(2,1,K)+COND
11 W(3,2,K)=0.0
12 TO=T
13 RETURN
C
CALCULATE THE ENERGY REQUIRED FOR PHASE CHANGE FROM LIQUID TO
SOLID.
C
5 QA=QA-QT
W(1,1,K)=NO
W(2,1,K)=W(2,1,K)+COND
QC=-QKEY*RATE*W(2,1,K)*HSF
IF(ABS(QPC),GT,Abs(QA))GO TO 12
QA=QA-QPC
W(3,1,K)=W(3,1,K)+W(2,1,K)
W(2,1,K)=0.0
10 GO TO 11
C
C
T=32.00, PART LIQUID AND PART SOLID CONDENSATE.
C
12 WPC=-QKEY*QA/(RATE*HSF)
13 W(1,2,K)=NO
14 W(2,2,K)=W(2,1,K)-WPC
15 W(3,2,K)=W(3,1,K)+WPC
16 TO = 32.00
17 RETURN
CALCULATE THE ENERGY REQUIRED TO CONDENSE TO ICE.

\[ TH = \text{MIN}(32.00, TI) \]

\[ TL = -50. \]

\[ TF = TH \]

\[ NCL = 0 \]

\[ T = (TH + TL)/2. \]

\[ TAVG = (TF + T)/2. \]

\[ CP = \text{SPECHI}(TAVG, P, W, I, K) \]

\[ HSUB = \text{HSG}(TAVG, P, W, Q, K) \]

\[ W = \text{WSAT}(T, P) \]

\[ \text{COND} = W(1, I, K) - W0 \]

\[ QS = \text{QKEY*RATE*CP*(T-TF)} \]

\[ QL = \text{QKEY*RATE*COND*HSUB} \]

\[ QT = QS + QL \]

\[ \text{IF}(|ABS(QA-QT)|.LE.0.05) \text{GO TO 13} \]

\[ NCL = NCL + 1 \]

\[ \text{IF}(NCL.GE.50) \text{CALL ERROR('NCL')} \]

\[ \text{IF}(ABS(QA)-ABS(QT))14,14,15 \]

\[ TL = T \]

\[ \text{GO TO 16} \]

\[ TH = T \]

\[ \text{GO TO 16} \]

\[ \text{RETURN} \]

\[ \text{C} \]

\[ \text{HEATING MODE} \]

\[ \text{C} \]

\[ \text{SEE IF THE INPUT IS SATURATED.} \]
166: C
167: 1 IF(W(2,1,K).EQ.0. AND.W(3,1,K).EQ.0.)GO TO 17
168: C
169: C  CALCULATE THE ENERGY REQUIRED TO SUBLIME ICE.
170: C
171: C  FIND THE TEMPERATURE AT WHICH SATURATION OCCURS.
172: C  IF(TI.GT.32.00) GO TO 18
173: NT=W(1,1,K)+W(3,1,K)
174: TS=TSAT(P,WT)
175: K1=0
176: TH=AMIN1(32.00,TS)
177: TL=TI
178: NCL=0
179: 24 T=(TH+TL)/2.
180: TAVG=(T+TI)/2.
181: IF(K1.EQ.0)TAVG=T
182: IF(K1.EQ.0)T=TH
183: CP=SPECHT(TAVG,P,W,1,K)
184: HSAT=HSG(TAVG,P,W,1,K)
185: W0=WSAT(T,P)
186: EVAP=W0-W(1,1,K)
187: QS=QKEY*RATE*CP*(T-TI)
188: QL=QKEY*RATE*EVAP*HSAT
189: OT=QS+QL
190: IF((K1.EQ.0).AND.(ABS(Q).GT.ABS(QT)).AND.(T.EQ.32.))GO TO 19
191: IF((K1.EQ.0).AND.(ABS(Q).GT.ABS(QT)).AND.(T.LT.32.))GO TO 34
192: IF(K1.EQ.0)GO TO 20
193: IF(ABS(Q-QT).LE.0.05)GO TO 21
194: NCL=NCL+1
195: IF(NCL.GE.50)CALL ERROR('NCL')
196: IF(ABS(Q)-ABS(QT))22,22,23
197: 22 TI=T
198: GO TO 24
23 TL=1
20 TO 24
20 K1=1
20 CO TO 24
20 C DEFINES THE OUTLET HUMIDITY RATIOS.
20 C
21 n(1,2,K)=n0
20 n(2,2,K)=0.0
20 n(3,2,K)=n(3,1,K)-EVAP
20 TO=I
20 RETURN
21 C CALCULATE THE ENERGY REQUIRED FOR PHASE CHANGE FROM SOLID TO
21 C LIQUID.
21 C
21 19 CA=Q-QT
21 19 n(1,1,K)=n0
21 19 n(3,1,K)=n(3,1,K)-EVAP
21 19 OPC=QKEY*RATE*W(3,1,K)*HSF
21 19 IF(ABS(OPC),GT.ABS(QA))GO TO 25
21 20 CA=QA-OPC
21 21 W(2,1,K)=W(2,1,K)+W(3,1,K)
21 21 n(3,1,K)=0.0
21 20 GO TO 26
21 24 C T=32.00, PART LIQUID AND PART SOLID CONDENSATE.
21 24 C
21 25 WPC=QKEY*QA/(RATE*HSF)
21 25 n(1,2,K)=n0
21 25 n(2,2,K)=n(2,1,K)+WPC
21 25 n(3,2,K)=n(3,1,K)-WPC
21 25 TO=32.00
RETURN

CALCULATE THE ENERGY REQUIRED TO EVAPORATE LIQUID WATER.
FIND THE TEMPERATURE AT WHICH SATURATION OCCURS.

18 QA=0
20 wT=w(1,1,K)+w(2,1,K)
20 TS=TSAT(P,wT)
20 K1=0
20 TH=TS
20 TL=AMAX1(32,00,T1)
20 TF=TL
20 NCL=0
32 T=(TH+TL)/2.
32 TAVG=(TF+T)/2.
32 IF(K1.EQ.0)TAVG=T
32 IF(K1.EQ.0)T=TS
32 CP=SPECHT(TAVG,P,w1,K)
32 HVAP=HFG(TAVG,P,w1,K)
32 wO=wSAT(T,P)
32 EVAP=wO-w(1,1,K)
32 CS=QKEY*RATE*CP*(T-TF)
32 QL=QKEY*RATE*EVAP*HVAP
32 CT=OS+QL
32 IF((K1.EQ.0).AND.(ABS(QA).GT.ABS(QT)))GO TO 27
32 IF(K1.EQ.0)GO TO 28
32 IF(ABS(QA-QT).LE.0.05)GO TO 29
32 NCL=NCL+1
32 IF(NCL.GE.50)CALL ERROR('NCL')
30 TH=T
263: 30 GO TO 32
31 TL=T
GO TO 32
266:  K1=1
267:  GO TO 32

C
269:  C DEFINE THE OUTLET HUMIDITY RATIOS.

C
271:  29 W(1,2,K)=W0
272:  W(2,2,K)=W(2,1,K)-EVAP
273:  W(3,2,K)=0.0
274:  TO=T
275:  RETURN

C
276:  C CALCULATE THE OUTLET TEMPERATURE AT NON-SATURATED CONDITIONS.

C
279:  17 QA=Q
280:  T=I
281:  GO TO 33
282:  34 QA=Q
283:  27 QA=QA-QT
284:  W(1,1,K)=W0
285:  W(2,1,K)=0.0
286:  33 CP=SPECHT(T,P,W,1,K)
287:  TO=QKEY*QA/(RATE*CP)+T
288:  TAVG=(T+TO)/2.
289:  CP=SPECHT(TAVG,P,W,1,K)
290:  TO=QKEY*QA/(RATE*CP)+T
291:  W(1,2,K)=W(1,1,K)
292:  W(2,2,K)=0.0
293:  W(3,2,K)=0.0
294:  RETURN
295:  END
FUNCTION WRH(T,P,W)

C FIND RELATIVE HUMIDITY GIVEN HUMIDITY RATIO

C

PWW=Pw(P,W)

CALL PSAT(T,P,PP,PSW)

WRH=PWW/PSW*100.

RETURN

END
FUNCTION WSAT(T, P)
C
C CALCULATE HUMIDITY RATIO AT SATURATION.
C
CALL PSAT(T, P, PPAIN, PPV)
WSAT=HR(P, PPV)
RETURN
END
APPENDIX B

DATA AND DATA REDUCTION PROGRAMS FOR TUBE TESTS
## TEST DATA

3/8" OD Tube with Turbulator  
Test Section Heat Transfer Fluid -- Air

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# TEST DATA

3/8" OD Tube with Turbulator
Test Section Heat Transfer Fluid -- Air

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### TEST DATA

3/8" OD Tube with Turbulator

Test Section Heat Transfer Fluid -- Air

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## TEST DATA

*3/8*" OD Tube with Turbulator

Test Section Heat Transfer Fluid -- Air

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### TEST DATA

**3/8" OD Tube with Turbulator**

**Test Section Heat Transfer Fluid -- Air**

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3/8" OD Tube with Turbulator
Test Section Heat Transfer Fluid -- Air

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## TEST DATA

**3/8" OD Tube with Turbulator**  
**Test Section Heat Transfer Fluid -- Water**

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## TEST DATA

3/8" OD Tube with Turbulator  
Test Section Heat Transfer Fluid -- Water

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| Test Section Flow (gph water) | 14.0| 14.0| 16.0| 16.0| 18.0| 18.0|
| Outer Annulus Flow (gpm water) | 3.0 | 3.0 | 3.0 | 3.0 | 3.0 | 3.0 |
**TEST DATA**

3/8" OD Tube with Turbulator

Test Section Heat Transfer Fluid -- Water

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Data Reduction Program for 
Tests with Air as Test Section Heat Transfer Fluid

User Instructions

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| 31413014   | 02 | 0.000215198  | 32 |
| 17150000000| 03 | 0.0000000002 | 33 |
| 3141       | 04 | 1.32603      | 34 |
| 3636172737 | 05 | 460.0        | 35 |
| 3335       | 06 | 7.642        | 36 |
| 1331162727 | 07 | 17.87325     | 37 |
| 2135       | 08 | -0.12244     | 38 |
| 2415372432 | 09 | 0.00095      | 39 |
| 0          | 10 | 0.52068      | 40 |
| 0          | 11 | 0.0          | 41 |
| 0          | 12 | 0.0          | 42 |
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| 0          | 18 | 0.0          | 48 |
| 0          | 19 | 0.0          | 49 |
| 0          | 20 | 0.0          | 50 |
| 0          | 21 | 0.0          | 51 |
| 0          | 22 | 0.0          | 52 |
| 0          | 23 | 0.0          | 53 |
| 0          | 24 | 0.0          | 54 |
| 0.03696    | 25 | 0.000211315  | 55 |
| 0.00036455 | 26 | 0.073300000  | 56 |
| -0.0000000215 | 27 | 0.52246     | 57 |
| 0.23942    | 28 | -0.00025448  | 58 |
| 0.000051852| 29 | 0.000003037  | 59 |
Data Reduction Program for
Tests with Water as Test Section Heat Transfer Fluid

User Instructions

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Note: This program must be used with the FC-100A Print Cradle.
Store the following constants in the memories prior to program execution.

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APPENDIX C

LEAST SQUARES CURVE FIT PROGRAMS
Least-Squares Curve-Fit Program for Polynomial Equations

Use for finding coefficients to the polynomial equation
\[ f(x) = a_0 + a_1 x + a_2 x^2 + \ldots + a_n x^n \]
which has been curve-fit through a set of data points by the method of least squares. For this program, \( n \) has a maximum value of 6.
Use with a Texas Instruments TI-59 programmable calculator. Set the memory partition at 239.89.

User Instructions

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<td>( x_j, y_j )</td>
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To calculate a polynomial of a different order, repeat steps 3 thru 5.

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Repeat as often as desired.
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Least-Squares Curve-Fit Program for Exponential, Logarithmic and Power Equations

Use for finding coefficients $a$ and $b$ for equations of the form

$$f(x) = a \cdot b^x$$ \hspace{1cm} \text{(exponential)}

$$f(x) = a + b \ln(x)$$ \hspace{1cm} \text{(logarithmic)}

$$f(x) = a \cdot x^b$$ \hspace{1cm} \text{(power)}

which have been curve fit through a set of data points by the method of least squares. For the exponential equation, all $y_j$ must be greater than 0; for the logarithmic equation, all $x_j$ must be greater than 0; and, for the power equation, all $x_j$ and $y_j$ must be greater than 0. Use with a Texas Instruments TI-59 programmable calculator. Use the standard memory partition of 319.79.

**User Instructions**

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