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ANALYSIS OF A FORCED-FLOW BARE TUBE HEAT
EXCHANGER FOR CONDENSING SHELL-SIDE STEAM
IN THE PRESENCE OF AIR AND HYDROGEN

D. L. Ayers and D. T. Beecher
Heat Transfer and Fluid Dynamics

Research Report WERL-TAGOV-1

June 14, 1971

WESTINGHOUSE RESEARCH LABORATORIES

Pittsburgh, Pennsylvania 15235

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ABSTRACT

An analysis is developed to predict heat transfer and pressure drop for a forced-flow bare-tube heat exchanger condensing saturated shell-side steam in the presence of air and hydrogen. A digital computer program written to evaluate the analysis is described. The program is used in a parametric study of heat transfer performance as a function of exchanger geometry, gas velocity and coolant flow rate. Results show that an exchanger can be designed with a capacity which will exceed the desired heat transfer rate of 15000 Btu/sec in the prescribed geometric envelope of 6.5 by 5.3 by 2 feet.

TABLE OF CONTENTS

	<u>Page</u>
ABSTRACT	
I. Introduction	1
II. Conclusions.	3
III. Results of Parametric Study.	5
A. Effect of Transverse and Longitudinal Tube Pitch Ratios	5
B. Effect of Varying Gas Velocity and Coolant Flow Rate .	7
C. Effect of Tube Outside Diameter.	8
D. Effect of Tube Wall Thickness.	8
E. Effect of Inclining Tubes.	9
F. Required Fan System.	10
G. Capacity Conservatism.	11
H. Overall Conductance.	11
IV. Analysis	15
A. Gas-Film-Cooling Water Energy Balance.	15
B. Heat and Mass Transfer from Gas Stream	17
C. Heat Transfer Across Condensate Film	23
D. Tube to Cooling Water Heat Transfer.	30
E. Overall Heat Transfer.	33
F. Gas-Side Energy Balance.	34
G. Water-Side and Gas-Side Pressure Drop.	35
H. Evaluation of Analysis	37
Nomenclature	41
References	49

APPENDICES

A. Thermodynamic Properties of Gas, Condensate and Coolant	51
1. Gas Dynamic Viscosity	51
2. Gas Thermal Conductivity	52
3. Gas Diffusion Coefficient	53
4. Gas Density	54
5. Gas Specific Heat	54
6. Water Properties	54
7. Gas Stream Mole and Mass Fractions	54
B. Computer Program	57
1. Nomenclature	57
2. Listing	62
C. Tabulation of Results of Parametric Study	75

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I. INTRODUCTION

Pressurized water nuclear reactors, in the unlikely event of a primary coolant leak, can fill containment space with a mixture of high pressure steam and non-condensable gases. Assuming all the primary coolant can flash into steam and with no means provided for heat or steam mass removal, containment must be designed to withstand much higher pressure than would be necessary if some heat or steam mass removal equipment were employed. Various schemes have been considered (spraying, venting, contact heat exchangers) for maintaining containment pressure below a reasonable pressure level. Previous work in this field (Ref. 9) has shown the capacity of plate-fin heat exchangers for direct heat removal (steam condensation) in stationary power plants.

This report presents an analysis of the performance of a forced flow bare-tube heat exchanger in a high pressure steam-air-hydrogen environment. The analysis considers in detail the mass transfer of steam in the presence of non-condensable gases, the distribution and flow of condensed water on the outer tube walls, the flow of heat from tube walls to the cooling water, as well as the change in gas state from row to row. In addition, consideration is given to the required gas moving equipment. The analysis is programmed for a digital computer and the program is used in a parametric study of heat transfer capacity and pressure drop as a function of exchanger geometry, the gas inlet thermodynamic state, and gas and coolant flow rates.

It was specifically desired to find if a forced flow exchanger with maximum plan dimensions of 6.5 feet length (including headers) and 5.3 feet width could transfer 15000 Btu/sec. The height was to be no more than 2 feet including air moving equipment. The gas inlet state was to be 110.2 psi steam pressure (saturated), 1.88 psi air pressure and 2.92 psi hydrogen pressure. Gas flow rate was not specified but low values were desired to insure small fan power requirements. Water was to enter the exchanger at 85°F and its flow rate was not to exceed 2500 gallons/minute, with a pressure drop not to exceed 57 psi. No restrictions were placed on exchanger tube pitches, diameter or wall thickness but the tube material was to be 90-10 cupro-nickel having a thermal conductivity of 26 Btu/hour-foot-°F.

It was found that many combinations of the tube pitch, tube diameter, gas and water flow rates would give a capacity of 15000 Btu/sec. The results are summarized in Appendix C. The results show that small diameter tubes with the larger tube pitch and low gas velocity will give the desired capacity and a low fan power requirement. Water flow rate must be traded off against overall height (number of rows) in the exchanger.

II. CONCLUSIONS

The analysis has shown that a forced-flow bare tube heat exchanger condensing steam with air and hydrogen can be designed to transfer 15000 Btu/sec when the plan dimensions are 6.5 feet length by 5.3 feet width. The parametric study shows that heat transfer is water-side limited and that lower gas velocities are desirable since they only slightly degrade performance but greatly reduce fan horsepower. The study also shows that larger tube pitches and smaller diameter tubes are desirable from the standpoint of fan horsepower and heat transfer.

Final design selection can be made by trading off coolant flow rate with total number of exchanger rows, the governing criteria being overall heat transfer rate, fabrication cost and coolant pumping requirements. Selection of gas moving equipment depends upon the final choice of exchanger geometry.

An exchanger with 1 inch diameter tubes and having 1.25 by 1.25 pitch ratios, using 5000 gallons/minute coolant and removing 1000 scfm of non-condensables at the exchanger exit will not have the desired conductance of $516 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$.

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III. RESULTS OF PARAMETRIC STUDY

The computer program has been run for 75 cases in which tube diameter, tube wall thickness, transverse and longitudinal pitch ratio, gas inlet velocity and water volumetric flow rate have been varied. The maximum plan dimensions for all runs were 6.5 feet length by 5.3 feet width, while the height varied with longitudinal tube pitch and the number of tube rows. Tube diameters of 1.0, 0.75 and 0.5 inches were considered, with varying wall thickness for each. Cooling water inlet temperature was maintained at 85°F. Tube fouling factors ff_o and ff_i were both zero for all runs. Prior to usage in the event of a reactor incident, the system will be hydraulically tested with fresh water and the system will then be drained. Thus ff_i will be negligible. The exchanger will be located in the reactor compartment where the atmosphere will be controlled so one need not be concerned with fouling on the outer tube surface. The gas inlet state to the first tube row was always held at $P_g = 110.2$ (saturated), $P_a = 1.88$ and $P_h = 2.92$ psia. The results of the parametric study are tabulated in Appendix C. The following conclusions can be drawn from the data:

A. Effect of Transverse and Longitudinal Tube Pitch Ratios

Runs 1-9, 10-18 and 19-27 were made for 1 inch, 0.75 inch and 0.5 inch BWG 14 condenser tubes, respectively. Gas face velocity was maintained at 480 feet/minute while total coolant flow was held at a nominal 1500 gallons/minute. The coolant flow was divided equally among all tubes that would exist in a ten row exchanger. If it was found that the desired capacity of 15000 Btu/sec was achieved in less than ten rows, the run was truncated there. Thus for those runs showing less than ten rows, the flow rate of coolant is proportionally less. In each of these three sets of runs all possible combinations of transverse and longitudinal tube pitch ratios of 1.25, 1.50 and 1.75 were used. Taking the

runs in groups of 3, i.e., 1-3, 4-6, ..., one can see the effect of increasing transverse pitch ratio while keeping all other parameters constant. The effect is a slight decrease in overall capacity. For those sets of runs where one or more runs were truncated with less than ten rows this effect is not obvious. When the pitch ratio increases, the total number of tubes in an exchanger decreases and so overall heat transfer area is reduced. At the same time, with coolant flow held constant, the fewer tubes each carry more coolant which causes the water-side convective film coefficient to increase. As Eq. (31) shows, water-side film coefficient varies as coolant flow rate to the .8 power. If one were to double the transverse tube pitch ratio the number of tubes and the heat transfer surface area would be 1/2 the previous value while the coolant convective film coefficient would increase by $2^{.8}$. If all the temperature drop between gas and water were in water-side convection, the net result would be $Q_{t2} \sim Q_{t1} \cdot 2^{-.2}$. The conclusion is, for a given number of tube rows, increasing transverse tube pitch ratio will decrease overall capacity. In the foregoing discussion the effect of changes in pitch ratio on gas side coefficients and its subsequent effect on overall capacity was not included because the values of gas velocity and coolant flow used therein were such that overall heat transfer was water-side limited. Had the coolant flow rate been much higher and the gas velocity much lower, heat transfer would become gas-side limited and the change in pitch would not have had a marked effect on overall heat transfer as a result of the coolant film coefficient but the reduction in tube surface area would cause heat transfer reduction approximately in proportion to reduction in area.

As all sets of data show, increasing transverse tube pitch ratio causes a marked reduction in gas-side pressure drop and in ideal fan horsepower. Since increase in pitch causes only small reduction in heat transfer but greatly reduces fan horsepower, it would appear that the best design would incorporate the larger pitch ratios. The subsequent reduction in total tubes for a given tube diameter would also reduce exchanger material and fabrication costs. The exchanger volume/unit heat transfer would increase, however.

In comparing runs 1, 4 and 7, as well as 2, 5 and 8 etc., one can see the effect of increasing longitudinal tube pitch ratio while keeping all other parameters constant. The effect is most apparent in runs 1, 4 and 7 and 2, 5 and 8 where the total number of tube rows is the same and comparisons of listed overall capacity are valid. In these sets of runs, increasing longitudinal pitch does not change the total number of tubes so the pitch variation does not alter either coolant film coefficient or heat transfer surface area. The overall heat transfer is seen to increase slightly with increasing longitudinal pitch. This is due to changes in the coefficient b^* and exponent n in Eqs. (11) and (13), as listed in Table I as a function of pitch ratios. Gas-side pressure drop and ideal fan horsepower are about the same for all of these runs. The same is true for runs 2, 5 and 8 where the transverse pitch is 1.5 rather than 1.25. It must be concluded that transverse pitch ratio has more of an effect on gas-side pressure drop than does longitudinal pitch ratio.

B. Effect of Varying Gas Velocity and Coolant Flow Rate

Runs 28-43 were made with constant pitch ratios of 1.5 and .75 inch BWG 14 tubes while varying gas velocity and coolant flow rate. Comparing runs 28 and 40 the effect of reducing gas velocity from 600 to 240 feet/minute with a volumetric flow rate of coolant of 1000 gallons/minute can be seen. The reduction in heat transfer is only 1%. The reduction in ideal fan horsepower, however, is from 15.99 to 0.4. Comparing runs 31 and 43, the gas velocity is again reduced from 600 to 240 feet/minute but the coolant flow rate is 2000 gallons/minute. The resulting reduction in heat transfer is only 1.85%. The ideal fan horsepower is reduced from 11.46 to 0.3. Both exchangers have eight rows. It is obvious from these results that for the range of gas velocities and coolant flow rates considered, heat transfer is water-side limited. One would logically choose the lower gas velocities in order to reduce fan horsepower with little reduction in heat transfer capacity. One

then trades off water velocity with the required total number of tube rows to obtain desired heat transfer capacity. This can be seen in runs 41-43 where flow rates of 1500, 1800 and 2000 gallons/minute of water produce essentially the same heat transfer capacity in ten, nine and eight row exchangers, respectively. It might be desirable to avoid very low coolant flow rates as they tend to produce, for a given heat transfer capacity, a higher cooling water temperature rise.

C. Effect of Tube Outside Diameter

Comparing runs 9, 18 and 27, the tube pitch ratios are constant at 1.75, the gas velocity is 480 feet/minute and coolant flow is 1500 gallons/minute. All tubes are BWG 14 but the tube outer diameters are 1.0, 0.75 and 0.5 inches respectively. The heat transfer rate is seen to increase from 11951 to 14337 to 16418 Btu/sec, and the 0.5 inch exchanger has nine rather than ten rows. This dramatic increase in capacity stems from the fact that with constant coolant flow rate, the water-side convective heat transfer coefficient h_w varies inversely with inside tube diameter to the 1.8 power. Ideal fan horsepower is essentially constant for all three runs. However, the total number of tubes increases from 355 to 475 to 653. Thus if one desires to increase capacity with no regard to material and fabrication cost, the smaller diameter tubes are most attractive.

D. Effect of Tube Wall Thickness

Runs 44-75 were made with gas face velocity held constant at 240 feet/minute, coolant flow rates of 1200 and 1600 gallons/minute, tube outer diameter of 0.5 inch and tube pitch ratios having all possible combinations of 1.5 and 1.75. Tube wall thickness was varied, using BWG 20, 18, 16 and 12 tubes. Comparing runs 44, 52, 60 and 68, all parameters are constant except wall thickness which is 0.035, 0.049, 0.065, and 0.109 respectively. All are eight row exchangers. The heat transfer capacity is seen to actually increase with an increase in wall thickness, but only slightly. The capacity for run 68 is slightly lower than 60. When wall thickness increases the thermal resistance increases and one would expect heat transfer to decrease. However, we again have

the cooling water convective film coefficient effect to consider. When wall thickness increases, coolant flow area decreases for a given number of tubes. For the constant coolant flow rate, the reduced flow area increases coolant velocity and convective film coefficient. Apparently the increase in thermal resistance due to wall thickening is about counterbalanced by the reduction in resistance due to higher coolant film coefficient. The same observations can be made for runs 45, 53, 61 and 69 where the coolant flow rate is held constant at 1400 gallons/minute rather than 1200. The conclusion is that for constant cooling water flow rate, tube wall thickness has little effect on capacity.

E. Effect of Inclining Tubes

The entire analysis of this report and the results in all 75 computer runs are for horizontal tubes. It is desirable to know what would happen to heat transfer capacity should the entire exchanger be inclined. First let us consider rotation of the exchanger about an axis parallel to the tubes. This condition causes the condensate from the first tube to fall not on the top of but to one side of the top of the tube below it in the third row. Eckert and Drake (3) discuss the Ginabat condenser in which tubes are staggered such that condensate from the first row falls to one side of the top of the tube beneath in the third row. This configuration enhances heat and mass transfer on one side of the lower tube and lowers them on the other side. The net result is that tubes in third row transfer more heat than if they were directly below the first row. Our design with tube rows directly below one another will resemble the Ginabat condenser if our condenser is rotated about an axis parallel to the tube axis. The conclusion is that rotation about this axis enhances overall heat transfer.

Inclination about the pitch axis normal to tube axis will cause condensate to flow axially along a given tube as well as downward. References 1 and 8 both claim that for small angles of inclination the

overall heat transfer is little affected by inclination. Reference 8, Fig. 4 shows that 30 degrees of inclination reduces performance by only 5% and that performance is unaffected by gas velocity.

F. Required Fan System

The choice of a fan or fans for this application is a compromise between a uniform air flow distribution and the small minimal head room determined as a space constraint. At a 240 feet/minute face velocity a typical configuration (run 43) would have a face area of 32.4 ft^2 and pass a flow of $7770 \text{ ft}^3/\text{minute}$ with a pressure drop of 0.009 psi or 0.2458 inches of water. A single fan with a 30 inch diameter, a 30° pitch and rotating at 1140 rpm could supply the required amount of gas. It would be necessary to locate the fan at least one diameter above the exchanger to attain any semblance of flow uniformity and even then a diffusing structure would be required. Since this amount of head room is not available, three alternatives present themselves. To use four fans on each 62 x 74 inch installation, each fan blowing into one quadrant of the heat exchanger would require a flow of only $1925 \text{ ft}^3/\text{minute}$. If four 30 inch fans were used to cover the heat exchanger face area they could be belt driven at speeds as low as 500 rpm by four separate explosion proof motors located beside the heat exchanger. This configuration would require an additional 8 inches in plane length at either end of the heat exchange but could be constructed with head room of the order of 18 inches. A second alternative would be the use of a single 60 inch fan similarly belt driven at very low speeds. This would give somewhat poorer but still adequate flow distribution. The final choice might be the remote location of the fan and motor connected to the heat exchanger by a duct terminating in a baffled diffuser.

For low head rises a wide choice of adequate commercial fan blades exist. For larger losses (greater than 2 inches of water) high speed blowers may be required if smaller diameter fans are used.

G. Capacity Conservatism

In the analysis it was assumed that the effect of gas viscous drag on the condensate film thickness would be neglected. With drag present the film thickness will be less than predicted and heat transfer will actually be larger than the values listed in Appendix C. However, in all the runs it was found that heat transfer is cooling water limited and so the effect of condensate thinning would not be great. Reference 1 cites several experiments which are compared to the Nusselt theory. Conservatism in the theory exists due to: a) presence of local dropwise rather than film condensation, b) gas drag, c) local turbulence in the film due to dripping from one tube to the next and d) local ripples in the film due to film instability. Jakob recommends augmenting film coefficients by 10% to account for this conservatism. Had this been done here there would have been an insignificant increase in overall performance.

H. Overall Conductance

The overall exchanger conductance is defined as:

$$\bar{U} = \frac{\delta}{T_g - \bar{T}_w} \quad (40)$$

and varies from row to row in an exchanger. It was requested that it be determined whether an exchanger with the following characteristics would experience a conductance of 516 Btu/hr-ft²-°F when the flow rate at the exchanger exit is 1000 SCFM of non-condensables:

Tube pitch:	1.25 (triangular)
Tube diameter:	1 inch
Tube wall:	.165 inch
Cooling water velocity:	9.0 feet /second (5000 gallons/minute)
Overall length:	9.03 feet
Overall width:	5.21 feet

Three computer runs were made with this geometry and having gas inlet velocities of 480, 360 and 240 feet/minute. The desired mass flow rate of non-condensables equivalent to 1000 scfm is 32.1 lbm/minute which corresponds to an inlet gas velocity of 102 feet/minute. Table I gives the results for these three runs. The data show that an exchanger with 480 feet/minute inlet face velocity cannot meet capacity requirements and the lower inlet velocities have slightly less capacity. The 240 feet/minute inlet velocity case has a gas Reynolds number of 14760 in the last row, which would be about 6300 for an inlet velocity of 102 feet/minute. One might question the use of turbulent heat transfer correlations for the latter rows of the low flow case but, if they were to apply, the capacity for the low flow case would be less than for an inlet velocity of 240 feet/minute. If one were to use laminar correlations there the discrepancy between the 240 and 102 feet/minute cases would be even larger. The conclusion must be that an exchanger of this geometry and with 102 feet/minute inlet gas velocity will have conductance of less than $516 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$.

TABLE I

Average Overall Conductance of a ten row, 1 inch tube/diameter exchanger with 1.25 tube pitches and 5000 gallons/minute cooling water and face dimensions of 9.03 by 5.21 feet.

Face Velocity (feet/minute)	Flow Rate Non-cond. (lbm/minute)	Average Overall Conductance (Btu/hr-ft ² -°F)
480	150.6	412.2
360	112.9	411.0
240	75.3	406.6

Fig. 1—Schematic diagram of tube bundle-fan system to be analyzed

IV. ANALYSIS

Figure 1 shows the system to be analyzed. It is a staggered bank of single-pass tubes which are fed from a common header. Cooling water is pumped through the tubes while a fan/blower forces the steam-air-hydrogen mixture down through the tube bundle. Steam condenses on the cool tube walls and the difference in steam density between the free stream and the area adjacent to the condensate on the tubes causes steam to diffuse through the non-condensable gases toward the tubes. The steam which has condensed on the tube walls flows downward due to gravity, dripping on the next lower tube. As the free stream gas mixture flows downward and steam is condensed out, the partial pressure of steam in the free stream decreases while gas total pressure remains constant. The change in gas composition from row-to-row changes the rate of diffusion of steam toward the tubes. The object of this analysis is to determine in each tube row the rate at which energy is removed from the gas, the gas pressure drop and the water pressure drop in the tubes.

A. Gas-Film-Cooling Water Energy Balance

Consider a single tube in the exchanger, as shown in Fig. 2. In the steady state the rate at which energy is removed from the gas must equal the rate at which energy crosses the condensate film as well as the rate at which energy flows through the tube wall to the cooling water. That is:

$$\dot{q} = \dot{q}_g = \dot{q}_f = \dot{q}_w \quad (1)$$

What must be done is to derive expressions for the three heat transfer rates and determine a set of temperatures T_f and T_s for the given T_g and T_w such that the rates $\dot{q}_g$, $\dot{q}_f$ and $\dot{q}_w$ are equal.

Fig. 2—Schematic of an exchanger tube with free-stream energy removal, condensate film heat transfer and heat transfer to cooling water

B. Heat and Mass Transfer From Free Stream

The rate at which steam mass is transferred from the free stream to the condensate film on a tube is defined by:

$$\dot{m}_{sc} = h_D (\rho_{sg} - \rho_{sf}) \quad (2)$$

The steam in the free stream is considered saturated at free stream temperature T_g . Also, the steam at the condensate surface is saturated at that surface temperature, T_f . The saturation density of steam is tabulated as a function of temperature (Ref. 7); when both T_g and T_f are known, the densities in Eq. (2) can be determined. The diffusion coefficient h_D can be determined from the heat-mass transfer analogy (Ref. 1, 2, 3). The analogy is that for a convecting system, if one knows Nusselt number, Nu , as a function of Reynolds number, Re , and Prandtl number, Pr :

$$Nu = \frac{h d_{to}}{k} = f(Re, Pr) \quad (3)$$

then the corresponding Sherwood number, $h_D d_{to}/D$, is given by the same relationship but in terms of the Reynolds number and Schmidt number, Sc . That is:

$$Sh = \frac{h_D d_{to}}{D} = f(Re, Sc) \quad (4)$$

For the situation at hand, Ref. 2 presents the correlation of Grimson for heat transfer from air to tube bundles. The correlation is of the form:

$$Nu = b Re^n \quad (5)$$

where b and n are functions of lateral and transverse tube pitch to tube diameter ratios, x_1 and x_t . Equation (5) is not exactly in the form

desired since it does not contain the Prandtl number. The coefficient b can be transformed to a form:

$$Nu = b^* Re^n (Pr)^{1/3} = b Re^n \quad (6)$$

or

$$b^* = \frac{b}{Pr^{1/3}} \quad (7)$$

The one-third power of Prandtl number is characteristic of correlations of the data for forced convection heat transfer to cylindrical surfaces. Grimison's data is for air and so a Prandtl number $Pr = 0.72$ was used in Eq. (7) to determine values of b^* . Note that the Prandtl number of air is rather insensitive to temperature and taking the one-third power further reduces error which might occur in not knowing the temperature applicable to Grimison's data. Actually Grimison correlated the data of many investigators. Table II presents the Grimison exponents and revised coefficients. The diffusion coefficient is now given by:

$$h_D = \frac{D}{d_{to}} b^* Re^n Sc^{1/3} \quad (8)$$

Reference 2 states that the values of the coefficients b and exponents n are for ten row exchangers. He also shows how the coefficients for exchangers with lesser and greater numbers of rows are obtained. This is done by multiplying the values of b^* by a modifying factor, F_1 , which is a function of the number of rows in the exchanger. It is possible to generate modifying factors for individual rows from these data. Table III presents the data from Ref. 2 in the second column in the table. The third column in the table is the value of the modifying factor for a given row that would be necessary to give the average value for an exchanger with the given number of rows. Taking the data in column 3 of Table III, one can check the results with actual data for air.

TABLE II

Values of the coefficient b^* and the exponent n for Eq. (6).
(Ref. 2, modified for inclusion of Prandtl number.)

$x_t \rightarrow$ $x_1 \downarrow$	1.25		1.50		2.00		3.00	
	b^*	n	b^*	n	b^*	n	b^*	n
.60	--	--	--	--	--	--	.238	.636
.90	--	--	--	--	.498	.571	.448	.581
1.0	--	--	.555	.558	--	--	--	--
1.125	--	--	--	--	.533	.565	.578	.560
1.25	.578	.556	.564	.554	.579	.556	.583	.562
1.5	.503	.568	.513	.562	.504	.568	.545	.568
2.0	.451	.572	.464	.568	.538	.556	.501	.570
3.0	.346	.592	.397	.580	.491	.562	.470	.574

TABLE III

Row-dependent modifying factors F_i
for Eq. (9) (Ref. 2).

Row	Avg. modifier for these rows	Modifier for this row, F_i
1	.68	.68
2	.75	.82
3	.83	.99
4	.89	1.07
5	.92	1.04
6	.95	1.10
7	.97	1.09
8	.98	1.05
9	.99	1.07
10	1.0	1.09
12	1.01	1.06
15	1.022	1.08

Reference 5, Fig. 17-25, gives average Nusselt number on an individual tube in a ten row exchanger. If one takes the value for each row and finds an average for ten rows and divides this into the value for a given row, the result should be equivalent to column 3 of Table III. This is shown in Fig. 3 where the reduced data of Snyder and Table III are presented. The agreement appears good. The mass transfer coefficient now becomes:

$$h_{oi} = \frac{D}{d_{to}} F_i b^* Re^n Sc^{1/3} \quad (9)$$

Finally, Ref. 3 states that the mass transfer coefficient must be modified by a pressure correction term to compensate for a convective flow which accompanies diffusion. The pressure correction term is given by:

$$\bar{P} = \frac{P_t}{(P_{gf} - P_{gg})} \ln \left(\frac{P_{gf}}{P_{gg}} \right) \quad (10)$$

Combining Eqs. (2), (9) and (10) we have:

$$\dot{m}_{sc} = \frac{D}{d_{to}} F_i b^* \bar{P} Re^n Sc^{1/3} (P_{sg} - P_{sf}) \quad (11)$$

The total rate at which energy is removed from the gas is given by an energy balance at the surface of the condensate. The balance considers the rate at which mass is being condensed and the rate at which heat is convected. That is:

$$\dot{q}_g = \dot{m}_{sc} \lambda + h (T_g - T_f) \quad (12)$$

where the latent heat λ is evaluated at T_f and the convective heat transfer coefficient comes from Eq. (6):

$$h = \frac{k}{d_{to}} b^* F_i Re^n Pr^{1/3} \quad (13)$$

Fig. 3—Comparison of the modifying factor F_i , Table II, for an individual exchanger row with normalized data for air (reference 5)

The foregoing suffices to define the rate at which energy is removed from the free stream, except for defining fluid properties which are required for the dimensionless numbers Re, Pr, Nu and Sc. Appendix A shows how these properties for the gas mixture are defined.

C. Heat Transfer Across Film (Ref. 1)

Consider Fig. 4 which depicts a segment of tube surface on which steam is condensing. The condensate film has a local thickness Y which varies with ϕ . At any value of y the shear stress is given by:

$$s = \frac{\mu}{g_c} \frac{dv}{dy} \quad (14)$$

A force balance on the condensate element gives:

$$ds (r+y) d\phi = -\gamma_w \sin \phi (r+y) d\phi dy$$

or

$$\frac{ds}{dy} = -\gamma_w \sin \phi \quad (15)$$

Combining Eqs. (14) and (15) we obtain:

$$\frac{d^2 v}{dy^2} = -\frac{g_c \gamma_w}{\mu} \sin \phi$$

Integrating twice we have:

$$v = -\frac{g_c \gamma_w}{2\mu} y^2 \sin \phi + C_1 y + C_2 \quad (16)$$

At the wall, $y = 0$, the velocity is zero. Thus $C_2 = 0$. At the film-gas interface, $y = Y$, the gas produces a tangential drag defined by:

$$\mu \frac{dv}{dy} = C_D \frac{\rho_g V_g^2}{2} \quad (17)$$

24

Equations (16) and (17) combine to yield:

$$C_i = C_D \frac{\rho_g V_g^2}{2\mu} + \frac{g_c \gamma_w Y}{\mu} \sin \phi$$

and Eq. (16) becomes:

$$N = -\frac{g_c \gamma_w Y^2}{2\mu} \sin \phi + Y \left(C_D \frac{\rho_g V_g^2}{2\mu} + \frac{g_c \gamma_w Y}{\mu} \sin \phi \right) \quad (18)$$

The average film velocity is given by:

$$N_m = \frac{1}{Y} \int_0^Y N(Y) dY \quad (19)$$

Applying Eq. (18) in (19) we have:

$$N_m = \frac{g_c \gamma_w Y^2}{3\mu} \sin \phi + C_D \frac{\rho_g V_g^2}{4\mu} Y \quad (20)$$

The rate at which mass flows along the tube surface per unit of tube length is:

$$\begin{aligned} \dot{m}_c &= \rho_w Y N_m \\ d\dot{m}_c &= \rho_w d(Y N_m) \end{aligned} \quad (21)$$

Substituting Eq. (20) into (21) gives :

$$d\dot{m}_c = \rho_w \left(\frac{g_c \gamma_w}{3\mu} (Y^3 \cos \phi d\phi + 3Y^2 \sin \phi dY) + C_D \frac{\rho_g V_g^2}{2\mu} Y dY \right) \quad (22)$$

Dwg. 860A170

Fig. 5—Segment of tube surface and condensate film - showing contributions to the energy balance

An energy balance for the segment of condensate shown in Fig. 5 gives:

$$dq_f = \frac{k_w r (T_f - T_s)}{Y} d\phi = \lambda dm_c + h(T_g - T_f) r d\phi \quad (23)$$

Substituting Eq. (22) into (23) yields:

$$d\phi \cdot \frac{k_w r (T_f - T_s)}{Y} = \frac{g_c \rho_w \lambda}{\mu} \sin \phi Y^2 dY + \frac{g_c \rho_w \lambda}{3\mu} \cos \phi Y^3 d\phi$$

$$\frac{\lambda \rho_w C_0 \rho_g V_g^2}{2\mu} Y dY + h(T_g - T_f) r d\phi \quad (24)$$

The contribution to the energy balance of convective heat transfer can be neglected when compared to the mass transfer contribution. The drag contribution is quite difficult to evaluate as no data are available on the shear drag term C_D in tube bundles. If we neglect the drag term we know the solution for condensate film thickness will be conservatively large. Neglecting convective heat transfer contribution and gas drag Eq. (24) becomes:

$$B d\phi = \frac{3\mu k_w r (T_f - T_s)}{\lambda \rho_w \rho_g g_c} d\phi = Y^4 \cos \phi d\phi + \frac{3}{4} d(Y^4) \sin \phi$$

Reference 1 shows that the substitution $Z = \psi^4 = Y^4/B$ gives:

$$1 = Z \cos \phi + \frac{3}{4} \frac{dZ}{d\phi} \sin \phi$$

which has the solution:

$$Z = \frac{1}{\sin^{4/3} \phi} \left(\frac{4}{3} \int_0^\phi \sin^{1/3} \phi d\phi + C \right) \quad (25)$$

For the top tube in a bundle the film thickness must be finite at $\phi = 0$. Thus the constant $C = 0$ for the first row. Reference 1 has tabulated Z as a function of ϕ which was obtained by graphical integration. These

authors have performed the integration by Gaussian quadrature and consider their results more accurate. They are listed in Table IV for the first tube row.

Once the function Z is defined one can define a local film heat transfer coefficient, given by:

$$h_f = \frac{k_w}{Y}$$

and the mean coefficient for the tube is:

$$h_{fm} = \frac{1}{\pi} \int_0^{\pi} \frac{k_w}{Y} d\phi = \frac{1}{\pi^4 \sqrt{B}} \int_0^{\pi} \frac{k_w}{\psi} d\phi \quad (26)$$

For the first tube row the tabulated values of ψ have been used and the integral in Eq. (26) evaluated, again using Gaussian quadrature. The result is, for the first row:

$$h_{fm} = \frac{.728 k_w}{\sqrt[4]{B}}$$

Reference 1 states (confirmed by the authors) that the average mean heat transfer coefficient for N_R rows is given by:

$$\bar{h}_{fm} = \frac{.728 k_w}{\sqrt[4]{N_R B}} \quad (27)$$

Equation (27) can be used to define the mean coefficient for each row. The first row's condensate drips on the third tube row and thus:

$$\frac{h_{fm1} + h_{fm3}}{2} = \frac{.728 k_w}{\sqrt[4]{2B}}$$

or

$$h_{fm3} = \frac{.728 k_w}{\sqrt[4]{B}} \left(\sqrt[4]{2} - 1 \right)$$

In the same manner the mean coefficient for the 5th, 7th, etc. rows can be determined. Likewise the second tube row has a film coefficient the

TABLE IV

Tabulated values of the parameter $1/\psi$ as a function of local angle ϕ .
Obtained by Gaussian quadrature

ϕ	$1/\psi$	$4/3 \int_0^\phi \sin^{1/3} \phi d\phi$	ϕ	$1/\psi$	$4/3 \int_0^\phi \sin^{1/3} \phi d\phi$
0	1.0000	.0000	100	.8413	1.9559
5	1.0069	.0376	110	.8055	2.1859
10	1.0013	.0964	120	.7649	2.4110
20	.9950	.2440	130	.7186	2.6286
30	.9869	.4183	140	.6651	2.8358
40	.9759	.6115	150	.6016	3.0289
50	.9619	.8187	160	.5227	3.2033
60	.9447	1.0363	170	.4124	3.3509
70	.9242	1.2614	175	.3263	3.4097
80	.9003	1.4913	180	0	3.4475
90	.8727	1.7236			

same as the first row, from which the mean coefficient for the 4th, 6th, etc. rows can be determined. Table V gives the values of the numerical coefficients for each row thus derived. As one might expect the buildup of condensate on the higher numbered rows causes a lower film coefficient there. For the i^{th} row, now that its mean film coefficient is defined, the heat transfer rate across the film is given by:

$$q_{fi} = h_{fmi} (T_f - T_s) \quad (28)$$

D. Tube to Cooling Water Heat Transfer

Heat transfer from the outer surface of a tube to the cooling water is depicted schematically in Fig. 6. An energy balance for the differential segment of the tube gives:

$$\dot{m}_w C_w dT_w = dq_w = U (T_s - T_w(x)) dx \quad (29)$$

where

$$U = \frac{1}{\frac{1}{\pi d_{ti} h_w} + \frac{\ln(d_{to}/d_{ti})}{2\pi k_t} + \frac{ff_o}{\pi d_{to}} + \frac{ff_i}{\pi d_{ti}}} \quad (30)$$

and the convective film coefficient for the water, as recommended by Ref. 2 is:

$$Nu_w = \frac{h_w d_{ti}}{k_w} = 0.023 \left(1 + \left(\frac{d_{ti}}{L} \right)^{.7} \right) Re_w^{.8} Pr_w^{1/3} \quad (31)$$

Equation (29) can be rearranged and integrated from water inlet to exit:

$$\int_{T_{wi}}^{T_{wo}} \frac{dT_w}{T_w - T_s} = - \int_0^L \frac{U dx}{\dot{m}_w C_w}$$

TABLE V

Values of the condensate film heat transfer coefficient parameter J_i for the first 20 rows in a staggered tube bundle

$$\left(h_{fmi} = \frac{k_w}{4\sqrt{B}} J_i \right)$$

Row	J_i	Row	J_i
1	.7276	11	.3551
2	.7276	12	.3551
3	.4940	13	.3405
4	.4940	14	.3405
5	.4331	15	.3285
6	.4331	16	.3285
7	.3978	17	.3184
8	.3978	18	.3184
9	.3734	19	.3096
10	.3734	20	.3096

Dwg. 860A172

Fig. 6—Thermal schematic of a section of tubing - to determine heat transfer from outer tube surface to the cooling water

which integrates to :

$$T_s = \frac{T_{wo} - T_{wi} e^{-UL/\dot{m}_w C_w}}{1 - e^{-UL/\dot{m}_w C_w}} \quad (32)$$

Further, if the overall heat transfer rate to the water in a single tube is q_w , then:

$$\dot{q}_w A_w = \dot{m}_w C_w (T_{wo} - T_{wi}) = \dot{q}_g A_g$$

or

$$T_{wo} = T_{wi} + \frac{\dot{q}_g A_g}{\dot{m}_w C_w} \quad (33)$$

Substituting Eq. (33) into (32) gives:

$$T_s = T_{wi} + \frac{\dot{q}_g A_g / \dot{m}_w C_w}{1 - e^{-UL/\dot{m}_w C_w}} \quad (34)$$

E. Overall Heat Transfer

Sections B, C and D have derived expressions which describe the heat transfer rates q_g , q_f and q_w in terms of the governing temperatures T_g , T_f , T_s and T_{wi} . The following procedure has been used to find the correct values of these temperatures which force the heat transfer rates to be equal:

1. Compute the gas transport properties applicable to Section B using the gas state at the inlet of a tube row, T_{gi} .
2. Assume a gas-film interface temperature T_f and compute q_g as outlined in Section B.
3. Multiplying q_g by the tube outside to inside surface area ratio to get q_w and find T_s from Section D, Eq. (34).
4. Knowing T_f , q_g and the value of T_s required to make $q_g = q_w$, return to the analysis in Section C and compute q_f . If $q_f \neq q_g$ then adjust T_f and return to Step 2 above.

F. Gas-Side Energy Balance

Once the iteration in Section E is completed the effect of heat and mass transfer on the gas state must be introduced in order to determine the proper gas inlet state for the next tube row. The following iterative procedure has been used. It is based upon the fact that the non-condensable gas mass flow rate must not change from tube row to tube row:

1. At the inlet to the exchanger compute the mass flow rate of non-condensable gases, $\dot{m}_{ni}$.
2. Establish the mass flow rate of steam entering a row. This is accomplished by knowing the volumetric flow rate in, $Q_{gi} = V_{gi}A$, and the steam density at inlet [since the steam in the free stream remains saturated, $\rho_{si} = \rho_{sat}(T_{gi})$]. Then $\dot{m}_{si} = \rho_{si}Q_{gi}$.
3. Having solved for $q_g = q_f = q_w$ in the row, find the loss of steam due to condensation in the row - Eq. (11) - multiplied by the total tube surface area in the row.
4. Assume a gas temperature leaving the row, T_{go} .
5. Compute saturated steam pressure P_{so} and steam density ρ_{so} at the assumed exit temperature.
6. Compute non-condensable gas pressure from $P_{no} = P_t - P_{so}$.
7. Compute steam mass flow rate at the exit of the row from $\dot{m}_{so} = \dot{m}_{si} - \dot{m}_{sc}A_g$.
8. Compute exit volumetric flow rate of gas from $Q_{go} = \dot{m}_{so}/\rho_{so}$.
9. Compute non-condensable density at exit condition $\rho_{no} = \frac{P_{no}}{R_n T_{go}}$.
10. Compute mass flow rate of non-condensables at the exit from $\dot{m}_{no} = \rho_{no}Q_{go}$.

11. If the value $\dot{m}_{no} \neq \dot{m}_{ni}$, assume a new gas exit temperature and return to Step 4.

G. Water-Side and Gas-Side Pressure Drop

The water-side pressure drop was computed from the relationship:

$$\Delta P_w = \left(\frac{4 f_w L}{d_{ti}} + 1.5 \right) \frac{\rho_w V_w^2}{2 g_c} \quad (35)$$

where the friction factor f_w is given by Ref. 2 as:

$$f_w = 0.0014 + 0.125 / Re_w^{.32}$$

and the factor 1.5 in Eq. (35) accounts for losses encountered in entering and leaving the tubes. There will be added pressure losses in the exchanger headers and in the connecting lines. The losses in the headers have been assumed to be 1/2 the loss given by Eq. (35) which can be attained by proper header design. This is explained in Section III.

Gas-side pressure drop was computed from correlations given in Ref. 5. For lower Reynolds numbers Ref. 5 gives:

$$\Delta P_g = \frac{2 f_g \rho_g V_{gm}^2}{g_c} N_R \quad (36)$$

where f_g is defined by:

$$f_g = .75 Re_c^{-.2} \quad \left(100 \leq \frac{(S_t - d_{to}) \rho_g V_{gm}}{\mu_g} \leq 20000 \right) \quad (37)$$

while Ref. 5 uses the same expression [Eq. (36)] for ΔP_g but where f_g is given as:

$$f_g = Re_o^{-.16} \left(0.25 + \frac{0.1175}{(S_t/d_{to} - 1)^{1.08}} \right) \quad \left(5000 \leq \frac{\rho_g d_{to} V_{gm}}{\mu_g} \leq 40000 \right) \quad (38)$$

Fig. 7—Logic flow diagram for the digital computer program listed in Appendix B

In the analysis the Reynolds number based on tube diameter was checked and if it was in the range specified by Eq. (38), that equation was used. Otherwise Eq. (37) was used to define f_g .

Equation (36) gives gas pressure drop for dry heat transfer. With the addition of mass transfer there is a reduction of ΔP_g due to deceleration of some of the steam mass (that which was condensed). A momentum balance across a tube row gives:

$$F = A \left(\frac{m_g V_g}{g_c} \right) = \frac{P_{go} Q_{go}}{A g_c} (Q_{gi} - Q_{go})$$

and the force/unit area is the pressure rise due to mass flow rate change. This is:

$$\frac{F}{A} = \Delta P_{gc} = \frac{P_{go} Q_{go}}{A^2 g_c} (Q_{gi} - Q_{go}) \quad (39)$$

The total gas pressure drop for a row is given by Eq. (36) minus Eq. (39).

H. Evaluation of Analysis

The analysis presented in Sections II-B through G requires iterative solution, iteration being required to solve for the proper film temperature T_f and for gas exit temperature T_{go} . A digital computer program has been written in the ALGOL language to evaluate the analysis. Appendix B is a program listing with corresponding nomenclature. Figure 7 is a logic flow diagram which indicates the steps taken to perform the basic calculations and the method of iteration.

The program accepts as data the applicable gas inlet thermodynamic state, gas inlet velocity, coolant inlet state and volumetric flow rate and the geometric variables required to define the exchanger: tube longitudinal and transverse pitch ratios, tube inside and outside diameters, maximum exchanger length and width and the maximum-desired number of tube rows. The maximum width is slightly adjusted downward so

Dwg. 860A167

Fig. 8—Schematic diagram of the assumed inlet pipe configuration for a header

that the number of tubes in the direction transverse to gas flow is an integer. That is, given a maximum width, the program finds the maximum number of tubes possible in the transverse direction (for the given transverse tube pitch ratio and tube outer diameter) that will fit in this width and adjusts width downward so that the shroud touches the outer tubes.

Actual tube length is determined from the input maximum length and the space required for headers at either end of the exchanger. Headers are assumed as having three equally spaced inlet ports as shown in Fig. 8. The depth of a header in the direction of coolant flow Z_n is determined by allowing the pressure drop in a header to be one half that which occurs in the tubes (including entrance and exit losses), Eq. (35). This insures that the flow of coolant will be essentially uniform in all tubes. Required header depth is subtracted from maximum length to give the actual tube length.

The total number of tubes required for the desired number of tube rows is determined after the number of tubes in the first row is computed (see discussion of width calculations above). The tubes being staggered, the second row has one less tube than the first, the third the same as the first, etc. Knowing the total number of tubes, the flow of coolant is computed as being distributed equally in each tube. The flow per tube is used to next compute the coolant pressure drop in flowing through a tube. Finally, the gas inlet densities and mass flow rates are determined, as well as the coefficient and exponent for the Grimson heat transfer correlation.

The first iteration loop is then entered wherein the proper values of T_f and T_s are determined in order that $q_g = q_f = q_w$ for a given row. When convergence is achieved the iteration loop to determine the gas thermodynamic exit state is entered. When a gas exit temperature T_{go} is found such that the mass flow rate of air and hydrogen leaving the row equals the rate entering the row, the program prints the results for the given row and sets the gas inlet state for the next row. The process is repeated until either all rows are considered or until the

total heat transfer capacity of all rows previously considered equals or exceeds 15000 Btu/sec, the desired capacity of the entire exchanger. The final overall results are printed, including gas pressure drop and ideal fan horsepower, and the program returns to consider the next size heat exchanger. This continues until all input exchanger sizes have been dealt with.

NOMENCLATURE

A	Face area of exchanger, ft^2 .
A_{ij}	A weighting factor for components i and j in the correlation for thermal conductivity of the gas mixture, defined in Eq. (A-4), dimensionless.
A_g	Heat transfer area on the outside surface of a single exchanger tube, ft^2 .
A_w	Heat transfer area on the inside surface of a single exchanger tube, ft^2 .
B	A thermal property parameter, $3\mu k_w r(T_f - T_s)/\lambda \gamma_w \rho_w$, ft^4 .
b	The coefficient in the correlation for convective heat transfer for a gas flowing over a tube bundle, dimensionless.
b^*	The same as b but modified for Prandtl number, Eq. (7), dimensionless.
C	Integration constant, Eq. (25), dimensionless.
C_D	Viscous drag coefficient for gas flowing over tubes in a bundle, dimensionless.
C_w	Specific heat of water, $\text{Btu/lbm-}^\circ\text{F}$.
C_{pi}	Specific heat of the i^{th} gas component, $\text{Btu/lbm-}^\circ\text{F}$.
C_{pm}	Specific heat of the gas mixture, $\text{Btu/lbm-}^\circ\text{F}$.
C_1, C_2	Integration coefficients, Eq. (16), dimensionless.
D	Diffusion coefficient for steam through air and hydrogen, ft^2/sec .
D_{ij}	Diffusion coefficient for component i through component j, ft^2/sec , except in Eq. (A-7) where units are cm^2/sec .

d_{ti}	Inside tube diameter, ft.
d_{to}	Outside tube diameter, ft.
F	Force on gas caused by change in gas momentum, lbf.
F_i	Row-dependent factor for modifying the mass transfer coefficient for flow in a tube bundle, Eq. (9) and Table II, dimensionless.
f_g	Friction factor for gas flow over tubes in a bundle, dimensionless.
f_w	Friction factor for water flowing in tubes, dimensionless.
ff_i	Fouling factor for heat transfer, on inside wall of tube, $\text{sec-ft}^2\text{-}^\circ\text{F/Btu}$.
ff_o	Fouling factor for heat transfer, on outside wall of tube, $\text{sec-ft}^2\text{-}^\circ\text{F/Btu}$.
g	Acceleration due to gravity, ft/sec^2 .
g_c	Gravitational constant, lbm-ft/lbf-sec^2 .
h	Gas convective heat transfer coefficient in a tube bundle, $\text{Btu/sec-ft}^2\text{-}^\circ\text{F}$.
h_D	Mass transfer coefficient, from gas to condensate film, ft/sec.
h_{Di}	Mass transfer coefficient for the i^{th} row of a tube bundle, ft/sec.
h_f	Local heat transfer coefficient across the condensate film $\text{Btu/sec-ft}^2\text{-}^\circ\text{F}$.
h_{fm}	Average heat transfer coefficient across the condensate film, $\text{Btu/sec-ft}^2\text{-}^\circ\text{F}$.
$\bar{h}_{fm}$	Average heat transfer coefficient across the condensate film for the first N_R rows of an exchanger, $\text{Btu/sec-ft}^2\text{-}^\circ\text{F}$.
h_{fmi}	Average heat transfer coefficient across the condensate film for the i^{th} row in an exchanger, $\text{Btu/sec-ft}^2\text{-}^\circ\text{F}$.
h_w	Convective heat transfer coefficient for cooling water, $\text{Btu/sec-ft}^2\text{-}^\circ\text{F}$.

J_i	Row-dependent numerical coefficient for heat transfer coefficient across the condensate film, Table IV, dimensionless.
K	Boltzmann's constant, joules/°K.
k	Thermal conductivity, Btu/sec-ft-°F.
k_i	Thermal conductivity of the i^{th} gas component, Btu/sec-ft-°F.
k_t	Thermal conductivity of the tube wall, Btu/sec-ft-°F.
k_w	Thermal conductivity of condensate water, Btu/sec-ft-°F.
L	Active length of heat exchanger tubes, ft.
M_i	Molecular weight of the i^{th} gas component, lbm/mole.
$\dot{m}_c$	Mass flow rate of condensate along a tube surface per unit of tube length, lbm/sec-ft.
$\dot{m}_{ni}$	Mass flow rate of non-condensable gases entering a tube row, lbm/sec.
$\dot{m}_{no}$	Mass flow rate of non-condensable gases leaving a tube row, lbm/sec.
$\dot{m}_{sc}$	Mass flow rate of steam being condensed on a tube, lbm/ft ² -sec.
$\dot{m}_{si}$	Mass flow rate of steam entering a tube row, lbm/sec.
$\dot{m}_{so}$	Mass flow rate of steam leaving a tube row, lbm/sec.
$\dot{m}_w$	Mass flow rate of cooling water in a single tube, lbm/sec.
N_R	Row number, dimensionless.
n	Exponent in the correlation for convective heat transfer for a gas flowing over a tube bundle, Eq. (5) and Table I, dimensionless.
Nu	Gas Nusselt number, hd_{to}/k , dimensionless.
Nu_w	Cooling water Nusselt number, dimensionless.

$\bar{P}$	A pressure correction term for the mass transfer coefficient to compensate for convective flow. Total pressure divided by log-mean pressure difference of non-condensables between the condensate film and the free stream, dimensionless.
P_i	Partial pressure of the i^{th} gas component, lb/ft^2 .
P_{gf}	Partial pressure of non-condensable gases at the condensate film surface, lb/ft^2 .
P_{gg}	Partial pressure of non-condensable gases in the free stream, lb/ft^2 .
P_{ij}	The sum of the partial pressures of gas components i and j in the free stream, Eq. (A-7), atmospheres.
P_{no}	The partial pressure of non-condensable gases leaving a tube row, lb/ft^2 .
P_{so}	The partial pressure of steam leaving a tube row, lb/ft^2 .
P_t	Total gas pressure, lb/ft^2 .
Pr	Prandtl number of gas, $\mu_g C_{pm}/k$, dimensionless.
Pr_w	Cooling water Prandtl number, dimensionless.
q	Heat transfer rate, $\text{Btu}/\text{sec}\text{-ft}^2$.
q_f	Heat transfer rate across the condensate film, $\text{Btu}/\text{sec}\text{-ft}^2$.
q_{fi}	Heat transfer rate across the condensate film in the i^{th} tube row, $\text{Btu}/\text{sec}\text{-ft}^2$.
q_g	Rate at which energy is removed from the gas stream, $\text{Btu}/\text{sec}\text{-ft}^2$.
Q_{gi}	Volumetric flow rate of gas into a tube row, ft^3/sec .
Q_{go}	Volumetric flow rate of gas out of a tube row, ft^3/sec .
Q_t	Total heat transfer rate for a given tube row, Btu/sec .
q_w	Heat transfer rate from outer tube surface to cooling water, $\text{Btu}/\text{sec}\text{-ft}^2$.

Q_w	Total volumetric flow rate of cooling water, ft^3/sec .
r	Tube outer radius, ft.
Re	Reynolds number, $V_g \rho_g d_{to} / \mu_g$.
Re_D	Reynolds number based on outer tube diameter, Eq. (38), dimensionless.
Re_c	Reynolds number based on transverse flow opening between tubes, Eq. (37), dimensionless.
Re_w	Reynolds number of cooling water, dimensionless.
R_i	Gas constant for the i^{th} non-condensable gas, $\text{ft-lbf/lbm-}^\circ\text{R}$.
R_n	Gas constant for the non-condensable gas mixture, $\text{ft-lbf/lbm-}^\circ\text{R}$.
S	Shear stress in condensate film, lbf/ft^2 .
Sc	Schmidt number, $\mu_g / \rho_g D$, dimensionless.
Sh	Sherwood number, $h_D d_{to} / D$, dimensionless.
S_i	Sutherland constant in the correlation for diffusion coefficient, for component i , $^\circ\text{K}$.
S_{ij}	Sutherland constant in the correlation for diffusion coefficient, for mixture of components i and j , $^\circ\text{K}$.
S_l	Longitudinal tube pitch, ft.
S_t	Transverse tube pitch, ft.
T_{bi}	Boiling point for the i^{th} gas component, $^\circ\text{K}$.
T_f	Temperature at the outer surface of condensate film, $^\circ\text{F}$.
T_g	Free stream gas temperature, $^\circ\text{F}$, except Eq. (A-4), (A-7) and (A-8) $^\circ\text{K}$, and Eq. (A-9) $^\circ\text{R}$.
T_{gi}	Free stream gas temperature entering a tube row, $^\circ\text{F}$.
T_{go}	Free stream gas temperature leaving a tube row, $^\circ\text{F}$.
T_s	Temperature on the outer surface of a tube, $^\circ\text{F}$.

T_w	Cooling water temperature, °F.
T_{wi}	Cooling water temperature entering exchanger, °F.
T_{wo}	Cooling water temperature leaving exchanger, °F.
U	Heat transfer conductance between outer tube surface and cooling water, Btu/sec-ft-°F.
$\bar{U}$	Overall conductance between gas and cooling water, Eq. (40), Btu/sec-ft ² -°F.
v	Condensate film velocity, ft/sec.
V_g	Velocity of gas, ft/sec.
V_{gm}	Maximum gas velocity, ft/sec.
V_{gi}	Gas velocity entering a tube row, ft/sec.
v_m	Average condensate film velocity at any angle ϕ , ft/sec.
x	Dimension along the tube, ft.
x_i	Mass fraction of gas component i, dimensionless.
x_l	Longitudinal tube pitch ratio, dimensionless.
x_t	Transverse tube pitch ratio, dimensionless.
y	Direction normal to condensate film surface measured outward from tube surface, ft.
Y	Local condensate film thickness, ft.
y_i	Mole fraction of gas component i, dimensionless.
Z	A film thickness parameter Y^4/B , dimensionless.
Z_n	Depth of an exchanger header, ft.
γ_w	Volumetric weight of condensate, lbf/ft ³ .
ΔP_g	Dry pressure drop of gas across tubes, lbf/ft ² .
ΔP_{gc}	Added gas pressure drop due to condensation, lbf/ft ² .

ΔP_w	Water pressure drop across tubes, lb/ft^2 .
ϵ_i	Force constant for the i^{th} gas component in the diffusion coefficient correlation, Eq. (A-8).
λ	Latent heat of vaporization of steam, Btu/lbm .
μ	Viscosity of condensate film, $\text{lbm}/\text{ft}\cdot\text{sec}$.
μ_g	Gas viscosity, $\text{lbm}/\text{ft}\cdot\text{sec}$.
μ_i	Viscosity of i^{th} gas component, $\text{lbm}/\text{ft}\cdot\text{sec}$.
μ_m	Viscosity of gas mixture, Eq. (A-1), $\text{lbm}/\text{ft}\cdot\text{sec}$.
ρ_g	Gas density, lbm/ft^3 .
ρ_i	Density of the i^{th} gas component, lbm/ft^3 .
ρ_{no}	Density of non-condensable gases leaving a tube row, lbm/ft^3 .
ρ_{sf}	Density of steam at the condensate film surface, lbm/ft^3 .
ρ_{sg}	Density of steam in the free stream, lbm/ft^3 .
ρ_{si}	Density of steam entering a tube row, lbm/ft^3 .
ρ_{so}	Density of steam leaving a tube row, lbm/ft^3 .
ρ_{sat}	Steam saturation density, lbm/ft^3 .
ρ_w	Density of water, lbm/ft^3 .
σ_{ij}	A force constant in the correlation for gas diffusion coefficient, Eq. (A-7), atmospheres.
ϕ	The angle between vertical and any point on a tube, Fig. 4, degrees.
ϕ_{ij}	Weighting factor for components i and j in the correlation for gas mixture viscosity, defined in Eq. (A-2), dimensionless.
ψ	The parameter $Y/\sqrt[4]{B}$, dimensionless.
Ω_D	Collision integral used in the correlation for gas mixture diffusion coefficient, Eq. (A-7).

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APPENDIX A

Thermodynamic Properties of the Gas, Condensate and Coolant

The correlations used to determine heat and mass transfer rates from the gas stream to the condensate film employ the dimensionless Reynolds, Prandtl, Nusselt and Sherwood numbers. These parameters are all dependent upon the thermodynamic properties of the gas stream: dynamic viscosity μ , thermal conductivity k , diffusion coefficient D , density ρ , and specific heat C_p . This appendix presents the methods used to determine these properties for the gas mixture as well as water thermodynamic properties for the condensate film and cooling water.

1. Gas Dynamic Viscosity

Reference 4 outlines the method of Wilkes for determining the viscosity of gas mixtures when the viscosity of the individual components and their mole fractions are known. The viscosity of the mixture is given by:

$$\begin{aligned} \mu_m = & \frac{\mu_s}{1 + (y_a/y_s)\phi_{sa} + (y_h/y_s)\phi_{sh}} + \frac{\mu_a}{1 + (y_s/y_a)\phi_{as} + (y_h/y_a)\phi_{ah}} \\ & + \frac{\mu_h}{1 + (y_a/y_h)\phi_{ha} + (y_s/y_h)\phi_{hs}} \end{aligned} \quad (A-1)$$

where, for example, ϕ_{ij} is given by:

$$\phi_{ij} = \frac{\left(1 + \left(\mu_i/\mu_j\right)^{1/2} \left(M_j/M_i\right)^{1/4}\right)^2}{2\sqrt{2} \left(1 + M_i/M_j\right)^{1/2}} \quad (A-2)$$

The viscosities of the individual components were taken from Ref. 3 which gives steam, air and hydrogen viscosity as a function of temperature at one atmosphere pressure. Corrections for pressure variation were not used as the corrections are very small for the range in pressures considered in the study. An interpolation routine was used to determine individual component viscosities for a given gas temperature.

2. Gas Thermal Conductivity

Reference 4 also presents the method of Lindsay and Bromley for obtaining the thermal conductivity of the gas mixture. The mixture conductivity is a function of the conductivities of the individual components and their mole fractions and is given by:

$$k_m = \frac{k_s}{1 + (y_a/y_s)A_{sa} + (y_h/y_s)A_{sh}} + \frac{k_a}{1 + (y_s/y_a)A_{as} + (y_h/y_a)A_{ah}} + \frac{k_h}{1 + (y_a/y_h)A_{ha} + (y_s/y_h)A_{hs}} \quad (A-3)$$

where the weighting factor A_{ij} is given by:

$$A_{ij} = \frac{1}{4} \left\{ 1 + \left[\frac{\mu_i}{\mu_j} \left(\frac{M_j}{M_i} \right)^{3/4} \left(\frac{1 + S_i/T_g}{1 + S_j/T_g} \right) \right]^{1/2} \right\}^2 \left(\frac{1 + S_{ij}/T_g}{1 + S_i/T_g} \right) \quad (A-4)$$

and the Sutherland constants are approximated by $S_i = 1.5 T_{bi}$, where T_{bi} is the normal boiling point of component i in degrees Kelvin, and S_{ij} is given by:

$$S_{ij} = (S_i S_j)^{1/2} \quad (\text{non polar components})$$

$$S_{ij} = 0.735 (S_i S_j)^{1/2} \quad (\text{one very polar component-water}) \quad (A-5)$$

Thermal conductivities of the individual components were taken from Ref. 3 and are given there as a function of temperature for one atmosphere pressure. Again no correction for pressure is necessary in the range of pressures of interest. The interpolation routine was used to determine an individual component conductivity for a given gas temperature.

3. Gas Diffusion Coefficient

Reference 4 outlines a method for computing the diffusion coefficient of a gas through two or more other gases. The diffusion coefficient for steam through hydrogen and air is given approximately by:

$$D = \frac{1 - y_s}{y_a/D_{sa} + y_h/D_{sh}} \quad (A-6)$$

For the binary diffusion coefficients D_{sa} and D_{sh} , Ref. 4 recommends the relationship:

$$D_{ij} = \frac{0.001858 T_g^{3/2} [(M_i + M_j)/M_i M_j]^{1/2}}{P_{ij} \sigma_{ij} \Omega_D} \quad (A-7)$$

where T_g is in degrees Kelvin and P_{ij} in atmospheres. The force constant is defined as $\sigma_{ij} = 1/2(\sigma_i + \sigma_j)$ where $\sigma_s = 2.649$, $\sigma_a = 3.617$ and $\sigma_h = 2.968$. The collision integral Ω_D is tabulated in Ref. 4, Table 8-1, as a function of the parameter KT_g/ϵ . For the mixture of components i and j :

$$\frac{KT_g}{\epsilon_{ij}} = \frac{KT_g}{\sqrt{\epsilon_i \epsilon_j}} \quad (A-8)$$

where $(K/\epsilon)_s = 356$, $(K/\epsilon)_a = 97$ and $(K/\epsilon)_h = 33.3$.

4. Gas Density

The density of the gas mixture is the sum of the densities of the individual components. For steam, the free stream steam always being saturated, a table of saturation density as a function of saturation temperature was used and interpolation was employed to find ρ_s at a given gas stream temperature. The ideal gas law was used to determine ρ_a and ρ_h :

$$\rho_i = \frac{P_i}{R_i T_g} \quad (A-9)$$

5. Gas Specific Heat

The specific heat of a gas mixture is given by:

$$C_{pm} = \sum_{i=1}^n C_{pi} X_i \quad (A-10)$$

where the specific heat of a component C_{p_i} was taken from Ref. 3. These data are also tabulated as a function of temperature and interpolation was used to find the correct value at a given gas stream temperature.

6. Water Properties

The thermodynamic properties of water are tabulated in Ref. 3 and 7 as a function of temperature. The tables for density and latent heat (Ref. 7) and viscosity and thermal conductivity (Ref. 3) were employed to determine the properties for both the cooling water and the condensate film. In Eq. (31) we require k_w , ρ_w , μ_w and C_w to determine cooling water film coefficient. Also, ρ_w and μ_w were used in Eq. (35) to determine ΔP_w as well as in Section C to determine condensate film properties.

7. Gas Stream Mole and Mass Fractions

The mass fraction for each component is gained through component densities:

$$x_i = \frac{P_i}{P_i + P_j + P_k + \dots} \quad (A-11)$$

where the component densities are defined in Section 4 of this appendix.

Since saturated steam departs appreciably from the ideal gas state one cannot say that the mole fraction of any component is the ratio of its partial pressure to total pressure. The mole fractions must be gained from densities and molecular weight. For any component, its mol volume is the ratio of density to molecular weight. The mole fraction is then the ratio of its mol volume to the sum of mol volumes of all components. That is:

$$y_i = \frac{P_i/M_i}{P_i/M_i + P_j/M_j + \dots} \quad (A-12)$$

Again, the density of any component is as defined in Section 4 of this appendix.

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APPENDIX B

COMPUTER PROGRAM

This appendix contains a listing of the digital computer program and its nomenclature. The variable names are arranged such that the first part of the name refers to the variable in question, and the latter part tells where it applies. For example, the variable MDOTSIN can be interpreted as MDOT-mass flow rate, -S-steam, -IN entering a tube row.

Nomenclature

A	Area; -T, cross section of a tube; -MIN, minimum gas flow area; -TUBEO, outside of a single tube; -FACE, face of heat exchanger.
A	Weighting factor A_{ij} in the correlation for gas mixture thermal conductivity; -SA, steam air; SH, steam hydrogen; AS, air steam, AH, air hydrogen; HS, hydrogen steam; HA, hydrogen air.
B2	The heat and mass transfer correlation coefficient b^* .
CP	Specific heat at constant pressure; -A, air; H, hydrogen; M, gas mixture; S, steam; W, water.
COEFHW	The numerical coefficient for the correlation of coolant heat transfer coefficient with coolant Reynolds and Prandtl numbers.
DIFCO	Diffusion coefficient; -SA of steam through air, -SH of steam through hydrogen, -M of steam through air and hydrogen.
DP	Pressure drop; -A of the gas across a row, -W of water across a tube.
DT	Tube diameter; -I inside, -O outside.

DT Temperature difference; -DIFF between free stream and condensate film, -FLOW between gas inlet to a row and exit.

EPSOK The ratio ϵ/K ; -A, air; -H, hydrogen; -S, steam; -AS, air and steam; -AH, air and hydrogen; -HS, hydrogen and steam.

EXN The exponent n in the correlation of heat transfer coefficient with Reynolds number.

FF Friction factor; -A, gas mixture; -W, cooling water.

GASCON Gas constant; -A, air; -H, hydrogen; AH for the mixture of air and hydrogen.

H Heat or mass transfer coefficient; -CONV, forced convection between gas and condensate film; -DIFF, mass transfer between gas and condensate film; -FILM, across the condensate film; -WI, cooling water on the inside of a tube.

HEIGHT Height exchanger height.

HFCOEF The numerical factor J_1 which is a row-dependent coefficient for heat transfer across the film.

HP Ideal fan horsepower.

LENGTH Heat exchanger tube length.

LHV Latent heat of vaporization of steam.

MASSF Mass fraction; -A, air; -H, hydrogen; -S, steam.

MDOT Mass flow rate; -A, air; -AH, air and hydrogen entering the exchanger; -C of steam to be condensed on a tube, -GO of air and hydrogen leaving a tube row; -H of hydrogen; -S of steam; -SIN of steam entering the exchanger; -SO of steam leaving a tube row; -W of water.

MOLE/FT3 Moles/ft³; -A of air; -H, hydrogen; -M, mixture of gases; -S, steam.

MOLEF Mole fraction; -A, air; -H, hydrogen; -S, steam.

MOLEW Molecular weight; -A, air; -H, hydrogen; -S, steam.

MU Dynamic viscosity; -A, air; -H, hydrogen; -M, gas mixture; -S, steam; -W, cooling water; -WF, condensate film

NT Number of tubes; -OT, total for this row and above; -TOT, total in exchanger; -R in this row.

OMEGA The collision integral Ω_D ; -SA, steam and air; -SH, steam and hydrogen.

P Pressure; -COR, pressure correction factor for convection in gas flow ($\bar{P}$); -TOT, total gas.

PHI Weighting factor ϕ in the correlation for gas mixture viscosity; -AH, air hydrogen; -AS, air steam; HA, hydrogen air; -HS, hydrogen steam; -SA, steam air; -SH, steam hydrogen.

PP Partial pressure; -A, air; -AHF, air and hydrogen at condensate film; -AHG of air and steam in the free stream; -AO, air leaving a row; -G ϕ of air and hydrogen leaving a tube row; -H, hydrogen; -HO, hydrogen leaving a row; -S, steam; -SF, steam at the condensate film surface; -SO, steam leaving a tube row.

PRA Prandtl number; -W of water.

QCFM Volumetric flow rate of gas; -IN into a tube row; -O out of a tube row.

Q Heat transfer rate; -PT for a tube from gas to film; -PTP for a single tube across film; -PR for one row of the exchanger; -TOT for the entire exchanger.

QW Volumetric flow rate of cooling water; -AT for entire exchanger; -PT for a single tube.

REY Reynolds number; -W of cooling water; -F of condensate film leaving last tube row; -C of gas based upon the dimension $(S_T - D_{to})$.

RHO Density; -A, air; -F of condensate film; -GO of air and hydrogen leaving a row; -H, hydrogen; -M of gas mixture; -S, steam; -SF of steam at the condensate film surface; -SO of steam leaving a tube row.

ROW Row number.

ROWMOD The numerical coefficient for heat and mass transfer from gas to film, F_i .

S Sutherland constant; -A, air; -AH, air and hydrogen; -H, hydrogen; -S, steam; -SA, steam and air; -SH, steam and hydrogen.

SIG The force constant σ in the diffusion coefficient correlation; -A, air; -AH, air and hydrogen; -H, hydrogen; -S, steam; -SA, steam and air; -SH, steam and hydrogen.

S Tube pitch; -L, longitudinal; -T, transverse.

SCH Sherwood number of gas mixture.

T Temperature; -BOILA, normal boiling of air; -BOILH, normal boiling of hydrogen; -BOILS, normal boiling of steam; -FAVG, average condensate film; -FILM, outer surface of condensate film; -GAS, gas entering a tube row; -IN, gas entering a tube row; -OUT, gas leaving a tube row; -SURF, outer surface of tube; -WAVG, average water in a tube; -WIN, water inlet; -WOUT, water exit.

TK Thermal conductivity; -A, air; -F, condensate film; -H, hydrogen; -M of gas mixture; -S, steam; -TUBE of tube material.

U Thermal conductance; $-T\theta$ between gas and cooling water; $-W$ between tube outer surface and cooling water.

VMAX Maximum gas velocity in a tube row.

WH Length of header.

WIDTH Width of exchanger.

X Tube pitch ratio; $-L$, longitudinal; $-T$, transverse.

LISTING

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BEGIN
FILE IN READER(2,10);
FILE OUT PRINTER 4(2,15);
$$ A INTERPOLATE
$$ A DBLINTERPOLATE
$$ A TIMEIT
REAL ARRAY TKST,CPST,MUST,STT[0:8];
REAL ARRAY PST,RHOST,DPSTT[0:16];
REAL ARRAY AHTT,CPAT,MUAT,TKAT,CPHT,MUHT,TKHT[0:7];
REAL ARRAY WTT,RHOWT,TKWT,MUWT,LHVWT[0:11];
REAL ARRAY R2T,EXNT[0:4,0:4];
REAL ARRAY XLT,XTT[0:4];
REAL ARRAY MFRWT,NROWT[0:12];
REAL ARRAY KTUEPST,OMEGAT[0:81];
REAL ARRAY HFCDEF[0:40];
REAL ARRAY NTR[0:40];
REAL AAH,AAS,AHA,AHS,AMIN,ARG,ASA,ASH,AT,ATUBED,B2,CDEFHW,CPA,CPH,CPM,
CPS,CPW,DEL,DIFCON,DIFCOSA,DIFCOSH,DTDIFF,DTFLOW,DTI,DTO,EPSOKA,
EPSOKAH,EPSOKAS,EPSOKH,EPSOKHS,EPSOKS,EXN,FFI,FFO,GASCONA,GASCONAH,
GASCONH,HCONV,HDIFF,HEIGHT,HFILM,HWI,LENGTH,L,LHV,LHVF,MASSFA,
MASSFH,MASSFS,MDOTA,MDOGTG,MDOTC,MDOTH,MDOOTS,MDOOTSQ,MDOTW,MOLEFA,
MOLEFH,MOLEFS,MOLEAFT3,MOLEHFT3,MOLSF3,MOLEMFT3,MOLEWA,MOLEWH,
MOLEWS,MUA,MUH,MUM,MUS,MUW,MUWF,NT,OMEGASA,OMEGASH,PCOR,PHIAH,PHIAS,
PHIHA,PHIHS,PHISA,PHISH,PI,PPA,PPAHF,PPAHG,PPGO,PPH,PPS,PPSQ,PPSF,
PRA,PAW,PTOT,QCFM,QCFMO,QPR,QPT,QPTP,QTOT,REY,REYW,RHQA,RHOF,RHOGN,
RHOH,RHOM,RHOS,RHOSF,RHOSO,RHOW,ROWMOD,SA,SAH,SH,SIGA,SIGAH,SIGH,
SIGS,SIGSA,SIGSH,SL,SCH,SS,SSA,SSH,ST,TBOILA,TBOILH,TBOILS,TFILM,
TGAS,TIN,TGASK,TKA,TKF,TKH,TKM,TKS,TKTUBE,TKW,TOLMDOOT,TOLQPT,TOUT,
TWAVG,TWIN,TSURF,UW,VFACE,VMAX,WIDTH,XL,XT,Z,ZA,AFACE;
INTEGER N,NITDIF,NITFLO,NROWS,NTTOT,ROW;
LABEL L1,L2,L3,L4;
BOOLEAN LL,UL;
REAL QW,QWPT,FW,DPW,TWOUT,DPA,PPR,MDOTAH,RHOGIN,TGIN;
BOOLEAN LLA,ULA;
BOOLEAN CP2,CP3,CP4,CP5,CP6,CP7;
REAL ARRAY XDTO,XDTI,XXL,XXT,XVFACE,XQW[0:10];
INTEGER I,N1,N2,N3,N4,N5,NDT,NXL,NXT,NVFACE,NQW;
REAL HP,QCFMIN,EROW,MDOGIN,REYF,TFAVG;
REAL FFA,REYC,PPAO,PPHO,QWAT;
REAL UTOT;

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REAL UTDIFFZ,UTFLOWZ,LENGTHZ,PPSZ,PPAZ,PPHZ,TINZ,WIDTHZ;
LABEL L5,L6;
LABEL L7;
REAL NTOT;
REAL WH;
COMMENT THE INPUT VARIABLES HAVE THE FOLLOWING DIMENSIONS;
CPW: BTU/LBM-F.
DTDIFF: F.
DTFLOW: F.
FFI: HR-FT2-F/BTU.
FFO: HR-FT2-F/BTU.
HEIGHT,LENGTH,WIDTH: INCHES
PPS,PPA,PPH: POUNDS/IN2.
TIN: F.
TKTUBE: BTU/HR-FT-F.
TOLMDUT: NONE.
TOLQPT: NONE.
TWIN: F.
DIO,DTI: INCHES.
VFACE: FT/MIN.
XL,XT: NONE.
QW: GAL/MIN.

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;
COMMENT THE OUTPUT VARIABLES HAVE THE FOLLOWING DIMENSIONS:

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TIN,TOUT: F.
PPA,PPAO,PPH,PPHO,PPS,PPSO: POUNDS/IN2.
QCFM,QCFMO,QCFMIN: FT3/MIN.
MDOTS,MDOTSO,MDOA,MDOH: LBM/MIN.
QPT,QPR: BTU/MIN.
QTOT: BTU/SEC.
QW,QWAT: GAL/MIN.
DPA,DPW: POUNDS/IN2.
HP: HORSEPOWER.
EROW: NONE.
REYF: NONE.
TWOUT: F.

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FORMAT FMT10(/,"FINAL RESULTS FOR THE OVERALL EXCHANGER",//,"QTOT =",
E15.8,X1,"TGIN =",E15.8,X1,"TOUT =",E15.8,X1,"PPAO =",E15.8,X1,"PPSO =",
E15.8,X1,"PPHO =",E15.8,X1,"DPW =",E15.8,X1,"DPA =",E15.8,X1,"HP =",
E15.8,X1,"QCFM =",E15.8,X1,"QWATR =",E15.8,X1,"REYF =",E15.8,X1,"TWOUT =",
E15.8,///);
FORMAT FMT11(/,"THE NUMBER OF TUBES IN EACH ROW",//,X1,"ROW",X3,"NTR");
FORMAT FMT12(I4,X2,I4);
FORMAT FMT13(/,"REYNOLDS NUMBER TOO LARGE FOR PRESSURE DROP CORRELATION"
,///);
FORMAT FMT14(/,"REYNOLDS NUMBER TOO SMALL FOR PRESSURE DROP CORRELATION"
,///);
FILL TKST[*] WITH 0,.0120,.0141,.0142,.0151,.0173,.0196,.0219,.0244;
FILL CPST[*] WITH 0,.500,.493,.492,.481,.473,.474,.477,.484;
FILL MUST[*] WITH 0,.00e-06,7.50e-06,8.54e-06,9.03e-06,10.25e-06,
11.45e-06,12.06e-06,13.89e-06;
FILL STT[*] WITH 0,0,212,224,260,350,440,530,620;
FILL DPST[*] WITH 0,100,120,140,160,180,200,220,240,260,280,300,320,
340,360,380,400;
FILL PST[*] WITH 0,.9492,1.6924,2.8886,4.741,7.510,11.526,17.186,24.969,
35.429,49.203,67.013,89.66,118.01,153.04,195.77,247.31;
FILL RHST[*] WITH 0,.00285,.00492,.00813,.01294,.01991,.02973,.04320,
.06126,.08501,.11567,.15466,.20350,.26399,.33818,.42827,.53668;
FILL AHST[*] WITH 0,80,170,260,350,440,530,620;
FILL CPAT[*] WITH 0,.2402,.2410,.2422,.2438,.2459,.2482,.2520;
FILL MUAT[*] WITH 0,1.241e-05,1.394e-05,1.536e-05,1.669e-05,1.795e-05,
1.914e-05,2.028e-05;
FILL TKAT[*] WITH 0,.01516,.01735,.01944,.02142,.02333,.02519,.02692;
FILL CPHT[*] WITH 0,3.419,3.448,3.461,3.463,3.465,3.471,3.472;
FILL MUHT[*] WITH 0,5.023e-06,6.689e-06,7.300e-06,7.915e-06,8.491e-06,
9.055e-06,9.599e-06;
FILL TKHT[*] WITH 0,.105,.119,.132,.145,.157,.169,.182;
FILL WTT[*] WITH 0,32,68,104,140,176,212,248,284,320,356,392;
FILL RHWT[*] WITH 0,62.57,62.46,62.09,61.52,60.81,59.97,59.01,57.95,
56.79,55.50,54.11;
FILL TKWT[*] WITH 0,.319,.345,.363,.376,.386,.393,.396,.395,.393,.390,
.384;
FILL MUWT[*] WITH 0,1.204e-03,.6764e-03,.4396e-03,.3162e-03,.2384e-03,
.1895e-03,.1570e-03,.1333e-03,.1159e-03,.1032e-03,.09307e-03;
FILL LHVWT[*] WITH 0,1075.8,1055.5,1034.9,1014.1,992.7,970.3,946.8,
921.8,894.9,865.6,833.6;
FILL B2T[0,*] WITH 0,0,0,0,0;
FILL B2T[1,*] WITH 0,.578,.564,.579,.583;
FILL B2T[2,*] WITH 0,.503,.513,.504,.545;
FILL B2T[3,*] WITH 0,.451,.464,.538,.501;
FILL B2T[4,*] WITH 0,.346,.397,.491,.470;

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FILL EXNT[0,*] WITH 0,0,0,0,0;
FILL EXNT[1,*] WITH 0,.556,.554,.556,.562;
FILL EXNT[2,*] WITH 0,.568,.562,.568,.568;
FILL EXNT[3,*] WITH 0,.572,.568,.556,.570;
FILL EXNT[4,*] WITH 0,.592,.580,.562,.574;
FILL XLT[*] WITH 0,1.25,1.50,2.00,3.00;
FILL XTT[*] WITH 0,1.25,1.50,2.00,3.00;
FILL MFROWT[*] WITH 0,.68,.82,.99,1.07,1.04,1.10,1.09,1.05,1.07,1.09,
1.06,1.08;
FILL NROWT[*] WITH 0,1,2,3,4,5,6,7,8,9,10,12,15;
FILL HFCDEF[*] WITH 0,.7276,.7276,.4940,.4940,.4331,.4331,.3978,.3978,
.3734,.3734,.3551,.3551,.3405,.3405,.3285,.3285,.3184,.3184,.3096,.3096,
.3020,.3020,.2952,.2952,.2891,.2891,.2836,.2836,.2785,.2785,.2739,.2739,
.2697,.2697,.2657,.2657,.2621,.2621,.2587,.2587;
FILL KTOEPST[*] WITH 0,.30,.35,.40,.45,.50,.55,.60,.65,.70,.75,.80,.85,
.90,.95,1.00,1.05,1.10,1.15,1.20,1.25,1.30,1.35,1.40,1.45,1.50,1.55,1.60,
1.65,1.70,1.75,1.80,1.85,1.90,1.95,2.00,2.1,2.2,2.3,2.4,2.5,2.6,2.7,2.8,
2.9,3.0,3.1,3.2,3.3,3.4,3.5,3.6,3.7,3.8,3.9,4.0,4.1,4.2,4.3,4.4,4.5,4.6,
4.7,4.8,4.9,5.0,6,7,8,9,10,20,30,40,50,60,70,80,90,100,200,400;
FILL OMEGAT[*] WITH 0,2.662,2.476,2.318,2.184,2.066,1.966,1.877,1.798,
1.729,1.667,1.612,1.562,1.517,1.476,1.439,1.406,1.375,1.346,1.320,1.296,
1.273,1.253,1.233,1.215,1.198,1.182,1.167,1.153,1.140,1.128,1.116,1.105,
1.094,1.084,1.075,1.057,1.041,1.026,1.012,.9996,.9878,.9770,.9672,.9576,
.9490,.9406,.9328,.9256,.9186,.9120,.9058,.8998,.8942,.8888,.8836,.8788,
.8740,.8694,.8652,.8610,.8568,.8530,.8492,.8456,.8422,.8124,.7896,.7712,
.7556,.7424,.6640,.6232,.5960,.5756,.5596,.5464,.5352,.5256,.5130,.4644,
.4170;
WRITE(PRINTER,FMT1);
COMMENT COMPUTE PRELIMINARY VARIABLES;
TBOILS = 373;
TBOILA = 78.5;
TBOILH = 20.39;
SS = TBOILS*1.5;
SA = TBOILA*1.5;
SH = TBOILH*1.5;
SSA = .735*SQRT(SS*SA);
SSH = .735*SQRT(SS*SH);
SAH = SQRT(SA*SH);
MOLEWS = 18.02;
MOLFWA = 28.96;
MOLFWH = 2.016;
SIGS = 2.649;
SIGA = 3.617;
SIGH = 2.968;
SIGSA = (SIGS+SIGA)/2;

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SIGSH + (SIGS + SIGH) / 2;
SIGAH + (SIGA + SIGH) / 2;
EPSOKA + 97;
EPSOKS + 356;
EPSOKH + 33.3;
EPSOKAS + SQRT(EPSOKA * EPSOKS);
EPSOKAH + SQRT(EPSOKA * EPSOKH);
EPSOKHS + SQRT(EPSOKH * EPSOKS);
PI + 3.1415926535;
CDEFHW + .023;
GASCONA + 1545/MOLEWA;
GASCONH + 1545/MOLEWH;
READ(READER,/,CP2,CP3,CP4,CP5,CP6,CP7);
IF CP7 THEN WRITE(PRINTER,FMT2,SS,SA,SH,SSA,SSH,SAH,SIGSA,SIGSH,SIGAH,
EPSOKAS,EPSOKAH,EPSOKHS,GASCONA,GASCONH);
IF CP7 THEN TIMEIT(PRINTER);
TIMEIT(PRINTER);
READ(READER,/,CPW,DTDIFF,DTFLOW,FFI,FFO,HEIGHT,LENGTH,PPS,PPA,PPH,TIN,
TKTUBE,TCLMDOI,TOLQPT,TWIN,WIDTH);
READ(READER,/,NITDIF,NITFLO,NROWS);
DTDIFFZ + DTDIFF;
DTFLOWZ + DTFLOW;
LENGTHZ + LENGTH;
PPSZ + PPS;
PPAZ + PPA;
PPHZ + PPH;
TINZ + TIN;
WIDTHZ + WIDTH;
READ(READER,/,NDT,FOR I + 1 STEP 1 UNTIL NOT DO [XDTO[I],XDTI[I]]);
READ(READER,/,NXL,FOR I + 1 STEP 1 UNTIL NXL DO XXL[I]);
READ(READER,/,NXT,FOR I + 1 STEP 1 UNTIL NXT DO XXT[I]);
READ(READER,/,NVFACE,FOR I + 1 STEP 1 UNTIL NVFACE DO XVFACE[I]);
READ(READER,/,NQW,FOR I + 1 STEP 1 UNTIL NQW DO XQW[I]);
FOR N1 + 1 STEP 1 UNTIL NOT DO BEGIN
DTO + XDTO[N1];
DTI + XDTI[N1];
FOR N2 + 1 STEP 1 UNTIL NXL DO BEGIN
XL + XXL[N2];
FOR N3 + 1 STEP 1 UNTIL NXT DO BEGIN
XT + XXT[N3];
FOR N4 + 1 STEP 1 UNTIL NVFACE DO BEGIN
VFACE + XVFACE[N4];
FOR N5 + 1 STEP 1 UNTIL NQW DO BEGIN
QW + XQW[N5];
COMMENT COMPUTE SYSTEM VARIABLES FROM INPUT VALUES:

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DTDIFF ← DTDIFZ;
DTFLOW ← DTFLWZ;
LENGTH ← LENGTHZ;
PPS ← PPSZ;
PPA ← PPAZ;
PPH ← PPHZ;
TIN ← TINZ;
WIDTH ← WIDTHZ;
SL ← XL×DTO;
ST ← XT×DTO;
NT ← (WIDTH-DTO)/ST + 1;
NT ← ENTIER(NT); NTR[1] ← NT;
WIDTH ← (NT-1)×ST + DTO;
HEIGHT ← SL×(NROWS-1) + DTO;
NTTOT ← NTR[1];
FOR N ← 2 STEP 1 UNTIL NROWS DO BEGIN
  NTR[N] ← NTR[1] - ((N+1)/2-ENTIER((N+1)/2))×2;
  NTTOT ← NTTOT + NTR[N]; END;
AT ← PI×DTI×DTI/(4×144);
WH ← AT×144×NTTOT/(6×HEIGHT) + .5;
LENGTH ← LENGTH - 2×WH;
AMJN ← (WIDTH - NT×DTO)×LENGTH/144;
ATURFO ← PI×DTO×LENGTH/144;
MOLEFA ← PPA/(PPA+PPH);
MOLEFH ← 1-MOLEFA;
GASCONAH ← 1/(MOLEFA/GASCONA + MOLEFH/GASCONH);
L ← LENGTH;
AFACE ← LENGTH×WIDTH/144;
QCFM ← VFACE×AFACE;
QWPT ← QW/NTTOT;
RHOW ← INTERPOLATE(WTT,RHOWT,TWIN,11,LL,UL);
MDOTW ← RHOW×QWPT/7.48;
B2 ← DBLINTERPOLATE(XLT,XTT,R2T,XL,XT,4,4,LL,UL,LLA,ULA);
EXN ← DBLINTERPOLATE(XLT,XTT,EXNT,XL,XT,4,4,LL,UL,LLA,ULA);
TGIN ← TIN;
RHOS ← INTERPOLATE(DPSTT,RHOST,TIN,16,LL,UL);
RHDA ← PPA×144/(GASCONA×(TIN+460));
RHOH ← PPH×144/(GASCONH×(TIN+460));
RHGIN ← RHOS+RHDA+RHOH;
QCFMIN ← QCFM;
MDQSIN ← RHOS×QCFM;
MUW ← INTERPOLATE(WTT,MUWT,TWIN,11,LL,UL);
REYW ← MDUTW×DTI/(12×AT×MUW×60);
FFW ← .0014 + .125/((REYW)×.32);
ARG ← MDOTW×MDOTW/(3600×AT×AT×RHOW);

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DPW + (FFW*LENGTH*4/DTI + 1.5)*ARG/(32.2*2*144);
PPR + PPA/PPH;
MDQTAH + (RHOA+RHOH)*QCFM;
WRITE(PRINTER,FMT3,CPW,DTD(FF,DTFLOW,DTI,DTO,FFI,FFO,HEIGHT,LENGTH,PPS,
PPA,PPH,TIN,TKTUBE,TOLMOT,TOLQPT,TWIN,VFACE,WIDTH,XL,XT,QW));
WRITE(PRINTER,FMT4,NITDIF,NITFLO,NROWS);
IF CP6 THEN WRITE(PRINTER,FMT5,SL,ST,NT,WIDTH,HEIGHT,AMIN,ATUBFO,AT,
MOLEFA,MOLEFH,GASCONAH,AFACE,QCFM,QWPT,MOTW,B2,EXN,RHOS,RHOA,RHOH,
RHOGIN,REYW,FFW,DPW,PPR,MDQTAH);
IF CP6 THEN TIMEIT(PRINTER);
WRITE(PRINTER,FMT11);
FOR N + 1 STEP 1 UNTIL NROWS DO WRITE(PRINTER,FMT12, N ,NTR[N]);
COMMENT START OF ROW-BY-ROW CALCULATIONS;
QTOT + 0; TWOUT + 0;
QWAT + 0;
DPA + 0;
NTOT + 0;
FOR ROW + 1 STEP 1 UNTIL NROWS DO BEGIN
COMMENT START OF DETERMINING GAS PROPERTIES FOR THE GIVEN ROW;
BEGIN REAL BREAKIT;
NTOT + NTOT + NTR[ROW];
EROW + ROW;
TGAS + TIN; TGASK + 5/9*(TGAS-32)+273;
RHOS + INTERPOLATE(DPSTT,RHOST,TGAS,16,LL,UL);
RHOH + PPH*144/(GASCONH*(TGAS+460));
RHOA + PPA*144/(GASCONA*(TGAS+460));
RHOM + RHOS+RHOH+RHOA;
MUS + INTERPOLATE(STT,MUST,TGAS,8,LL,UL);
MUH + INTERPOLATE(AHTT,MUHT,TGAS,7,LL,UL);
MUA + INTERPOLATE(AHTT,MUAT,TGAS,7,LL,UL);
MASSFH + RHOS/RHOM;
MASSFA + RHOA/RHOM;
MASSFH + RHOH/RHOM;
PPS + INTERPOLATE(DPSTT,PST,TGAS,16,LL,UL);
PTOT + PPS+PPA+PPH;
MOLESFT3 + RHOS/MOLEWS;
MOLEAFT3 + RHOA/MOLEWA;
MOLEHFT3 + RHOH/MOLEWH;
MOLEMFT3 + MOLESFT3+MOLEAFT3+MOLEHFT3;
MOLFFS + MOLESFT3/MOLEMFT3;
MOLFFA + MOLEAFT3/MOLEMFT3;
MOLEFH + MOLEHFT3/MOLEMFT3;
PHISA + ((1+((MUS/MUA)*.5)*((MOLEWA/MOLEWS)*.25))*2)/(2.828*((1+MOLEWS/
MOLEWA)*.5));
PHISH + ((1+((MUS/MUH)*.5)*((MOLEWH/MOLEWS)*.25))*2)/(2.828*((1+MOLEWS/

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MOLEWH)*.5));
 PHIAS + ((1+((MUA/MUS)*.5)*((MOLEWS/MOLEWA)*.25))*2)/(2.828*((1+MOLEWA/
 MOLEWS)*.5));
 PHIAH + ((1+((MUA/MUH)*.5)*((MOLEWH/MOLEWA)*.25))*2)/(2.828*((1+MOLEWA/
 MOLEWH)*.5));
 PHIHS + ((1+((MUH/MUS)*.5)*((MOLEWS/MOLEWH)*.25))*2)/(2.828*((1+MOLEWH/
 MOLEWS)*.5));
 PHIHA + ((1+((MUH/MUA)*.5)*((MOLEWA/MOLEWH)*.25))*2)/(2.828*((1+MOLEWH/
 MOLEWA)*.5));
 MUM + MUS/(1+PHISA*(MOLEFA/MOLEFS)+PHISH*(MOLEFH/MOLEFS))+MUA/(1+PHIAS*(
 MOLEFS/MOLEFA)+PHIAH*(MOLEFH/MOLEFA))+MUH/(1+PHIHS*(MOLEFS/MOLEFH)+
 PHIHA*(MOLEFA/MOLEFH));
 TKS + INTERPOLATE(STT,TKST,TGAS,8,LL,UL);
 TKA + INTERPOLATE(AHTT,TKAT,TGAS,7,LL,UL);
 TKH + INTERPOLATE(AHTT,TKHT,TGAS,7,LL,UL);
 ASA + 1/4*((1+((MUS/MUA)*((MOLEWA/MOLEWS)*.75)*(1+SS/TGASK)/(1+SA/TGASK)
)*.5))*2*(1+SSA/TGASK)/(1+SS/TGASK);
 ASH + 1/4*((1+((MUS/MUH)*((MOLEWH/MOLEWS)*.75)*(1+SS/TGASK)/(1+SH/TGASK)
)*.5))*2*(1+SSH/TGASK)/(1+SS/TGASK);
 AAS + 1/4*((1+((MUA/MUS)*((MOLEWS/MOLEWA)*.75)*(1+SA/TGASK)/(1+SS/TGASK)
)*.5))*2*(1+SSA/TGASK)/(1+SA/TGASK);
 AAH + 1/4*((1+((MUA/MUH)*((MOLEWH/MOLEWA)*.75)*(1+SA/TGASK)/(1+SH/TGASK)
)*.5))*2*(1+SAH/TGASK)/(1+SA/TGASK);
 AHS + 1/4*((1+((MUH/MUS)*((MOLEWS/MOLEWH)*.75)*(1+SH/TGASK)/(1+SS/TGASK)
)*.5))*2*(1+SSH/TGASK)/(1+SH/TGASK);
 AHA + 1/4*((1+((MUH/MUA)*((MOLEWA/MOLEWH)*.75)*(1+SH/TGASK)/(1+SA/TGASK)
)*.5))*2*(1+SAH/TGASK)/(1+SH/TGASK);
 TKM + TKS/(1+ASA*(MOLEFA/MOLEFS)+ASH*(MOLEFH/MOLEFS))+TKA/(1+AAS*(MOLEFS/
 MOLEFA)+AAH*(MOLEFH/MOLEFA))+TKH/(1+AHS*(MOLEFS/MOLEFH)+AHA*(MOLEFA/
 MOLEFH));
 VMAX + QCFM/(60*AMIN);
 MDOA + RHOA*QCFM;
 MDOH + RHOH*QCFM;
 MDOS + RHOS*QCFM;
 REY + VMAX*DTU*RHOM/(12*MUM);
 CPA + INTERPOLATE(AHTT,CPAT,TGAS,7,LL,UL);
 CPH + INTERPOLATE(AHTT,CPHT,TGAS,7,LL,UL);
 CPS + INTERPOLATE(STT,CPST,TGAS,8,LL,UL);
 CPM + MASSFS*CPS+MASSFAX*CPA+MASSFH*CPH;
 PRA + MUM*CPM/TKM*3600;
 ARG + TGASK/EPKOKAS;
 OMEGASA + INTERPOLATE(KTOEPST,OMEGAT,ARG,81,LL,UL);
 DIFCOSA + .001858*((TGASK)*1.5)*((MOLEWS+MOLEWA)/(MOLEWS*MOLEWA)*.5)/
 ((PPS+PPA)/14.696*SIGSA*SIGSA*OMEGASA);
 ARG + TGASK/EPKOKHS;

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OMEGASH + INTERPOLATE(KTOEPST,OMEGAT,ARG,81,LL,UL);
DIFCOSH + .001858*((TGASK)*1.5)*(((MOLEWS+MOLEWH)/(MOLEWS*MOLEWH))*1.5)/
((PPS+PPH)/14.696*SIGSH*SIGSH*OMEGASH);
DIFCOM + (1-MOLEFS)/(MOLEFA/DIFCOSA+MOLEFH/DIFCOSH);
DIFCOM + DIFCOM/929.03;
SCH + MUM/(RHOM*DIFCOM);
ARG + ROW;
ROWMOD + INTERPOLATE(MROWT,MFROMT,ARG,12,LL,UL);
HDIFF + DIFCOM*12/DT0*B2*ROWMOD*(SCH*.333)*(REY*EXN)*60;
HCONV + TKM*12/DT0*B2*ROWMOD*(PRA*.333)*(REY*EXN)/60;
IF CP5 THEN WRITE(PRINTER,FMT6,TGAS,RHOS,RHOA,RHOH,RHOM,MUS,MUH,MUA,
MASSF5,MASSF6,MASSFH,PPA,PPS,PPH,PTOT,MOLEFS,MOLEFA,MOLEFH,TKS,TKA,TKH,
TKM,MUM,VMAX,MDOTA,MDOTS,MDOTH,REY,CPS,CPA,CPH,CPM,PRA,DIFCOSA,DIFCOSH,
DIFCOM,SCH,ROWMOD,HDIFF,HCONV);
IF CP5 THEN TIMEIT(PRINTER);
END BREAKIT;
COMMENT START OF GAS-TO-FILM HEAT TRANSFER CALCULATIONS;
TFILM + TGAS - DTDIFF; DEL + DTDIFF/NITDIF; ZA + 0;
L1:
RHOSF + INTERPOLATE(DPSTT,RHOST,TFILM,16,LL,UL);
PPSF + INTERPOLATE(DPSTT,PST,TFILM,16,LL,UL);
PPAHF + PTOT-PPSF;
PPAHG + PPA+PPH;
PCOR + PTOT*LN(PPAHF/PPAHG)/(PPAHF-PPAHG);
MDUTC + HDIFF*PCOR*(RHOS-RHOSF);
LHV + INTERPOLATE(WTT,LHVWT,TFILM,11,LL,UL);
QPT + (LHV*MDUTC + HCONV*(TGAS-TFILM))*ATUBE0;
COMMENT START OF TUBE-TO-WATER HEAT TRANSFER CALCULATIONS;
TWAVG + TWIN + QPT/(2*MDOTW*CPW);
TKW + INTERPOLATE(WTT,TKWT,TWAVG,11,LL,UL);
RHQW + INTERPOLATE(WTT,RHOWT,TWAVG,11,LL,UL);
MUW + INTERPOLATE(WTT,MUWT,TWAVG,11,LL,UL);
REYW + MDOTW*DTI/(12*60*AT*MUW);
PRAW + MUW*CPW*3600/TKW;
HWI + TKW*12/DTI*COEFHW*(1+((DTI/L)*.7))*(REYW*.8)*(PRAW*.333);
UW + 1/(12/(PI*HWI*DTI)+LN(DTO/DTI)/(2*PI*TKTURE)+12*FF0/(PI*DTI)+FFI*12
/(PI*DTI));
TSURF + QPT/(MDOTW*CPW*(1-EXP(-UW*L/(60*MDOTW*CPW*12)))) + TWIN;
IF TSURF > TFILM THEN BEGIN
DTDIFF + UTDIFF/2; TFILM + TGAS - DTDIFF;
GO TO L1;
END;
COMMENT START OF HEAT TRANSFER ACROSS THE FILM;
TFAVG + (TFILM+TSURF)/2;
TKF + INTERPOLATE(WTT,TKWT,TFAVG,11,LL,UL);

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RHOF + INTERPOLATE(WTT,RHOWT,TF AVG,11,LL,UL);
LHVF + INTERPOLATE(WTT,LHVWT,TF AVG,11,LL,UL);
MUWF + INTERPOLATE(WTT,MUWT,TF AVG,11,LL,UL);
HFILM + HFCOE[ROW]*((TKF*TKF*TKF*RHOF*RHOF*LHVF*12*32.2*3600/(MUWF*DTN
*(TFILM-TSURF)))*.25);
QPTP + HFILM*ATUBEDX*(TFILM-TSURF)/60;
IF CP4 THEN WRITE(PRINTER,FMT7,TFILM,DEL,RHOSF,PPSF,PPAHF,PPAHG,PCOR,
MDOTC,LHV,QPT,TWAVG,REYW,PRAW,HWI,UW,TSURF,TF AVG,HFILM,QPTP,QPT);
IF CP4 THEN TIMEIT(PRINTER);
IF ABS((QPT-QPTP)/QPT) ≤ TOLQPT THEN GO TO L2;
Z + SIGN(QPT-QPTP);
IF Z - ZA ≠ 0 THEN DEL + DEL/2;
TFILM + TFILM + Z*DEL;
ZA + Z;
GO TO L1;
COMMENT START OF DETERMINING GAS STATE CHANGE ACROSS THIS ROW;
L2:
TOUT + TIN - DTFLOW; DEL + DTFLOW/NITFLO; ZA + 0;
L3:
RHOSD + INTERPOLATE(OPSTT,RHOST,TOUT,16,LL,UL);
PPSD + INTERPOLATE(OPSTT,PST,TOUT,16,LL,UL);
PPGD + PTOT - PPSD;
RHOGD + PPGD*144/(GASCONAH*(TOUT+460));
MDOTSD + MDOTS-MDOTC*ATUBEDX*NTR[ROW];
QCFMD + MDOTSD/RHOSD;
MDOTGD + RHOGD*QCFMD;
IF CP3 THEN WRITE(PRINTER,FMT8,TOUT,RHOSD,PPSD,PPGD,RHOGD,MDOTSD,QCFMD,
MDOTGD,MDOTAH);
IF CP3 THEN TIMEIT(PRINTER);
IF ABS((MDOTGD - MDOTAH)/MDOTGD) ≤ TOLMDOT THEN GO TO L4;
Z + SIGN(MDOTGD - MDOTAH);
IF Z - ZA ≠ 0 THEN DEL + DEL/2;
TOUT + TOUT + Z*DEL;
ZA + Z;
GO TO L3;
COMMENT SUMMATION OF EFFECTS OF THIS ROW;
L4:
QPR + QPT*NTR[ROW];
QTOT + QTOT + QPR/60;
QWAT + QWAT + QWPT*NTR[ROW];
TWOUT + TWOUT + (TWIN + QPT/(MDOTW*CPW))*NTR[ROW];
PPAD + PPR*PPGD/(1+PPR);
PPHD + PPGD - PPAD;
IF REY > 40000 THEN GO TO L5 ELSE IF REY < 5000 THEN GO TO L5 ELSE
FFA + (.25 + .1175/((ST/DTN-1)*1.08))/((REY)*.16);

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GO TO L6;
L5:
REYC = VMAX*(ST-DT0)*RHQM/(12*60*MUM);
IF REYC > 20000 THEN WRITE(PRINTER,FMT13) ELSE IF REYC<100 THEN WRITE
(PRINTER,FMT14) ELSE FFA = .75/((REYC)*.2);
L6:
DPA = DPA + 2*FFA*RHQM*VMAX*VMAX/(32.2*144);
DPA = DPA*(RHUGO+RHOSO)*QCFMO*(QCFM-QCFMO)/(AFACE*AFACF*32.2*144*3600);
HP = QCFMIN*DPA*144/33000;
DTDIFF = TGAS - TFILM;
DTFLOW = TIN - TOUT;
UTOT = HFILM*(TFILM-TSURF)/(TGAS-TWAVG);
IF CP2 THEN WRITE(PRINTER,FMT9,TIN,TOUT,PPA,PPAO,PPH,PPHO,PPS,PPSO,QCFM,
QCFMO,MDOIS,MDOISO,MDOIA,MDOIH,QPT,QPR,QTOT,QWAT,DPA,HP,EROW,HFILM,UTOT)
;
TIMEIT(PRINTER);
TIN = TOUT;
PPA = PPAO;
PPH = PPHU;
PPS = PPSO;
QCFM = QCFMO;
IF POW = NROWS THEN REYF = 2*12*(MDOIS-MDOISO)/(NTR[NROWS]*MUWFX60*L);
IF QTOT > 50000 THEN GO TO L7;
END;
COMMENT COMPUTE AND PRINT OVERALL RESULTS;
L7:
TWOUT = TWOUT/NTOT;
WRITE(PRINTER,FMT10,QTOT,TGIN,TOUT,PPAO,PPSO,PPHO,OPW,DPA,HP,QCFMIN,QW,
REYF,TWOUT);
TIMEIT(PRINTER);
WRITE(PRINTER,PAGE);
END; END; END; END; END;
END.

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APPENDIX C

TABULATION OF RESULTS OF PARAMETRIC STUDY

This appendix is a tabulation of the results of 75 cases where tube transverse and longitudinal pitch ratios, tube inside and outside diameters, gas flow rate and cooling water flow rate were varied to determine their effect on exchanger performance.

APPENDIX C TABLE OF COMPUTER RESULTS

Run No.	d _{to} Inches	d _{ti} Inches	Tube Length Inches	Exch Width Inches	Exch Height Inches	Transverse Pitch Ratio	Lateral Pitch Ratio	Total Tubes	Rows	Face Vel ft/min	Total Cooling Water GPM	Q _{total} Btu/sec	Water Exit Temp °F	ΔP Air psi	Ideal Fan Power HP	ΔP Water psi
1	1.000	.834	71.49	63.50	12.25	1.25	1.25	505	10	480	1500	13744	151.0	.342	22.60	.082
2	1.000	.834	72.83	62.50	12.25	1.50	1.25	415	10	480	1500	12638	145.7	.122	8.06	.119
3	1.000	.834	73.72	62.25	12.25	1.75	1.25	355	10	480	1500	11745	141.4	.026	1.74	.160
4	1.000	.834	72.66	63.5	14.50	1.25	1.50	505	10	480	1500	13951	152.0	.342	22.98	.083
5	1.000	.834	73.79	62.50	14.50	1.50	1.50	415	10	480	1500	12791	146.4	.122	8.17	.120
6	1.000	.834	74.54	62.25	14.50	1.75	1.50	355	10	480	1500	11866	142.0	.026	1.76	.161
7	1.000	.834	73.51	63.50	16.75	1.25	1.75	505	10	480	1500	14099	152.7	.342	23.25	.084
8	1.000	.834	74.49	62.50	16.75	1.50	1.75	415	10	480	1500	12903	147.0	.122	8.25	.121
9	1.000	.834	75.14	62.25	16.75	1.75	1.75	355	10	480	1500	11951	142.4	.026	1.78	.162
10	0.75	.584	72.44	63.56	8.25	1.25	1.25	608	9	480	1350	15328	166.7	.318	21.28	.242
11	0.75	.584	73.61	62.63	9.19	1.50	1.25	555	10	480	1500	15335	158.6	.049	3.26	.349
12	0.75	.584	74.38	62.44	9.19	1.75	1.25	475	10	480	1500	14133	152.9	.026	1.78	.467
13	0.75	.584	73.46	63.56	9.75	1.25	1.50	608	9	480	1350	15519	167.7	.318	21.59	.244
14	0.75	.584	74.44	62.63	10.88	1.50	1.50	555	10	480	1500	15490	159.4	.049	3.30	.352
15	0.75	.584	75.10	62.44	10.88	1.75	1.50	475	10	480	1500	14254	153.4	.026	1.80	.470
16	0.75	.584	74.20	63.56	11.25	1.25	1.75	608	9	480	1350	15651	168.4	.318	21.82	.246
17	0.75	.584	75.05	62.63	12.56	1.50	1.75	555	10	480	1500	15602	159.9	.049	3.33	.354
18	0.75	.584	75.62	62.44	12.56	1.75	1.75	475	10	480	1500	14337	153.8	.026	1.81	.472
19	0.50	.334	74.21	63.00	4.25	1.25	1.25	704	7	480	1050	15603	191.9	.185	12.59	1.53
20	0.50	.334	74.97	63.50	4.88	1.50	1.25	676	8	480	1200	16012	181.1	.042	2.87	2.11
21	0.50	.334	75.54	63.50	5.50	1.75	1.25	653	9	480	1350	16262	171.7	.025	1.73	2.80
22	0.50	.334	74.95	63.00	5.00	1.25	1.50	704	7	480	1050	15722	192.8	.185	12.73	1.55
23	0.50	.334	75.60	63.50	5.75	1.50	1.50	676	8	480	1200	16120	181.8	.042	2.90	2.13
24	0.50	.334	76.08	63.50	6.50	1.75	1.50	653	9	480	1350	16354	172.2	.025	1.74	2.82
25	0.50	.334	75.49	63.00	5.75	1.25	1.75	704	7	480	1050	15811	193.4	.185	12.82	1.56
26	0.50	.334	76.05	63.50	6.63	1.50	1.75	676	8	480	1200	16199	182.2	.042	2.92	2.14
27	0.50	.334	76.47	63.50	7.50	1.75	1.75	653	9	480	1350	16418	172.5	.025	1.75	2.83
28	0.75	.584	74.44	62.63	10.88	1.50	1.50	555	10	600	1000	12813	177.3	.189	15.99	.168
29	0.75	.584	74.44	62.63	10.88	1.50	1.50	555	10	600	1500	15518	159.5	.173	14.67	.352
30	0.75	.584	74.44	62.63	9.75	1.50	1.50	500	9	600	1800	15901	148.6	.154	13.02	.595
31	0.75	.584	74.44	62.63	8.63	1.50	1.50	449	8	600	2000	15562	141.0	.135	11.46	.895
32	0.75	.584	74.44	62.63	10.88	1.50	1.50	555	10	480	1000	12796	177.2	.051	3.43	.168
33	0.75	.584	74.44	62.63	10.88	1.50	1.50	555	10	480	1500	15490	159.4	.049	3.30	.352
34	0.75	.584	74.44	62.63	9.75	1.50	1.50	500	9	480	1800	15865	148.4	.044	2.97	.595
35	0.75	.584	74.44	62.63	8.63	1.50	1.50	449	8	480	2000	15525	140.9	.039	2.67	.895
36	0.75	.584	74.44	62.63	10.88	1.50	1.50	555	10	360	1000	12762	176.9	.028	1.43	.168
37	0.75	.584	74.44	62.63	10.88	1.50	1.50	555	10	360	1500	15436	159.1	.027	1.35	.352
38	0.75	.584	74.44	62.63	9.75	1.50	1.50	500	9	360	1800	15801	148.2	.024	1.22	.595
39	0.75	.584	74.44	62.63	8.63	1.50	1.50	449	8	360	2000	15450	140.6	.022	1.10	.895
40	0.75	.584	74.44	62.63	10.88	1.50	1.50	555	10	240	1000	12684	176.4	.012	0.40	.168
41	0.75	.584	74.44	62.63	10.88	1.50	1.50	555	10	240	1500	15295	158.4	.011	0.37	.352
42	0.75	.584	74.44	62.63	9.75	1.50	1.50	500	9	240	1800	15630	147.5	.010	0.33	.595
43	0.75	.584	74.44	62.63	8.63	1.50	1.50	449	8	240	2000	15273	140.0	.009	0.30	.895
44	0.50	.430	73.36	63.50	5.75	1.50	1.50	676	8	240	1200	15709	179.3	.009	0.32	.653
45	0.50	.430	73.36	63.50	5.00	1.50	1.50	592	7	240	1400	15717	165.8	.008	0.28	1.10
46	0.50	.430	74.16	63.50	6.50	1.75	1.50	653	9	240	1350	16034	170.5	.006	0.19	.868
47	0.50	.430	74.16	63.50	5.75	1.75	1.50	580	8	240	1600	16100	157.5	.005	0.18	1.46
48	0.50	.430	74.12	63.50	6.63	1.50	1.75	676	8	240	1200	15834	180.0	.009	0.32	.658
49	0.50	.430	74.12	63.50	5.75	1.50	1.75	592	7	240	1400	15843	166.4	.008	0.28	1.11
50	0.50	.430	74.81	63.50	7.50	1.75	1.75	653	9	240	1350	16139	171.0	.006	0.20	.873
51	0.50	.430	74.81	63.50	6.63	1.75	1.75	580	8	240	1600	16209	158.0	.005	0.18	1.47
52	0.50	.402	74.07	63.50	5.75	1.50	1.50	676	8	240	1200	15803	179.8	.009	0.32	.894
53	0.50	.402	74.07	63.50	5.00	1.50	1.50	592	7	240	1400	15755	166.0	.008	0.29	1.50
54	0.50	.402	74.77	63.50	6.50	1.75	1.50	653	9	240	1350	16082	170.7	.006	0.20	1.19
55	0.50	.402	74.77	63.50	6.63	1.75	1.50	580	8	240	1600	16095	157.5	.005	0.18	2.00
56	0.50	.402	74.73	63.50	6.63	1.50	1.75	676	8	240	1200	15908	180.5	.009	0.32	.900
57	0.50	.402	74.73	63.50	5.75	1.50	1.75	592	7	240	1400	15863	166.5	.008	0.29	1.51
58	0.50	.402	75.34	63.50	7.50	1.75	1.75	653	9	240	1350	16173	171.2	.006	0.20	1.19
59	0.50	.402	75.34	63.50	6.63	1.75	1.75	580	8	240	1600	16186	157.9	.005	0.18	2.01
60	0.50	.370	74.82	63.50	5.75	1.50	1.50	676	8	240	1200	15873	180.3	.009	0.32	1.32
61	0.50	.370	74.82	63.50	5.00	1.50	1.50	592	7	240	1400	15757	166.0	.008	0.29	2.21
62	0.50	.370	75.41	63.50	6.50	1.75	1.50	653	9	240	1350	16106	170.9	.006	0.20	1.75
63	0.50	.370	75.41	63.50	5.75	1.75	1.50	580	8	240	1600	16057	157.3	.005	0.18	2.94
64	0.50	.370	75.38	63.50	6.63	1.50	1.75	676	8	240	1200	15960	180.8	.009	0.33	1.33
65	0.50	.370	75.38	63.50	5.75	1.50	1.75	592	7	240	1400	15846	166.5	.008	0.29	2.23
66	0.50	.370	75.90	63.50	7.50	1.75	1.75	653	9	240	1350	16179	171.3	.006	0.20	1.76
67	0.50	.370	75.90	63.50	6.63	1.75	1.75	580	8	240	1600	16126	157.6	.005	0.18	2.95
68	0.50	.282	76.57	63.50	5.75	1.50	1.50	676	8	240	1200	15841	180.1	.009	0.34	4.68
69	0.50	.282	76.57	63.50	5.00	1.50	1.50	592	7	240	1400	15524	164.8	.009	0.30	7.86
70	0.50	.282	76.92	63.50	6.50	1.75	1.50	653	9	240	1350	15938	170.0	.006	0.20	6.19
71	0.50	.282	76.92	63.50	5.75	1.75	1.50	580	8	240	1600	15692	155.6	.005	0.19	10.40
72	0.50	.282	76.90	63.50	6.63	1.50	1.75	676	8	240	1200	15887	180.4	.009	0.34	4.70
73	0.50	.282	76.90	63.50	5.75	1.50	1.75	592	7	240	1400	15568	165.0	.009	0.30	7.89
74	0.50	.282	77.20	63.50	7.50	1.75	1.75	653	9	240	1350	15977	170.2	.006	0.21	6.21
75	0.50	.282	77.20	63.50	6.63	1.75	1.75	580	8	240	1600	15725	155.8	.005	0.19	10.43

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KEYWORDS - computers, condensation, flow, heat exchangers,
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An analysis is developed to predict heat transfer and
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coolant flow rate. Results show that an exchanger can be designed with a capacity which will exceed the desired heat transfer rate of 15000 Btu/sec in the prescribed geometric envelope of 6.5 by 5.3 by 2 feet.

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coolant flow rate. Results show that an exchanger can be designed with a capacity which will exceed the desired heat transfer rate of 15000 Btu/sec in the prescribed geometric envelope of 6.5 by 5.3 by 2 feet.

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