NASA Contractor Report 177643

A Computer Program for Condensing Heat Exchanger Performance in the Presence of Noncondensable Gases

Boris Yendler

The Bionetics Corporation
10th Floor, Suite 1000
Harbour Centre Building
2 Eaton Street
Hampton, VA 23669-4062

Prepared for
Ames Research Center
CONTRACT NAS2-13273
June 1994

National Aeronautics and
Space Administration

Ames Research Center
Moffett Field, California 94035-1000

Abstract

A computer model has been developed which evaluates the performance of a heat exchanger. This model is general enough to be used to evaluate many heat exchanger geometry and a number of different operating conditions. The film approach is used to describe condensation in the presence of noncondensables. The model is also easily expanded to include other effects like fog formation or suction.

Introduction

It would be convenient in many situations to have a tool that could evaluate heat exchanger performance based on physical phenomena and general correlation only. This tool should stand alone, and should be able to predict heat exchanger performance in the absence of experimental data. This tool is needed as some heat exchangers (such as used in humidity control systems for space applications) might have little or no experimental data. Without this tool, prediction of performance for an exchanger becomes impossible.

Existing commercial programs for calculating heat exchanger performance are usually too bulky. A source code of the program is not provided usually. Therefore, modifications of the code, which are often necessary to accommodate new effects, are next to impossible. This situation forced us to create a general program that can be customized easily for specific applications. In addition, the model can be included into simulation programs like Computer Aided System Engineering and Analysis (CASE/A) or MATLAB widely used by NASA Ames scientific community.

If there is no phase change taking place in a heat exchanger, the well-known Log-Mean Temperature Difference (LMTD) method is usually used to predict performance. For condensation of pure vapors, algebraic equations [1] are used to predict performance of the exchanger. For vapor condensation in the presence of noncondensable gases (air), a simple analytic solution is no longer possible. In order to describe energy transport in this situation, a reliable model must be developed. This model must be accurate and should be easily expanded to include other effects such as fog formation or suction at the wall between the hot and cold stream to remove condensate. The model should also be easily customized to accommodate many types of geometry and flow conditions.

Under some operational conditions condensation can start away from an inlet. In this case a heat exchanger contains both dry and wet regions. The point between two regions is often described in the literature as a pinch point [2]. For dry region of the exchanger, the LMTD method can be used to predict performance. The LMTD method does not work for wet region in the case of condensation of vapor from vapor-air mixture. One of the most convenient way is to use a film model.

The film approach provides a simple physical model for condensation of vapor-air mixtures which is widely used in engineering applications [3-5]. It was found that the film model was accurate in predicting heat exchanger performance in the condensation of steam-air mixtures [5]. In another study, the film model was used successfully to analyze condensation of binary mixtures [4]. The film approach has been also used in this model to determine performance characteristics for condensing vapor-air mixtures.

During condensation of vapor from vapor-air mixtures, the noncondensable collects at the condensing surface as the vapor condenses [3]. This accumulation of noncondensable inhibits the flow of vapor to the interface and reduces the condensation rate. While the condensation rate is reduced, the heat transfer coefficient at the surface is actually increased. Diffusion induced velocity takes place as there is a higher concentration of vapor in the bulk than at the condensate film surface. Enhanced heat exchange was first predicted by Ackermann [4]. An excellent review of past work in film model correction factors is presented in [4]. In [6], researchers observed an increase of heat transfer coefficient due to diffusion induced velocity experimentally.

A Description of the System Being Modeled

The system being modeled is a counter-flow heat exchanger system, and a schematic can be seen in **Figure 1.**

The hot stream consists of a vapor-air mixture while the cold stream consists of cooling water. The properties of the two inlet streams to the exchanger are known. For the air stream we know the inlet temperature, relative humidity, and flow rate. For the water side, we know the inlet temperature and flow rate. From this information, the performance of the exchanger and the properties of the outlet streams can be found. If there is no condensation in the exchanger the LMTD can be used to evaluate performance. If there is condensation in the presence of noncondensables, a more complex model must be used as LMTD theory is no longer accurate.

Describing the System Past the Pinch Point

In deriving the following equations, the common film approach assumptions have been made. The first assumption is that the hot stream has constant properties throughout its bulk. The second assumption is that all heat transfer takes place at the wall. In addition, the condensate film at the wall is assumed to be very thin. This is not a common assumption for the film model, but it allows the condensate and the wall to be at identical temperatures. The effects of vapor shear in the film layer are also disregarded. By applying the conservation laws of mass and energy to a differential area dA, the following equations are derived.

<u>AIR SIDE</u>

For temperature T_h:

$$C_{ph}\frac{d(T_h m_h)}{dA} = -h_h(T_h - T_w) \qquad (1)$$

Since $\frac{dm_h}{dA}$ is many orders of magnitude smaller than $\frac{dT_h}{dA}$, that term is ignored. Using this simplifying assumption, equation (1) is transformed to:

$$C_{ph}m_h \frac{dT_h}{dA} = -h_h(T_h - T_w) \qquad (2)$$

Considering vapor concentration in air stream is negligible small, we assume that mass flow rate m_h is defined by dry air flow rate, which is constant. We have then:

For specific humidity H_h:

$$m_h \frac{dH_h}{dA} = -J \qquad (3)$$

WATER SIDE

For temperature T_c:

$$C_{pc}m_c \frac{dT_c}{dA} = -h_c(T_w - T_c) \qquad (4)$$

WALL BOUNDARY CONDITION:

$$h_h(T_h - T_w)dA + H_{vap}JdA = h_c(T_w - T_c)dA \qquad (5)$$

The boundary condition (5) is needed to balance the energy fluxes at the wall. In the boundary condition it is assumed that the wall provides no resistance to heat transfer.

The air stream may reach a saturation at some point. The bulk temperature T_h can not go bellow a dew point temperature. Therefore, we assume that instead of (1) the bulk temperature T_h is governed by an equation that followed from saturation curve. This equation holds true downstream of the saturation point. It corresponds to an assumption that the convective heat transfer removes from air an amount of heat that is required to keep the bulk temperature exactly equal to the dew point temperature. Thus, taking into account (3) we get equations for downstream of saturation point :

For temperature T_h:

$$m_h \frac{dT_h}{dA} = -(Tsat)'_H J \qquad (2')$$

where $(Tsat)'_H$ is the derivative of saturation temperature over specific humidity H. This function can be found elsewhere.

WALL BOUNDARY CONDITION:

$$C_{ph}(Tsat)'_H JdA + H_{vap} JdA = h_c(T_w - T_c)dA \qquad (5')$$

Condensation Correlation

To complete our system of equations, the rate of vapor condensation at the wall J must also be determined. Without the condensation term (J), the film model equations are reduced to an analytic solution from which LMTD theory is derived. According to the laws of diffusion, the condensation rate should be proportional to the vapor concentration difference between the bulk and the surface where condensation is taking place. In engineering practice, the Berman and Fuks correlation [7] has been widely used to predict condensation rates in the presence of noncondensables [8]:

$$J = 0.52 \frac{D_{vap}}{D_0} \sqrt{\mathrm{Re}}\, (P_{vap,h} - P_{vap,w})^{2/3} P^{1/3} (1 - P_{vap,h}/P)^{-0.6} \qquad (6)$$

where the coefficient of vapor diffusion D_{vap} is related to the diffusion coefficient D by $D_{vap} = D/R_{sat}T_{sat}$. using the Gibbs-Dalton ideal-gas mixture relations, and taking $R_{sat}/R_h = 1.607$, equation (6) gives:

$$J = c_1 \frac{DP}{D_0 RsatTw} \sqrt{\mathrm{Re}}\,(SH_h - SH_w)^{2/3}(1 - 1.607\,SH_h)^{-0.6} \tag{7}$$

where c_1=0.713 is the constant. However, the value c_1=2.495 is used for calculations, because this value provides the best fit of experimental data [17].

It was assumed in (7) that the vapor diffusion coefficient is calculated at a local saturation temperature, that is, at the wall temperature, because the vapor-air mixture at the wall is assumed to be saturated. Therefore the amount of vapor present in the mixture is then determined by the wall temperature [9]. Thus the rate of condensation at the wall is a function of the bulk specific humidity and the wall temperature.

Heat Transfer Coefficient Correlations

The program uses a Nusselt number correlation to estimate the sensible heat transfer coefficient. The following correlations cover four different flow regimes on the hot and cold sides.

For fully developed laminar flow, similar to [10]:

$$Nu = Nu_0 + \frac{0.0677(\mathrm{Pr}\ \mathrm{Re}\ d_{hyd}/L)^{1.33}}{1 + 0.1\ \mathrm{Pr}\ (\mathrm{Re}\, d_{hyd}/L)^{0.83}}$$

where Nu_0 is a theoretical constant, which depends on a shape of tube cross section. For example, $Nu_0 = 4.36$ for a circular tube. For rectangular channels Nu_0 can be approximated by

$$Nu_0 = 4.62(a_{\min}/a_{\max} - 1)^2\, e^{-1.63(a_{\min}/a_{\max})} + 3.61$$

where $a_{\min}$ and $a_{\max}$ are the minimum and maximum sides of the rectangular cross section respectively.

For developing laminar flow [11]:

$$Nu = 1.86(\text{Pr Re } d_{hyd}/L)^{1/3}$$

For turbulent flow [10]:

$$Nu = 0.0235(\text{Re}^{0.8} - 230)(1.8\,\text{Pr}^{0.3} - 0.8)\left[1 + (d_{hyd}/L)^{2/3}\right]$$

For entrance flow [15]:

$$Nu = 0.33\,\text{Re}^{0.5}\,\text{Pr}^{0.33}(d_{hyd}/L_{ent})^{-0.1}$$

where the entrance length L_{ent} is defined as

$$L_{ent}/d_{hyd} = 0.055\,\text{Re Pr}$$

Miscellanies Effects

Fin efficiency:

The program calculates also fin efficiency according to [12]:
for radial fins with rectangular profile:

$$\eta = \frac{2 r_{rel}}{\phi(1+r_{rel})} \frac{I_1(\arg)K_1(r_{rel}\arg) - K_1(\arg)I_1(r_{rel}\arg)}{I_0(r_{rel}\arg)K_1(\arg) + K_0(r_{rel}\arg)I_1(\arg)} \quad ; \quad \arg = \frac{\phi}{1-r_{rel}}$$

$$\phi = (r_e - r_o)^{0.5} m \ ; \qquad m = \left(\frac{2h_h}{k\delta_0}\right)^{1/2} \quad ; \quad r_{rel} = \frac{r_o}{r_e} \quad .$$

where r_o and r_e are radii of the base (outer tube diameter) and the tip of the fin respectively, k is the heat conductivity coefficient, δ_0 is the fin thickness, and $I_i(\,)$, and $K_i(\,)$ (i=0,1) are i-Bessel functions.

for longitudinal fin with rectangular profile:

$$\eta = \frac{\tanh mb}{mb}$$

where b is the fin height

Pressure drop:

Pressure drop is defined as:

$$\Delta p = (f\frac{L}{d_{hyd}} + 1.5)\frac{\rho V^2}{2} \tag{8}$$

where f is the friction coefficient. The value of 1.5 in (8) takes into account energy losses at an entrance and an exit of a tube. For turbulent flow (Re > 3000) the friction coefficient is [13]

$$f = 0.0032 + \frac{0.221}{\mathrm{Re}^{0.237}}$$

Capillary rise:

As a result of a capillary force, condensed water may accumulate on the lower part of heat exchanger and block a portion of airpath. The blockage can be significant for heat exchangers with dense fin arrangement. The program calculates the height of capillary rise H_{cap}:

$$H_{cap} = \frac{2\sigma}{\rho g d_{fin}}$$

where σ is the surface tension, d_{fin} is the distance between fins, and g is the gravity. Note, that capillary rise formula is not applicable at microgravity. The capillary rise then is subtracted from fin height to determine actual vertical dimension of airpath.

Cross-counter flow.

Most heat exchangers employ cross-counter flow arrangement, which is less efficient than pure counter flow. The program determines the cross-counter flow factor η_k, which shows the reduction in heat transfer in a cross-counterflow heat exchanger vs. a pure counterflow heat exchanger [10] :

$$\eta_k = 1 - \frac{C_h}{C_c}\frac{\psi^2}{8} \quad \text{with} \quad \psi < 1 \quad ; \quad \psi = \frac{T_{hin} - T_{hout}}{N(\Delta T_m)^{cont}}$$

where N is the number of rows (in the air flow direction), $(\Delta T_m)^{cont}$ is the Mean Log Average Temperature Difference calculated for pure counterflow. Then, the heat transfer in the given cross-counterflow heat exchanger Q^{cross} is:

$$Q^{cross} = \eta_k Q^{cont} \qquad (9)$$

where Q^{cont} is the calculated heat transfer for pure counterflow heat exchanger.
Outlet air T_{hout}^{cross} and water T_{cout}^{cross} temperatures for cross-counterflow are calculated then according to (9):

$$\begin{aligned} T_{hout}^{cross} &= T_{hin} + \eta_k (T_{hin} - T_{hout}^{cont}) \\ T_{cout}^{cross} &= T_{cin} - \eta_k (T_{cin} - T_{cout}^{cont}) \end{aligned} \qquad (10)$$

where T_{hout}^{cont} and T_{cout}^{cont} are the outlet air and water temperatures respectively calculated for pure counterflow. For rare occasions when $\psi > 1$, the program asks a user to use Fig. 8-22 from [10] to determine the cross-counter flow factor η_k. and to correct outlet temperatures according to (11).

Described above procedure is applied to a dry heat exchanger. In a case of condensation, an overall heat transfer coefficient multiplied by the cross-counter flow factor η_k is used in calculations.

Program Method of Solution

The flow chart given in **Figure 2** describes the method of solution for the program. The region in the exchanger in which no condensation takes place at the interface is called the dry region of the heat exchanger. In the dry region of the exchanger, an analytic solution is used to determine the amount of heat transferred between the hot and cold streams [10]. In wet region of the exchanger (where condensation takes place), equations (2) - (5) are integrated to determine hot, cold side and wall temperatures and specific humidity profiles. Remind again, that the equations (2') and (5') are used instead of (2) and (5) for integration downstream of stream saturation point.

The program integrates equations by using a Runge-Kutta integration scheme [14]. The Runge-Kutta method of integration solves an initial value problem, thus all conditions on one side of the exchanger must be known. As can be seen in **Figure 1**, on the one side of the exchanger the air inlet conditions are known and the water outlet conditions are unknown. In order to use the Runge-Kutta method, a water outlet temperature is estimated and the variables integrated down the length of the exchanger. The calculated inlet water temperature is then found and can be compared to the known inlet water temperature. It may take more than one estimation to match the calculated cold stream inlet temperature to the actual temperature so the shooting method described in the flow chart is needed.

Initially, the program calculates various constants that will be needed to find solutions in the dry portion of the heat exchanger (See **Figure 1**). These constants include the dew point of the hot stream at the inlet (Thdin), the heat transfer coefficients on the hot and cold sides, the overall heat transfer coefficient (U) for the dry portion of the heat exchanger, anticipated capillary rise, fin efficiency, and etc. Then the program determines if the entire heat exchanger is dry. This is accomplished by assuming that the wall temperature at the outlet of the exchanger (Twout) is at the dew point of the hot stream (Thdin). If the calculated cold stream inlet temperature (Tcinc) is less than the actual cold stream inlet temperature (Tcin), then the entire heat exchanger is dry and the analytical solution of [10] can be used to find outlet temperatures for the heat exchanger.

If a wet section of the system is present, first the program checks to see if the entire exchanger is wet. The inlet wall temperature (Twin) is assumed to equal to the inlet dew point temperature (Thdin). Making these two assumptions, an inlet cold stream temperature is calculated (Tcinc) by integrating equations from the inlet to the outlet of the exchanger. If the calculated cold stream inlet temperature (Tcinc) is greater than the actual inlet temperature (Tcin), then the entire heat exchanger is wet. The inlet wall temperature is then lowered until the calculated inlet temperature (Tcinc) is identical to the actual temperature (Tcin).

If the exchanger is not entirely wet, then there is a pinch point somewhere in the exchanger. The program initially assigns the pinch point to the middle of the exchanger. The program uses the analytic solution [10] to describe heat transfer in the dry section (air upstream of the pinch point). For the area in the exchanger where condensation takes place, equations (2 or 2')-(5 or 5') are integrated to the end of the exchanger. Once the calculated cold stream inlet temperature (Tcinc) has been found, it is compared to the actual cold stream inlet temperature. If the calculated temperature is too high, the pinch point is moved towards the inlet. If the calculated temperature is too low, the pinch point is moved towards the outlet. This process is continued until the calculated inlet temperature (Tcinc) matches the actual inlet temperature (Tcin).

The program then reports the outlet flow rate, relative humidity, and temperatures on both sides to the user. The program is designed to provide accurate results for heat exchangers in which the hot stream bulk temperature remains above or equal to its dew point. If the bulk temperature drops below its dew point, fog formation may start in the bulk [16]. This topic, and the adaptation of this program to include fog formation will be covered in a later paper.

Comparison with Experimental Results

The program has been used to calculate performances of a plate-fin heat exchanger made by Hamilton Standard (Windsor Locks, CN) [17] and a tube-fin heat exchanger made by Super Radiator Coils (Minneapolis, MN). The comparison of data is shown in Table 1.

Table 1.- Experimental Data Compared to Theoretical Model

	"Hamilton Standard" plate-fin HX		"Super Radiator Coils" tube-fin HX			
	Mrf. data	Our results	Mrf. data *	Our results	Mrf. data **	Our results
air flow (scfm)	330		224.0		150	
air in temp. C°	40.4		19.0		35.8	
air in r.h. %	22.2		67.0			
water flow lpm	7.62		3.78		1.89	
water in temp. C°	6.4		10.7		26	
air out temp. C°	11.3	12.25	11.8	11.73	26.8	26.21
air out r.h. %		100	100	100		
water out temp. C°	17.1	16.82	14.9	14.83	31.8	31.83
condensation rate/day	23.8	22.9	6	5.97	0	0
sensible heat W	5029	4912	931	922	766	
latent heat W	683	649	180	171	0	
heat transfer W	5723	5561	1111	1093	766	769

* coil model 6 x 37.5 - 8R - 38/192

** coil model 4 x 50 - 7R - 38/144

Data in Table 1 shows that the developed program gives results very close to those given by manufacturers. It indicates that this general program can be used for different specific applications.

The next improvement of the program is to be inclusion of fog formation. It allows us to determine more precise temperature distribution and condensation rate correspondingly.

Acknowledgment

The author would like to express his thanks to Carl Rhodes of Stanford University for help in preparation of the paper.

Appendix

<u>Nomenclature</u>

A - Area [m^2]

a - Duct size

b - Constant

c_1 - Constant

C_p - Heat Capacity $\left[\frac{J}{kgK}\right]$

D - Diffusion Coefficient $\left[\frac{m^2}{sec}\right]$

d_{hyd} - Hydraulic Diameter [m]

D_0 - Pipe Diameter [m]

H - Specific Humidity $\left[\frac{kg(water)}{kg(dry\ air)}\right]$

h - Heat transfer coefficient $\left[\frac{W}{m^2K}\right]$

Hvap - Heat of vaporization $\left[\frac{J}{kg}\right]$

J - Mass flux to wall $\left[\frac{kg}{m^2\,sec}\right]$

k - Heat conductivity coefficient $\left[\frac{W}{mK}\right]$

L - Exchange Length [m]

m - Mass flow rate $\left[\frac{kg}{sec}\right]$

n - Constant

MW - Molecular weight $\left[\frac{kg}{mol}\right]$

P - Pressure [Pa]

Pr - Prandtl Number

R - Gas Constant $\left[\frac{J}{mol\ K}\right]$

ρ - Density $\left[\frac{kg}{m^3}\right]$

Re - Reynolds Number

RH - Relative Humidity

Sc - Schmidt Number

T - Temperature [^{0}C]

U - Overall Heat Transfer Coefficient $\left[\frac{W}{m^2K}\right]$

y - Mole Fraction

<u>Subscripts used</u> :

c - Cold stream (water)

d - Dew Point

ent - Entrance

g - Noncondensable gas

h - Hot stream (gaseous)

in - Inlet (actual)

inc - Inlet (calculated)

out - Outlet

sat - Saturated

vap - Vapor

w - Wall

<u>Superscripts used</u>:

cont - Counterflow

cross - Cross-Counterflow

References

1. Rohsenow, W. M.; and Choi, H.: Heat, Mass, and Momentum Transfer. Prentice-Hall, Englewood Cliffs. 1961, pp. 237-256.
2. Wilmot, G. E.; and Christians, D. L.: Heat Exchanger Modelling Accounting for Condensation and Temperature Pinch. Thirteenth Intersociety Conference on Environmental Systems, San Francisco. 1983.
3. Peterson, P. F.: Species-Controlled Condensation in Enclosures, Annual Review of Heat Transfer. Vol. 4, Hemisphere Publishing Corp., New York, 1992, pp. 247-285.
4. Brouwers, H. J. H.; and Chesters, A. K.: Film Models For Transport Phenomena With Fog Formation: The Classic Film Model. Int. J. Heat Mass Transfer, vol. 35, 1992, pp. 1-11.
5. Webb, D. R.; and Panagoulias D.: An improved approach to condenser design using film models. Int. J. Heat Mass Transfer, vol. 30, 1987, pp. 373-378.
6. Aggarwal, J. K.; and Hollingsworth M. A.: Heat Transfer for Turbulent Flow With Suction in a Porous Tube. Int. J. Heat Mass Transfer, vol. 16, 1973, pp. 591-609 .
7. Berman, L. D.; and Fuks, S. N.: Mass Transfer in Condensers With Horizontal Tubes When the Steam Contains Air. Teploenergetica, vol. 5, 1958, pp. 66-74.
8. Davidson, B. J.: Steam Condenser Thermal Design Theories. von Karmen Institute Lecture Series 1983-06, Aerothermodynamics of Low Pressure Steam Turbines and Condensers, Rhode Saint Genese, Belgium, 1983.
9. Reid, R. C.; Prausnitz, J. M.; and Sherwood, T. K.: The Properties of Gases and Liquids, McGraw Hill, New York, 1977, pp. 184-185.
10. Hausen H.: Heat Transfer in Counterflow, Parallel Flow, and Cross Flow. McGraw-Hill, New York, 1983, pp. 31, 37, 144-169.
11. Incropera, F. P.; and DeWitt, D. P.: Fundamentals of Heat and Mass Transfer. John Wiley, New York, 1985.
12. Kern, D.Q.; and Kraus, A. D.: Extended Surface Heat Transfer. McGraw-Hill, New York, 1972.
13. Loystansky, L.G.: Fluid and Gas Mechanics., Publish House "Nauka," Moscow, 1970, pp. 730.
14. Press, W. H. et al.: Numerical Recipes - The Art of Scientific Computing. Cambridge University Press, New York, 1986 pp. 547-560.
15. Kogan, B. B.: Theoretical Principals of Processes of Chemical Engineering Technology in Russian. Publish House "Khimia," Leningrad, 1977, pp. 297, 299.
16. Steinmeyer, D. E.; Fog Formation in Partial Condensers, Chem Engng Prog., vol. 68, 1972, pp.64-68.
17. Humidity Control Heat Exchanger, in: "Component Data Handbook, Environmental Control and Life Support Systems." Hamilton Standard, Jan. 1980.

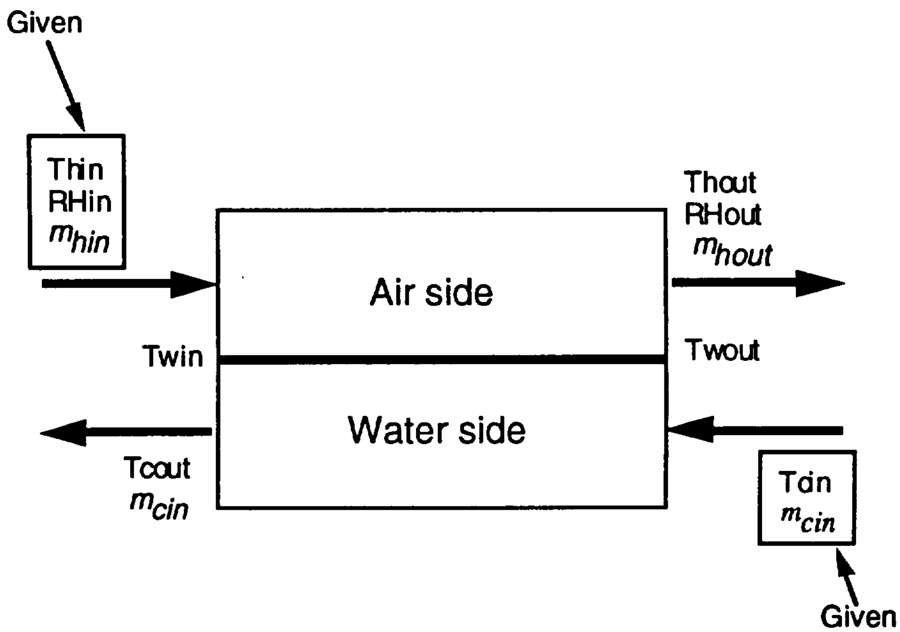

Figure 1.- Heat Exchanger Schematic

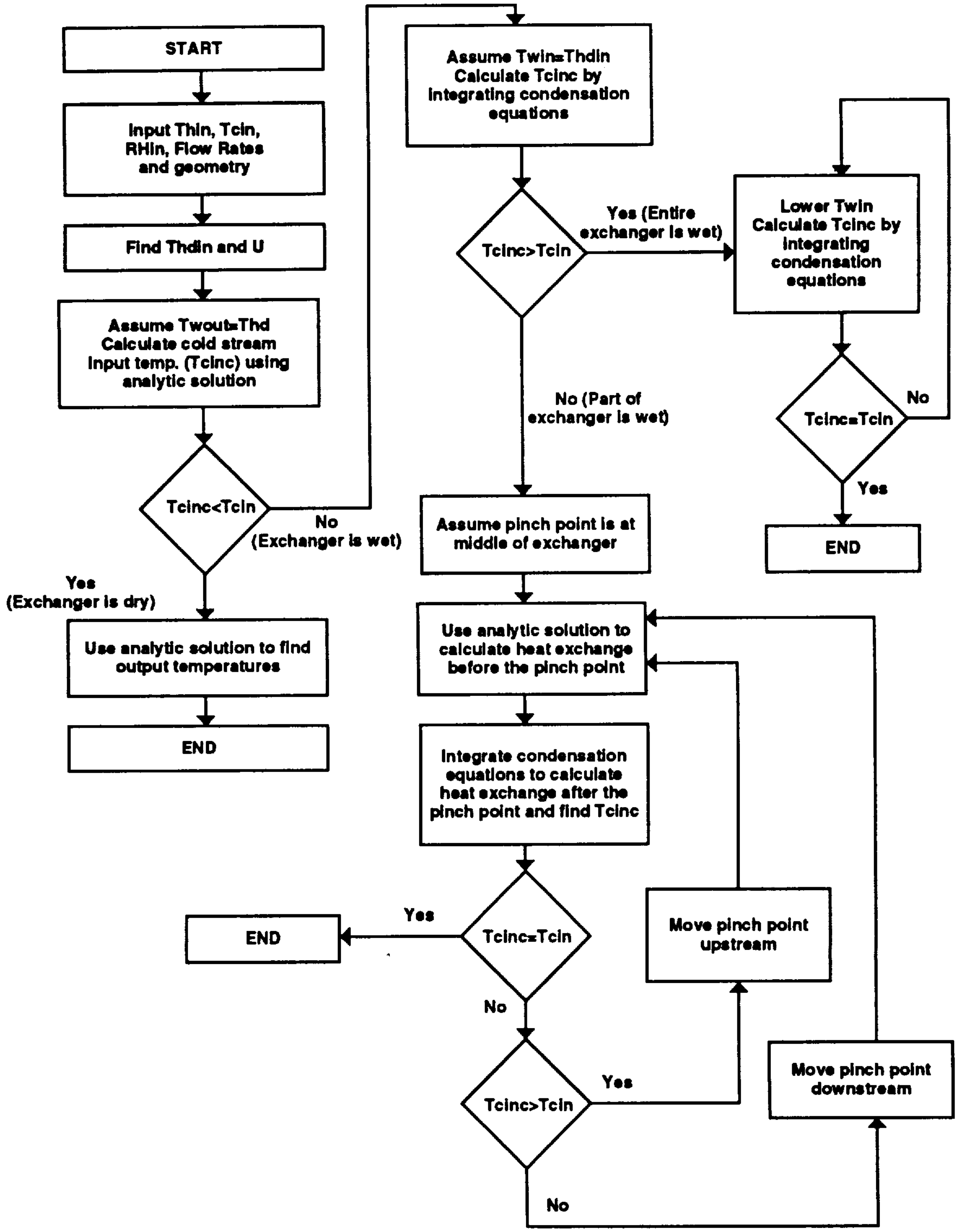

Figure 2.- Program Flow Chart

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY *(Leave blank)*	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	June 1994	Contractor Report

4. TITLE AND SUBTITLE

A Computer Program for Condensing Heat Exchanger Performance in the Presence of Noncondensable Gases

5. FUNDING NUMBERS

NAS2-13273

6. AUTHOR(S)

Boris Yendler

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

The Bionetics Corporation
10th Floor, Suite 100
Harbour Centre Building
2 Eaton Street
Hampton, VA 23669-4062

8. PERFORMING ORGANIZATION REPORT NUMBER

A-94097

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)

National Aeronautics and Space Administration
Washington, DC 20546-0001

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

NASA CR-177643

11. SUPPLEMENTARY NOTES

Point of Contact: R. D. MacElroy, Ames Research Center, MS 239-23, Moffett Field, CA 94035-1000; (415) 604-1264

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Unclassified — Unlimited
Subject Category 34

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

A computer model has been developed which evaluates the performance of a heat exchanger. This model is general enough to be used to evaluate many heat exchanger geometries and a number of different operating conditions. The film approach is used to describe condensation in the presence of noncondensables. The model is also easily expanded to include other effects like fog formation or suction.

14. SUBJECT TERMS

Heat exchanger, Computer program, Condensation

15. NUMBER OF PAGES

22

16. PRICE CODE

A03

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified		