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NATIONAL BUREAU OF STANDARDS, Ernest Ambler, *Director*

ABSTRACT

A mathematical model of the steady-state performance of an absorption heat pump is described. The model is compared with experimental data from a residential-sized water chiller. It is also used to determine the sensitivity of the heat pump performance to its design variables.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

1. INTRODUCTION

A mathematical model of the steady-state performance of an absorption heat pump has been developed and coded into a FORTRAN program. The model is based on mass and energy balances about each system component and it incorporates heat transfer rate limitations. A detailed description of the model appears in section 2.

One application for this model is to check the consistency of experimental data. Thus far, the model has been compared with the steady-state performance of an ARKLA 3-ton ammonia-water chiller (model no. ACC 3600). The experimental data are reported in section 3. Comparisons of the model to these data appear in section 4.

The model can also be a useful tool in the analysis and design of absorption heat pumps through sensitivity studies. Used in this manner, the model indicates how the steady-state performance is affected by changes in the design parameters or operating conditions. The results of a sensitivity study are reported in section 5.

The model has been written to be independent of the absorbent-refrigerant system; any absorbent-refrigerant system for which property data are available can be used with the model. The required property data subroutines are described in appendix A. A listing of the program appears in appendix B.

2. DESCRIPTION OF THE MODEL

The absorption heat pump model is coded in a modular manner; i.e., a separate FORTRAN subroutine is used for each system component. There are several advantages to this modular approach. First, it allows simulation of many different system configurations without major changes to the model since additional heat exchangers, pumps, throttles, etc. can be included as desired. Second, the modular approach simplifies the program development in that improved component models can be incorporated as they are developed. A disadvantage of the modular approach is that the set of equations describing the heat pump performance can not be solved explicitly. An iterative solution is required.

There are three distinct types of information associated with each component model: parameters, inputs, and outputs. The parameters are the design specifications of the component. In the component models described below, the parameters are primarily factors to account for heat transfer limitations. The inputs to a component are the characteristics of the fluid streams flowing into that component. Each stream has six characteristics:

- T - temperature [°F]
- P - pressure [psia]
- x - overall composition [lb refrigerant/lb solution]
- h - enthalpy [BTU/lb]
- q - quality [lb vapor/lb liquid + vapor]
- $\dot{m}$ - flow rate [lb/hr]

(Other stream characteristics are density, viscosity, thermal conductivity, specific heat, etc., but they are not needed in this model.) The outputs of a component are the characteristics of the outlet streams and any external heat flows such as the evaporator heat input. In steady-state operation, the inputs and outputs are constants with respect to time. The distinction between parameters and inputs is that parameters are known constants specified by the user; the inputs are generally unknown and must be found by an iterative solution technique.

The iterative solution proceeds in the following way. Each component subroutine is designed to calculate the outputs of the component using the supplied parameter and input values. The outputs of one component are inputs to other components in the system. The component subroutines are executed and with each iteration, the input/output values are improved. The order in which the subroutines are executed is unimportant. The iteration proceeds until the outputs (and thus the inputs) of all routines converge. The final solution is thus found by successive substitution.

A schematic diagram of an ARKLA 3-ton chiller is shown in figure 1. Due to its modular form, the model is not limited to absorption machines of this particular configuration, but the model was developed for the ARKLA unit. Sixteen flow stream locations are identified by numbers in the diagram and are referred to in the component model descriptions below.

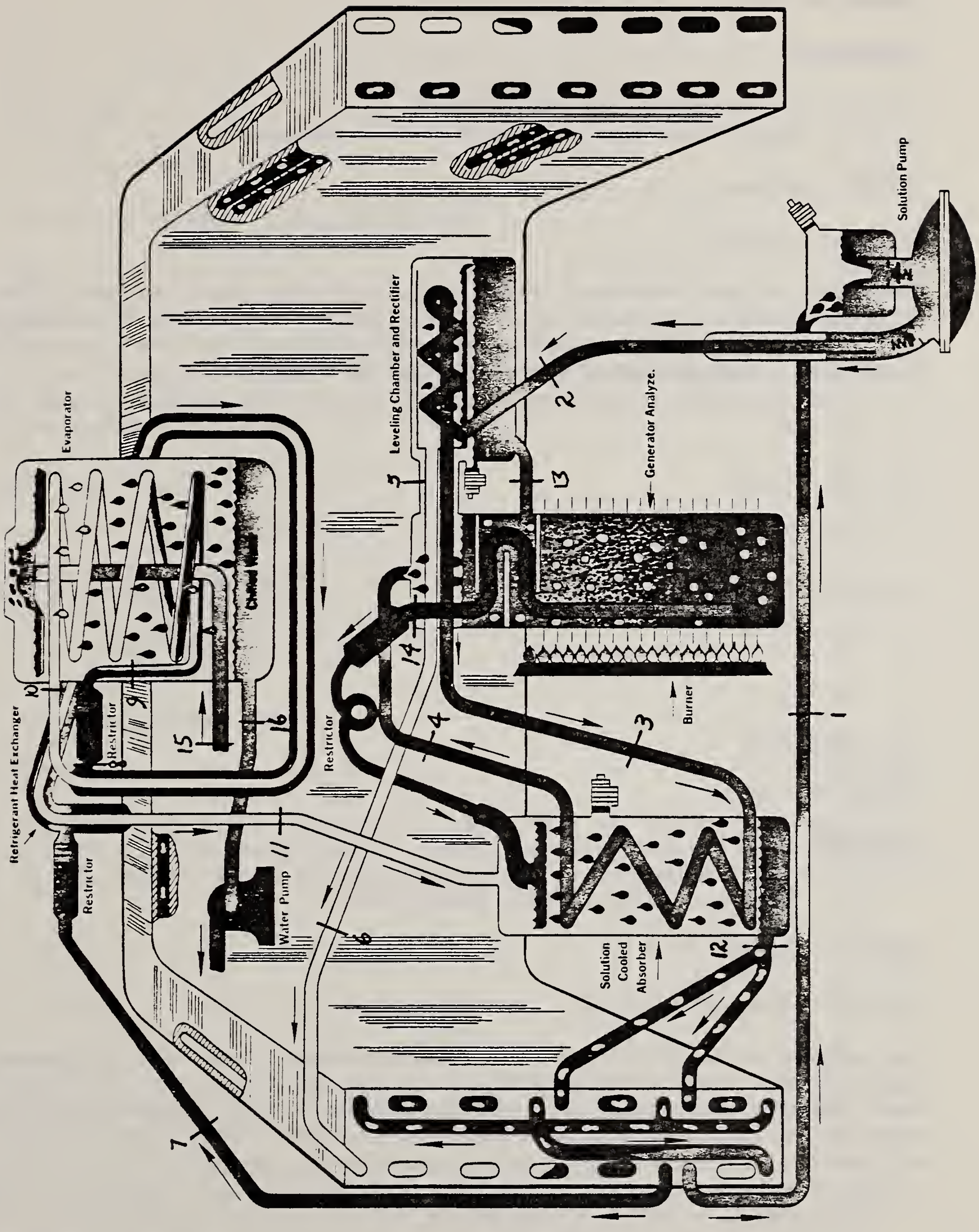

Figure 1. Schematic diagram of ARKLA chiller model ACC-36-00

2.1 SOLUTION PUMP

SUBROUTINE PUMP

PARAMETERS:

1. $\dot{m}$ - solution flow rate [lb/hr]
2. P_{high} - pressure on high side of cycle [psia]

INPUTS: Stream 1

OUTPUTS: Stream 2

The solution pump requires a relatively small work input and has little effect on thermodynamic properties other than pressure. The model is simplistic in that the work input is assumed to be negligible and the flow rate must be specified. A mathematical description is as follows.

$$\dot{m}_2 = \dot{m} \tag{2.1.1}$$

$$P_2 = P_{high} \tag{2.1.2}$$

$$T_2 = T_1 \tag{2.1.3}$$

$$x_2 = x_1 \tag{2.1.4}$$

$$h_2 = h_1 \tag{2.1.5}$$

$$q_2 = q_1 \tag{2.1.6}$$

2.2 RECTIFIER

SUBROUTINE RECT

PARAMETERS:

1. $(UA)_{rect}$ - overall heat conductance

INPUTS: Streams 2, 5

OUTPUTS: Streams 3, 6, 13

The purpose of the rectifier is to condense water vapor from the ammonia-water vapor mixture leaving the generator. In the ARKLA unit, the rectifier is a heat exchange coil acting as a partial condenser. In the model, it is assumed that the rectifier is adiabatic and that streams 6 and 13 are in equilibrium so that

$$T_6 = T_{13} \tag{2.2.1}$$

$$P_6 = P_{13} \tag{2.2.2}$$

Mass balances yield

$$\dot{m}_3 = \dot{m}_2 \tag{2.2.3}$$

$$x_3 = x_2 \tag{2.2.4}$$

$$\dot{m}_6 + \dot{m}_{13} = \dot{m}_5 \tag{2.2.5}$$

$$\dot{m}_6 x_6 + \dot{m}_{13} x_{13} = \dot{m}_5 x_5 \tag{2.2.6}$$

Pressure losses are assumed to be negligible.

$$P_6 = P_5 \tag{2.2.7}$$

$$P_3 = P_2 \tag{2.2.8}$$

The rate of heat transfer between the stream flowing within the coil and the condensing vapor is expressed in terms of an overall heat conductance and a log mean temperature difference.

$$\dot{Q} = (UA)_{rect}\, \Delta T_{lm} \tag{2.2.9}$$

where

$$\Delta T_{lm} = \frac{(T_5 - T_3) - (T_6 - T_2)}{\ln\left(\frac{T_5 - T_3}{T_6 - T_2}\right)} \tag{2.2.10}$$

An energy balance requires that

$$\dot{Q} = \dot{m}_2 (h_3 - h_2) = \dot{m}_5 h_5 - \dot{m}_6 h_6 - \dot{m}_{13} h_{13} \tag{2.2.11}$$

Equations 2.2.5 through 2.2.11 are solved iteratively as follows. A guess is made for T_6 (= T_{13}). This temperature is used with the property data subroutines to determine the composition and specific enthalpy of streams 6 (vapor) and 13 (liquid).

$$x_{13} = XLPT(P_{13}, T_{13})$$

$$h_{13} = HL(x_{13}, T_{13}, P_{13})$$

$$x_6 = XV(x_{13}, P_6)$$

$$h_6 = HV(x_{13}, x_6, T_6, P_6)$$

The mass flow rates, $\dot{m}_6$ and $\dot{m}_{13}$, are found from equations 2.2.5 and 2.2.6. Using these flow rates, the rate of heat transfer is evaluated form the right-hand side of equation 2.2.11, which then determines h_3. The temperature of state 3 is fixed by its enthalpy, pressure, and composition, and it can be determined from the property data subroutines as follows.

$$T_3 = \begin{cases} TH(x_3,\ h_3,\ P_3) \text{ if } T_3 < TSAT(x_3,\ P_3) \\ TQ(x_3,\ P_3,\ h_3,\ q_3) \text{ otherwise} \end{cases}$$

This value of T_3 is then used with equations 2.2.9 and 2.2.10 to determine the rate of heat transfer. The difference between this result and that obtained from equation 2.2.11 is used to find an improve value of T_6 (= T_{13}) by the Secant method.(1)

2.3 SOLUTION - COOLED ABSORBER

SUBROUTINE ABSSC

PARAMETERS:

1. $(UA)_{abss}$ - overall heat conductance

INPUTS: Streams 3, 11, 14

OUTPUTS: Streams 4, 12

Absorption of refrigerant in the ARKLA unit occurs in two components; a solution-cooled and an air-cooled absorber. In the solution-cooled absorber, the hot weak (in refrigerant) solution from the generator is sprayed over a coil containing the cooler rich solution which is thereby preheated before it enters the generator during this process. The refrigerant vapor is partially absorbed in the weak solution. It is also entrained so that a two-phase solution exits the absorber at state 12.

Mass balances require that

$$\dot{m}_4 = \dot{m}_3 \tag{2.3.1}$$

$$x_4 = x_3 \tag{2.3.2}$$

$$\dot{m}_{12} = \dot{m}_{11} + \dot{m}_{14} \tag{2.3.3}$$

$$\dot{m}_{12}x_{12} = \dot{m}_{11}x_{11} + \dot{m}_{14}x_{14} \tag{2.3.4}$$

Pressure losses are neglected.

$$P_4 = P_3 \tag{2.3.5}$$

$$P_{12} = P_{11} \tag{2.3.6}$$

The rate of heat transfer between the stream flowing in the coil and the absorbent is expressed in terms of an overall heat conductance and a log mean temperature difference.

$$\dot{Q} = (UA)_{abss}\ \Delta T_{lm} \tag{2.3.7}$$

where

$$\Delta T_{lm} = \frac{(T_{14} - T_4) - (T_{12} - T_3)}{\ln\left(\dfrac{T_{14} - T_4}{T_{12} - T_3}\right)} \tag{2.3.8}$$

Energy balances require that

$$\dot{Q} = \dot{m}_3(h_4 - h_3) = \dot{m}_{11}h_{11} + \dot{m}_{14}h_{14} - \dot{m}_{12}h_{12} \tag{2.3.9}$$

Equations 2.3.7 through 2.3.9 must be solved iteratively for T_4 and T_{12}. A guess is made for T_4. The enthalpy at state 4 is then obtained from property data routines. If the guess value of T_4 is less than the saturation temperature, stream 4 is subcooled. In this case

$$h_4 = HL(x_4, T_4, P_4) \text{ for } T_4 < TSAT(x_4, P_4)$$

$$q_4 = 0$$

For two-phase conditions ($T_4 > TSAT(x_4, P_4)$),

$$x_4^L = XLPT(P_4, T_4)$$

$$x_4^V = XV(x_4^L, P_4)$$

$$h_4^L = HL(x_4^L, T_4, P_4)$$

$$h_4^V = HV(x_4^L, x_4^V, T_4, P_4)$$

$$q_4 = (x_4 - x_4^L)/(x_4^V - x_4^L)$$

$$h_4 = (1 - q_4)h_4^L + q_4h_4^V$$

The enthalpy of state 12 and rate of heat transfer can now be evaluated from equation 2.3.9. The temperature and quality of state 12 are fixed by the pressure, overall composition and enthalpy of the mixture and can be determined from property data routine TQ.

$$T_{12} = TQ(x_{12}, P_{12}, h_{12}, q_{12})$$

The rate of heat transfer can now be calculated from equations 2.3.7 and 2.3.9. The difference between the values of Q calculated from equations 2.3.7 and 2.3.9 is used to find an improved value of T_4 by the Secant method. The iterations are terminated when two successive values of T_4 are within a specified tolerance.

2.4 AIR-COOLED ABSORBER

SUBROUTINE CONABS

This subroutine is used for evaluating the performance of the air-cooled absorber as well as for the condenser as outlined below.

PARAMETERS:

1. $(UA)_{absa}$ - overall heat conductance
2. T_{air} - air temperature

INPUTS: Stream 12
OUTPUTS: Stream 1, $\dot{Q}_{absa}$

A two-phase mixture enters the air-cooled absorber and it is cooled (and condensed) by heat transfer to the air stream. Continuity requires that

$$\dot{m}_1 = \dot{m}_{12} \tag{2.4.1}$$

$$x_1 = x_{12} \tag{2.4.2}$$

Pressure losses are neglected.

$$P_1 = P_{12} \tag{2.4.3}$$

The rate of heat transfer between the solution and the air is expressed using a log-mean temperature difference relation.

$$\dot{Q}_{absa} = (UA)_{absa}\,\Delta T_{lm} \tag{2.4.4}$$

where

$$\Delta T_{lm} = \frac{T_{12} - T_1}{\ln\left(\dfrac{T_{12} - T_{air}}{T_1 - T_{air}}\right)} \tag{2.4.5}$$

An energy balance requires that

$$\dot{Q}_{absa} = (h_{12} - h_1)\dot{m}_1 \tag{2.4.6}$$

Equation 2.4.4 through 2.4.6 are solved iteratively. A guess is made for T_1 and Q_{absa} is evaluated with equation 2.4.4. The enthalpy of state 1 can also be found using the assumed temperature. If state 1 is subcooled then

$$h_1 = HL(x_1, T_1, P_1) \text{ for } T_1 < TSAT(x_1, P_1)$$

$$q_1 = 0$$

If state 1 is two-phase then $T_1 \geq TSAT(x_1, P_1)$ and

$x_1^L = XLPT(P_1, T_1)$

$x_1^V = XV(x_1^L, P_1)$

$h_1^L = HL(x_1^L, T_1, P_1)$

$h_1^V = HV(x_1^L, x_1^V, T_1, P_1)$

$q_1 = (x_1 - x_1^L)/(x_1^V - x_1^L)$

$h_1 = (1 - q_1)h_1^L + q_1 h_1^V$

Q_{absa} can now be calculated from equation 2.4.6. The difference between this result and the value of Q_{absa} found from equation 2.4.4 is used to find an improved value of T_1 by the Secant method.

2.5 GENERATOR

SUBROUTINE GEN

PARAMETERS:

1. $\dot{Q}_{gen}$ - rate of heat input
2. ε_L - heat transfer effectiveness between the exiting weak solution and the entering rich solution
3. ε_V - heat transfer effectiveness between the exiting vapor and the entering rich solution

INPUTS: Streams 4, 13

OUTPUTS: Streams 5, 14

Refrigerant is separated from the absorbent by the heat input to the generator. The high pressure of the boiling refrigerant forces the weak solution at the lower part of the generator through a pipe leading to the solution-cooled absorber. In the ARKLA unit, the pipe is arranged so that the weak solution is placed in a heat exchange situation with the entering rich solution. Similarly, heat exchange occurs between the exiting vapor, and the incoming rich solution.

Mass balances yield

$$\dot{m}_5 + \dot{m}_{14} = \dot{m}_4 + \dot{m}_{13} \qquad (2.5.1)$$

$$\dot{m}_5 x_5 + \dot{m}_{14} x_{14} = \dot{m}_4 x_4 + \dot{m}_{13} x_{13} \qquad (2.5.2)$$

Pressure losses are neglected

$$P_5 = P_{14} = P_4 = P_{13} \qquad (2.5.3)$$

Internal heat exchanges between the incoming rich solution and the exiting streams are described in terms of effectiveness factors. For the exiting vapor,

$$T_5 = T_{gen} - \varepsilon_V(T_{gen} - T_4) \qquad (2.5.4)$$

and for the exiting weak solution,

$$T_{14} = T_{gen} - \varepsilon_L(T_{gen} - T_4) \qquad (2.5.4)$$

where T_{gen} is the temperature of the weak solution at the lower part of the generator where it enters the pipe.

An energy balance around the generator requires that

$$\dot{m}_5 h_5 + \dot{m}_{14} h_{14} = \dot{m}_{13} h_{13} + \dot{m}_4 h_4 + \dot{Q}_{gen} \qquad (2.5.6)$$

The above equations are solved iteratively. A guess is made for T_{gen}. The concentration (at equilibrium) of the weak solution is thus established.

$$x_{14} = XLPT(P_{14}, T_{gen})$$

The temperatures of the exiting streams are determined from equations 2.5.4 and 2.5.5. These temperatures are used to determine the specific enthalpies of the exiting streams and the composition of the vapor.

$$h_{14} = HL(x_{14}, T_{14}, P_{14})$$

$$x_5 = XV(x_5^L, P_5)$$

$$h_5 = HV(x_5^L, x_5, T_5, P_5)$$

where $x_5^L = XLPT(P_5, T_5)$

With the concentrations of the exiting streams determined, equations 2.5.1 and 2.5.2 are used to determine the mass flow rates, which are then used with the calculated enthalpies in equation 2.5.6. The difference between the left and right hand sides of equation 2.5.6 is used to find an improved value of T_{gen} by the Secant method. The iterations are terminated when two successive values of T_{gen} are within a specified tolerance.

2.6 CONDENSER

SUBROUTINE CONABS

PARAMETERS:

1. $(UA)_{cond}$ - overall heat conductance

2. T_{air} - air temperature

INPUTS: Stream 6

OUTPUTS: Stream 7, $\dot{Q}_{cond}$

Rerigerant vapor, along with a small amount of water is condensed and possibly subcooled in the condenser.

Continuity requires that

$$\dot{m}_7 = \dot{m}_6 \tag{2.6.1}$$

$$x_7 = x_6 \tag{2.6.2}$$

Pressure losses are neglected

$$P_7 = P_6 \tag{2.6.3}$$

The heat transfer rate between the refrigerant and the air is expressed using a log-mean temperature difference relation.

$$\dot{Q}_{cond} = (UA)_{cond} \, \Delta T_{lm} \tag{2.6.4}$$

where

$$\Delta T_{lm} = \frac{T_6 - T_7}{\ln\left(\frac{T_6 - T_{air}}{T_7 - T_{air}}\right)} \tag{2.6.5}$$

An energy balance requires that

$$\dot{Q}_{cond} = (h_6 - h_7)\dot{m}_7 \tag{2.6.6}$$

The heat transfer processes occurring in the condenser are identical to those in the air-cooled absorber. As a result, the same subroutine is used for both components. The program logic is described in section 2.4.

2.7 REFRIGERANT HEAT EXCHANGER

SUBROUTINE HXLV

PARAMETERS:

1. $(UA)_{hx}$ - overall heat conductance

INPUTS: Streams 7, 10

OUTPUTS: Streams 8, 11

Liquid refrigerant from the condenser is cooled further in the refrigerant heat exchanger by the refrigerant vapor leaving the evaporator. Mass balances require that

$$\dot{m}_8 = \dot{m}_{11} = \dot{m}_7 = \dot{m}_{10} \tag{2.7.1}$$

$$x_8 = x_{11} = x_7 = x_{10} \tag{2.7.2}$$

Pressure losses are neglected

$$P_8 = P_{11} = P_7 = P_{10} \tag{2.7.3}$$

The rate of heat exchange between the streams entering and exiting the evaporator is expressed

$$\dot{Q}_{hx} = (UA)_{hx} \frac{(T_8 - T_{10}) - (T_7 - T_{11})}{\ln\left(\frac{T_8 - T_{10}}{T_7 - T_{11}}\right)} \tag{2.7.4}$$

Also, an energy balance requires that

$$\dot{Q}_{hx} = (h_{11} - h_{10})\dot{m}_{10} = (h_8 - h_7)\dot{m}_7 \tag{2.7.5}$$

Equations 2.7.4 and 2.7.5 are solved iteratively. A guess is made for T_8, which establishes the enthalpy of T_8.

$$h_8 = HL(x_8, T_8, P_8)$$

The enthalpy of state 11 is then directly determined from equation 2.7.5 which establishes its temperature and quality.

$$T_{11} = TQ(x_8, P_8, h_8, q_8)$$

With T_8 and T_{11} known, the rate of heat exchange can be calculated from equation 2.7.4. The difference in the values of Q_{hx} obtained from equations 2.7.4 and 2.7.5 is used to improve the estimate of T_8 using the Secant method. Iteration stops when two successive values of T_8 are within a specified tolerance.

2.8 RESTRICTOR

SUBROUTINE THROT

PARAMETERS:

1. P_{evap} - low side pressure [psia]

INPUTS: Stream 8

OUTPUTS: Stream 9

In the ARKLA chiller, the refrigerant leaving the condenser is reduced to the evaporator pressure as it passes through two refrigerent restrictors. In the model, the drop in pressure is obtained by a single restrictor. Mass and energy balances require that

$$\dot{m}_9 = \dot{m}_8 \tag{2.8.1}$$

$$x_9 = x_8 \tag{2.8.2}$$

$$h_9 = h_8 \tag{2.8.3}$$

The restrictor outlet stream will be a two-phase mixture. The temperature and quality of this stream can be determined since its composition, pressure and enthalpy are known.

$$T_9 = TQ(x_9,\ P_9,\ h_9,\ q_9)$$

2.9 EVAPORATOR

PARAMETERS:

1. $(UA)_{evap}$ - overall heat conductance between the evaporating refrigerant and the chilled water

INPUTS: Streams 9, 15

OUTPUTS: Streams 10, 16, $\dot{Q}_{evap}$

In the ARKLA chiller, water is cooled as is drips over a coil through which the evaporating refrigerant flows. The refrigerant may exit the evaporator in a two-phase or superheated state.

In earlier versions of this program, the small amount of water (2-3 percent on a weight basis) with the refrigerant was neglected. It was found however, that the water has substantial effect on the evaporator performance and it must be accounted for.

Mass balances require that

$$\dot{m}_{10} = \dot{m}_9 \tag{2.9.1}$$

$$x_{10} = x_9 \tag{2.9.2}$$

$$\dot{m}_{16} = \dot{m}_{15} \tag{2.9.3}$$

Pressure losses are neglected

$$P_{10} = P_9 \tag{2.9.4}$$

The rate of heat transfer between the refrigerant and the water is expressed

$$\dot{Q}_{evap} = (UA)_{evap}\Delta T_{lm} \tag{2.9.5}$$

where

$$\Delta T_{lm} = \frac{T_{10} - T_9}{\ln\left(\frac{T_{15} - T_9}{T_{15} - T_{10}}\right)} \tag{2.9.6}$$

The log-mean temperature difference assumes this form since the design of the evaporator causes the refrigerant at both the inlet and outlet of the evaporator to be in heat exchange contact with the entering chilled water.

An energy balance requires that

$$\dot{Q}_{evap} = (h_{10} - h_9)\dot{m}_9 = (h_{16} - h_{15})\dot{m}_{15} \tag{2.9.7}$$

Equations 2.9.5 through 2.9.7 are solved iteratively. A guess is made for T_{10}. The log-mean temperature difference and rate of heat transfer are then calculated from equations 2.9.6 and 2.9.5, respectively.

With water present in the refrigerant, the refrigerant always exits in two-phase. Its enthalpy and quality are found as follows:

$$x_{10}^L = XLPT(P_{10}, T_{10})$$

$$x_{10}^V = XV(x_{10}^L, P_{10})$$

$$q_{10} = (x_{10} - x_{10}^L)/(x_{10}^V - x_{10}^L)$$

$$h_{10}^L = HL(x_{10}^L, P_{10})$$

$$h_{10}^V = HV(x_{10}^L, x_{10}^V, T_{10}, P_{10})$$

$$h_{10} = (1 - q_{10})h_{10}^L + q_{10}h_{10}^V$$

Using this enthalpy, the rate of heat transfer is calculated from equation 2.9.7. The difference between the values obtained from equations 2.9.5 and 2.9.7 is used to improve the estimate of T_{10} with the Secant method. Iteration is stopped when two successive values of T_{10} are within a specified tolerance.

3. EXPERIMENTAL DATA

The steady-state performance of an ARKLA water chiller (model no. ACC 36 00) was measured in an environmental chamber at air temperatures of 80°F, 95°F, and 100°F. In all tests, 55°F water was supplied to the unit at 7.2 GPM. The test apparatus and procedures were as described by Lindsay and Didion[2] with the following modifications.

1. A micro-computer (CROMEMCO) was interfaced with the FLUKE data logger to improve the analysis of the experimental data. Temperatures were measured at 48 locations shown in figure 2 and listed in table 1. After steady-state operation was achieved, these data were recorded at two minute intervals for 30 minutes and averaged over this period.

2. To measure the generator heat input, the CO_2 and O_2 content of the exiting flue gas were independently measured using a LIRA Model 300 Infrared Analyzer and a LYNN Model 6000-B Combustion Analyzer, respectively. The flue gas temperature was measured with a six-junction averaging thermocouple (location 30) and the gas usage was measured in cubic feet by a Spraque dry test meter. None of these measurements varied significantly with air temperature. Assuming the gas to be pure methane, the CO_2 and O_2 measurements indicate a theoretical amount of air ranging between 153 percent and 168 percent. A combustion analysis results in the generator heat input between 58100 and 58800 BTU/h.

3. A sample valve was installed in the line carrying the weak solution from the generator to the absorber (approximately at the location of thermocouple 3). This valve allowed the composition of the weak solution to be determined by titrating a known mass of sample with dilute sulfuric acid of known normality.

4. To install the sample valve (and two other valves intended to improve the cyclic performance of the chiller) it was necessary to remove the ammonia. The unit was recharged in a trial and error procedure until its capacity no longer was affected by small changes in the charge. The capacities measured in the steady-state tests agreed nearly identical to within 2 percent of those reported by Lindsay and Didion.[2] The evaporator and generator pressure measured in these tests, however, differ significantly from those reported by Lindsay and Didion. Pressure gauges were calibrated directly before these tests were run.

The experimental results for the steady-state tests at 80°F, 95°F, and 100°F are summarized in tables 2, 3, 4, respectively. The COP appearing in these tables is the ratio of the capacity to the generator heat input; it does not account for electrical power usage nor inefficiencies in combustion.

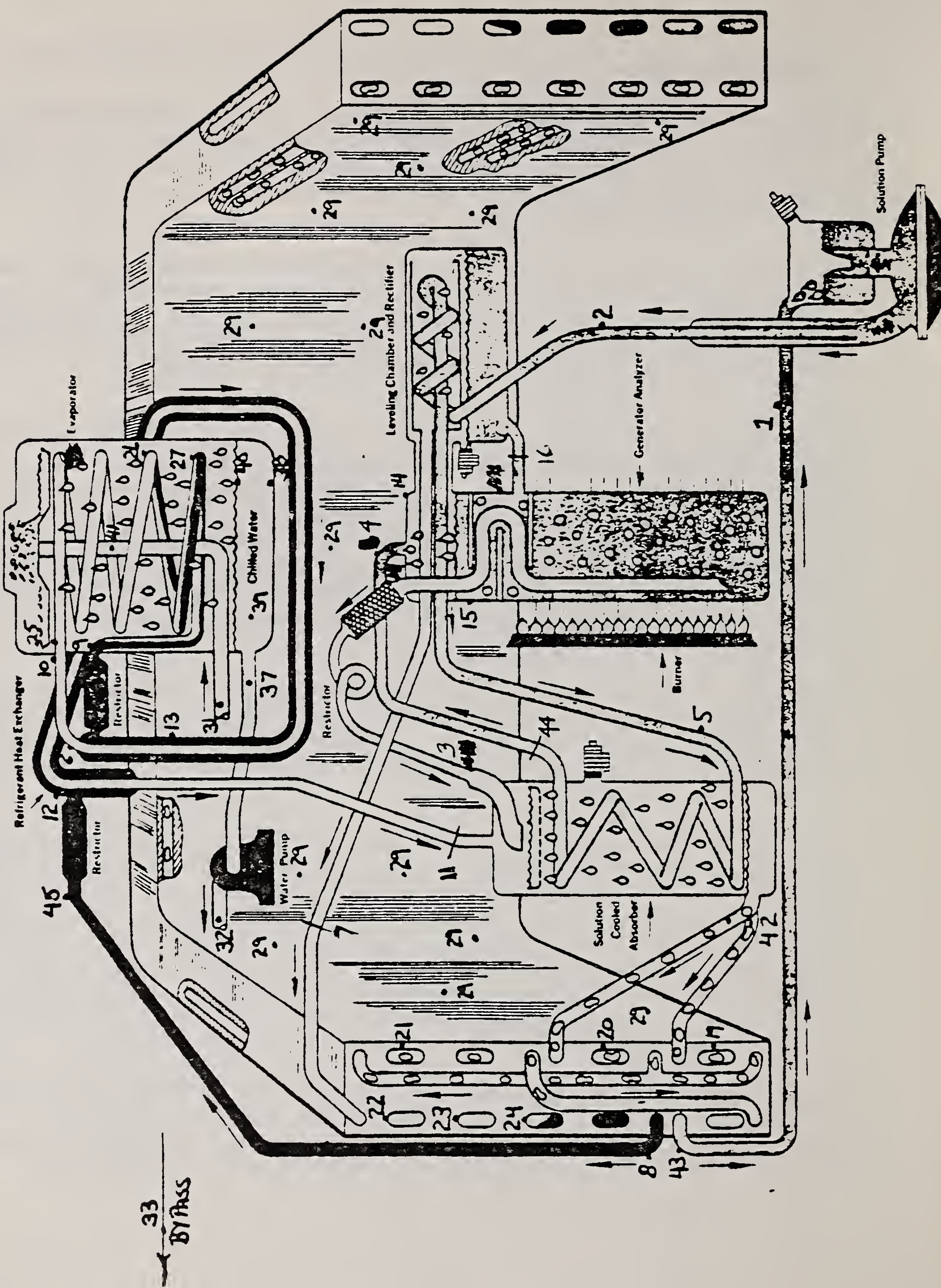

Figure 2. Thermocouple Locations.

Table 1. ARKLA Unit, Thermocouple Locations

Chanel No.	Location
0	Icebath
1	Pump in
2	Pump out
3	Solution out of Generator
4	Solution in to Generator
5	Solution into Absorber Heat Exchanger
6	Unit Plenum Air DB
7	Ammonia to Condenser
8	Ammonia out of Condenser
9	Ammonia into Evaporator
10	Ammonia out of Evaporator
11	Ammonia into Absorber
12	Liquid Ammonia into Refr. Heat Ex.
13	Liquid Ammonia out of Refr. Heat Ex.
14	Generator Top
15	Generator Middle
16	Generator Bottom
17	Solution cooled Absorber Top
18	Solution cooled Absorber Bottom
19	Air cooled Absorber Bottom
20	Air cooled Absorber Middle
21	Air cooled Absorber Top
22	Condenser Top
23	Condenser Middle
24	Condenser Bottom
25	Evaporator Top
26	Evaporator Middle
27	Evaporator Bottom
28	Air Inlet
29	Air off Condenser Coils
30	Stack Gases
31	Water Inlet
32	Water Outlet
33	Water Bypass
34	Ambient Air DB
35	Ambient Air WB
36	Temperature Difference Water Inlet/Outlet
37	Evaporator Water Discharge Line
38	Evaporator Water Bottom
39	Evaporator Water Middle
40	Evaporator Water Top
41	Evaporator Water Inlet Line
42	Solution out of Absorber
43	Solution out of Absorber Coils
44	Absorbent into Absorber
45	Refrigerant into Refr. Heat Ex.
46	Gas Supply Temperature
47	Icebath

Table 2. Steady-State Test Data at 80°F

Evaporator Pressure - 56.5 psia

Generator Pressure - 276.5 psia

Capacity - 35435 BTU/hr

CO_2% in Stack Gas - 9.0%

O_2% in Stack Gas - 7.5%

Generator Heat Input - 58100 to 58830 BTU/hr

Electrical Power Consumption - 1052 W

Weak Solution Concentration - 0.053 lbm NH_3/lbm soln.

COP - 0.602 to 0.610

Temperatures

(Thermocouple Location = Temperature °F)*

0= 32.49	1=106.16	2=104.20	3=231.54
4=220.56	5=139.08	6= 57.96	7=181.04
8= 94.92	9= 87.43	10= 69.21	11= 83.46
12= 94.62	13= 79.89	14=219.74	15=235.27
16=172.29	17=168.98	18=144.41	19=141.91
20=139.93	21=124.44	22=121.75	23=118.13
24=116.99	25= 55.18	26= 34.46	27= 33.07
28= 79.89	29=123.43	30=446.13	31= 55.11
32= 45.21	33= 57.46	34= 77.73	35= 77.93
36= 3.44	37= 44.96	38= 43.69	39= 40.98
40= 39.94	41=115.82	42=155.96	43=103.98
44=218.49	45= 96.00	46= 72.14	47= 32.64

* Channel 36 is a millivolt reading which yields the difference in temperature between the inlet and outlet chilled water when multiplied by 2.8539.

Table 3. Steady-State Test Data at 95°F

Evaporator Pressure - 324.6 psia

Generator Pressure - 71.6 psia

Capacity - 32755 BTU/hr

CO_2% in Stack Gas - 7.3%

O_2% in Stack Gas - 8.6%

Generator Heat Input - 58100 to 58830 BTU/hr

Electrical Power Consumption - 1056 W

Weak Solution Concentration - 0.087 lbm NH_3/lbm soln.

COP - 0.556 to 0.564

Temperatures

(Thermocouple Location = Temperature °F)*

0= 32.32	1=121.64	2=119.60	3=240.57
4=230.23	5=152.94	6= 51.20	7=194.00
8=119.37	9= 85.04	10= 66.69	11= 85.40
12= 93.78	13= 74.51	14=230.70	15=242.71
16=169.58	17=184.27	18=156.66	19=155.18
20=152.96	21=140.03	22=135.59	23=131.19
24=129.74	25= 53.06	26= 45.31	27= 43.19
28= 95.01	29=137.46	30=446.23	31= 54.81
32= 45.69	33= 58.36	34= 92.09	35= 92.38
36= 3.18	37= 45.36	38= 45.19	39= 44.68
40= 45.09	41=133.81	42=168.29	43=119.55
44=228.44	45= 95.49	46= 71.83	47= 32.51

* Channel 36 is a millivolt reading which yields the difference in temperature between the inlet and outlet chilled water when multiplied by 2.8539.

Table 4. Steady-State Test Data at 100.5°F

Evaporator Pressure - 340 psia

Generator Pressure - 73.5 psia

Capacity - 29050 BTU/hr

CO_2% in Stack Gas - 7.3%

O_2% in Stack Gas - 8.9%

Generator Heat Input - 58103 to 58830 BTU/hr

Electrical Power Consumption - 1068 W

Weak Solution Concentration - 0.085 lbm NH_3/lbm soln.

COP - 0.494 to 0.50

Temperatures

(Thermocouple Location = Temperature °F)*

0= 32.53	1=126.79	2=125.01	3=247.46
4=233.51	5=158.76	6= 49.21	7=199.71
8=127.70	9= 76.66	10= 62.50	11= 75.89
12= 87.03	13= 61.56	14=237.02	15=248.13
16=167.58	17=189.44	18=157.93	19=159.38
20=157.58	21=144.12	22=139.98	23=135.40
24=133.78	25= 52.36	26= 46.83	27= 44.74
28=100.46	29=141.84	30=449.32	31= 55.12
32= 46.97	33= 59.86	34= 98.09	35= 97.97
36= 2.82	37= 46.72	38= 46.53	39= 46.24
40= 46.48	41=139.23	42=170.58	43=124.55
44=231.46	45= 89.85	46= 72.80	47= 32.64

* Channel 36 is a millivolt reading which yields the difference in temperature between the inlet and outlet chilled water when multiplied by 2.8539.

4. COMPARISON OF THE MODEL TO EXPERIMENTAL DATA

Table 5 lists the parameters which must be specified to apply the steady-state model described in section 2; also listed are the values of these parameters used in the comparisons with the experimental data.

The heat transfer parameters were selected in a trial and error process by matching the calculated and experimental temperatures and capacity. A trial and error process was needed because the effects of these parameters are interrelated; a change in any one affects the temperatures at all other points in the cycle. One set of heat transer parameters was selected for all operating conditions based on the experimental results in table 3 for T_{air} = 95°F.

The calculated performance of the cycle for a 95°F air temperature and these parameters values appears in table 6. The tabular data in this table lists the location (figure 1), and corresponding temperature (T = °F), pressure (P = psia), composition (x = lb NH_3/lb soln), enthalpy (H = BTU/lb) flow rate (M = lb/hr) and quality (Q = lb vapor/lb liquid + vapor). The last two columns give the thermocouple number in figure 2 corresponding to the locatio and the experimental temperature (TOBS = °F) at this point. The calculated capacity appears in the table as QEVAP in BTU/h.

As indicated previously, the heat transfer parameters in table 5 were chosen to cause the calculated temperatures, capacity, and weak solution composition to agree with the experimental data for T_{air} = 95°F. The calculated capacity for this case is 32630 BTU/h which agrees well with the experimental value of 32755 BTU/h from table 3. The calculated weak solution composition is 0.085 lbm NH_3/lbm soln; the experimental value is 0.087 lb NH_3/lb soln. The calculated and experimental temperatures in table 6 are in reasonably good agreement. However, there are significant differences in the area of the evaporator and refrigerant heat exchanger (locations 8-11). For example, location 9 is downstream of the refrigerant restrictor and the temperature at this point should be the lowest refrigerant temperature in the cycle. The experimental value, however, is 85°F which is clearly not possible. An investigation of thermocouple 9 revealed that it is positioned at the refrigerant restrictor, rather than downstream of the restrictor as indicated in figure 2. A more reliable indication of the refrigerant temperature in the evaporator is provided by thermocouple 27, which indicated 43.2°F.

The temperature of the refrigerant exiting the evaporator (location 10) cannot be warmer than the inlet water (55°F) without violating the Second Law, yet the experimental value is 66.7°F. Thermocouple 10 was examined, calibrated with an ice bath and reinstalled with additional insulation. However, its reading did not change significantly. Perhaps the measurement is affected by conduction along the pipe.

The calculated temperature of the liquid overflow returning to the generator from the rectifier (location 13) is significantly higher than measured. It is likely, however, that the experimental value is low in this case because the overflow pipe was in an awkward location and as a result, poorly insulated.

Table 5. Parameter Values Used in Comparisons

$\dot{Q}_{gen}$ - 58460 BTU/h

$\dot{m}$ - 200 lb/h

$\dot{m}_w$ - 7.2 gpm

T_{15} - 55°F

$(UA)_{rect}$ - 80 BTU/°F-h

$(UA)_{abss}$ - 550 BTU/°F-h

$(UA)_{hx}$ - 65 BTU/°F-h

$(UA)_{evap}$ - 3000 BTU/°F-h

$(UA)_{absa}$ - 1000 BTU/°F-h

$(UA)_{cond}$ - 750 BTU/°F-h

ε_L - 0.8

ε_V - 0.9

T_{air} °F	P_{evap} psia	P_{gen} psia
80	56.5	275
95	71.6	325
100.4	73.5	340

Table 6. Calculated Results for T_{air} = 95°F

NO. OF ITERATIONS= 19
AIR TEMPERATURE= 95.0
LACK OF CLOSURE= -20.92
QGEN= 58460.0 QEVAP= 32630.1 QCOND=-41703.9 QABS=-49407.2
COOLING COP= .5582

LOC.	T	P	X	H	M	Q	TC	TOBS
1	120.3	71.6	.436	-11.3	200.0	.000	1	121.6
2	120.3	325.0	.436	-11.3	200.0	.000	2	119.6
3	150.3	325.0	.436	23.3	200.0	.000	5	152.9
4	224.9	325.0	.436	107.4	200.0	.000	4	228.4
5	240.4	325.0	.941	661.5	85.3	1.000	14	230.7
6	203.0	325.0	.978	623.0	78.6	1.000	7	194.0
7	118.7	325.0	.978	92.8	78.6	.000	8	119.4
8	83.3	325.0	.978	53.0	78.6	.000	13	71.7
9	40.6	71.6	.978	53.0	78.6	.102	9	85.0
10	47.0	71.6	.978	468.1	78.6	.854	10	66.7
11	56.1	71.6	.978	507.9	78.6	.917	11	85.4
12	180.5	71.6	.436	235.8	200.0	.266	42	168.3
13	203.0	325.0	.510	83.4	6.7	.000	16	169.6
14	255.9	325.0	.085	198.2	121.4	.000	3	240.6
15	55.0	20.0	.000	23.0	3610.0	.000	31	54.8
16	46.0	20.0	.000	14.0	3610.0	.000	32	45.7

It is possible that the agreement between the calculated and measured performance could be improved by further manipulation of the heat transfer parameters.

The parameters selected for the 95°F air temperature test were used to calculate the cycle performance at 80°F and 100°F air temperatures; the results appear in tables 7 and 8, respectively. The model predicts that capacity decreases with increasing air temperature, as observed experimentally. However, the calculated capacities are 3 percent high at 80°F and 6 percent high at 100°F compared to the experimental results in tables 2 and 4. The discrepancy results from using the parameters for the 95°F air temperature case. The agreement between the calculated and experimental temperatures is good. As for the 95°F case, significant differences occur primarily in the area of the evaporator and refrigerant heat exchanger. The calculated weak solution composition changed only slightly with changing air temperatures. The low value (0.053 lb NH_3/lb soln) observed experimentally for the 80°F test suggests that the solution flow rate is not constant with changing air temperature as assumed.

Table 7. Calculated Results for T_{air} = 80°F

NO. OF ITERATIONS= 28
AIR TEMPERATURE= 80.0
LACK OF CLOSURE= 59.77
QGEN= 58460.0 QEVAP= 36503.7 QCOND=-42277.5 QABS=-52626.4
COOLING COP= .6244

LOC.	T	P	X	H	M	Q	TC	TOBS
1	106.9	56.5	.434	-25.8	200.0	.000	1	106.0
2	106.9	275.0	.434	-25.9	200.0	.000	2	104.2
3	137.6	275.0	.434	9.0	200.0	.000	5	139.1
4	214.0	275.0	.434	97.1	200.0	.003	4	220.6
5	229.2	275.0	.943	655.3	85.1	1.000	14	219.7
6	189.7	275.0	.981	616.7	78.3	1.000	7	181.0
7	103.9	275.0	.981	77.1	78.3	.000	8	94.9
8	81.5	275.0	.981	51.7	78.3	.000	13	79.9
9	29.2	56.5	.981	51.7	78.3	.131	9	94.9
10	50.5	56.5	.981	517.9	78.3	.940	10	69.2
11	73.9	56.5	.981	543.3	78.3	.960	11	83.5
12	171.2	56.5	.434	237.7	200.0	.279	42	156.0
13	189.7	275.0	.510	68.4	6.7	.000	16	172.3
14	244.4	275.0	.082	185.7	121.7	.000	3	231.5
15	55.0	20.0	.000	23.0	3610.0	.000	31	55.1
16	44.9	20.0	.000	12.9	3610.0	.000	32	45.2

Table 8. Calculated Results for T_{air} = 100.5°F

NO. OF ITERATIONS= 22
AIR TEMPERATURE=100.5
LACK OF CLOSURE= 100.07
QGEN= 58460.0 QEVAP= 30890.0 QCOND=-41613.6 QABS=-47636.3
COOLING COP= .5284

LOC.	T	P	X	H	M	Q	TC	TOBS
1	124.7	73.5	.436	4.3	200.0	.017	1	126.8
2	124.7	340.0	.436	4.3	200.0	.000	2	125.0
3	162.2	340.0	.436	36.7	200.0	.000	5	158.8
4	226.0	340.0	.436	108.7	200.0	.000	4	233.5
5	241.9	340.0	.942	661.7	85.3	1.000	14	237.0
6	207.0	340.0	.977	625.2	79.1	1.000	7	199.7
7	124.6	340.0	.977	99.1	79.1	.000	8	127.7
8	85.5	340.0	.977	55.0	79.1	.000	13	61.6
9	41.9	73.5	.977	55.0	79.1	.103	9	76.7
10	47.0	73.5	.977	445.8	79.1	.813	10	62.5
11	52.1	73.5	.977	489.9	79.1	.888	11	75.9
12	183.4	73.5	.436	243.1	200.0	.271	42	170.6
13	207.0	340.0	.509	87.8	6.3	.000	16	167.6
14	257.8	340.0	.083	200.8	121.0	.000	3	247.5
15	55.0	20.0	.000	23.0	3610.0	.000	31	55.1
16	46.4	20.0	.000	14.4	3610.0	.000	32	47.0

5. EFFECTS OF MODEL PARAMETERS

The steady-state model described in section 2 is of limited value for interpolation or extrapolation of experimental data since it requires as input the values of the parameters listed in table 5; these parameter values must be determined from experimental data. A more detailed model would provide values for these parameters based on the physical dimensions of the components and their operating conditions. This model is, however, useful as an analysis and design tool in that the sensitivity of the heat pump performance to these design parameters can be assessed, indicating the components in the cycle for which the design is most critical.

Each of the parameters in table 5 was independently varied and the cycle performance was recalculated for the 95°F air temperature. The results of this sensitivity study are summarized in table 9 which lists the percent increase in capacity resulting from an increase in each parameter value from 5 percent below to 5 percent above its base value.

Relatively large sensitivities are observed to the inlet chilled water temperature (T_{15}) and the evaporator pressure, which affects the temperature of the evaporating refrigerant. These results indicate that, at a 95°F air temperature, the performance of the heat pump is constrained by the rate of heat transfer in the evaporator. This constraint becomes more apparent as the air temperature (and thus the evaporator pressure) is increased, causing a precipitous drop in capacity (as observed experimentally) between the 95°F and 100°F tests. It is unlikely that the experimental performance will be as sensitive to inlet water temperature as suggested in table 9 since the calculated result does not consider the effect the inlet water temperature will have on evaporator pressure.

The generator heat input appears to have only a small effect on capacity in the range of its base value of 58460 BTU/h. Both the increase and the decrease in generator heat input resulted in a slight decrease in capacity, indicating that capacity is maximized at the base value of the generator heat input. As the generator heat input is increased, the refrigerant vapor flow rate increases, but so do the temperatures of the vapor and the weak solution. Higher temperature vapor results in a higher percentage of water in the refrigerant after rectification. The higher weak solution temperature results in less effective absorption of refrigerant. These two effects counteract the beneficial effect of increased vapor flow rate. The coefficient of performance (ratio of evaporator to generator heat inputs) is reduced 9.6 percent by the increase in generator heat input.

The solution flow rate ($\dot{m}$) has a marked effect on the weak solution composition, but its effects on the refrigerant composition and the capacity are small. From a design standpoint, it appears that little improvement in performance could be expected from using a variable speed solution pump controlled as a function of operating conditions.

The capacity decreases slightly with the increase in $(UA)_{abss}$, the heat transfer conductance for the solution-cooled absorber. This non-intiutive result

Table 9. Sensitivity of the Calculated Performance to the Design Parameters (changed from 5 percent below to 5 percent above the base values)

Variable	Base Value	% Increase in Capacity
$\dot{Q}_{gen}$	58460 BTU/h	0.4
$\dot{m}$	200 lbm/h	1.8
$\dot{m}_{w}$	3610 gpm	0
T_{15}	55 °F	13.2
$(UA)_{rect}$	80 BTU/°F-h	0.7
$(UA)_{abss}$	550 BTU/°F-h	-1.1
$(UA)_{hx}$	65 BTU/°F-h	0.3
$(UA)_{evap}$	3000 BTU/°F-h	2.4
$(UA)_{absa}$	1000 BTU/°F-h	0.8
$(UA)_{cond}$	750 BTU/°F-h	0.8
ε_{L}	0.8	2.0
ε_{V}	0.9	5.3
P_{evap}	71.6 psia	-11.2
P_{gen}	325 psia	0.9

illustrates an interaction between the components. The generator is designed to allow contact between the exiting refrigerant vapor and the entering rich solution, as a result the vapor temperature approaches the rich solution temperature. This internal heat transfer improves performance since the rich solution is preheated and the water content of the refrigerant is reduced. (In the model, the temperature difference between the vapor and the rich solution is controlled by the effectiveness factor, ε_v. Capacity is relatively sensitive to this parameter.) An increase in $(UA)_{abss}$ increases the heat transfer rate in the solution-cooled absorber and thus the temperature of the rich solution. The increased heat transfer rate tends to increase the capacity since it increases the rate of absorption of refrigerant in the solution-cooled absorber. However, the higher rich solution temperature reduces the capacity since it causes the refrigerant vapor leaving the generator to be at a higher temperature. These competing effects indicate that there is an optimum value for $(UA)_{abss}$.

Only a small increase in capacity is observed to occur as a result of increases in the heat transfer parameters. Additional heat transfer surface area would not cause a significant improvement in the steady-state performance.

6. CONCLUSIONS

The usefulness of the sensitivity study in section 5 rests on the assumption that a small change in any one of the model parameters does not significantly affect any of the others. If these interaction effects can be neglected, the results of the sensitivity study reveal a number of interesting conclusions concerning the design of the ARKLA ACC-3600 chiller.

1. The unit is designed to operate at maximum capacity, rather than maximum coefficient of performance, at a 95°F air temperature.

2. There is an optimum heat exchanger surface area in the solution-cooled absorber.

3. Internal heat exchange between the streams entering and exiting the generator significantly improves the performance of the unit.

4. The rate at which the rich solution is pumped has only a small effect on the chiller performance. The rich solution at the pump inlet may be in two-phase under some operating conditions.

5. The chilled water inlet temperature has a significant affect on performance with higher temperatures resulting in increased capacity.

The absorption heat pump model presented here is simplistic in that pressure losses are not considered and heat transfer parameters must be supplied. In a more fundamental model, pressure losses and heat transfer coefficients would be calculated as a function of the physical characteristics of the system components and the operating conditions. Development of a model of this type would be a complex, but worthwhile project.

REFERENCES

1. The Secant method is a modification of the Newton-Raphson iteration method in which the derivative of the function with respect to the dependent variable is found numerically.

2. Lindsay, T. M. and Didion, D. A., "Investigation of the Cyclic Performance of a Gas-Fired Absorption Chiller," NBS internal report.

3. Jain, P. C. and Gable, G. K., "Equilibrium Property Data Equations for Aqua-Ammonia Mixtures," ASHRAE Trans., 77, 149, (1971).

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APPENDIX A

PROPERTY DATA ROUTINES

Equilibrium relationships between binary mixture properties are provided by eight FORTRAN function subprograms. A brief description of the independent variables and purpose of each subprogram is as follows:

FUNCTION TSAT(XL, P)

TSAT returns the bubble point temperature of a mixture at a total pressure P and a liquid refrigerant mass fraction XL.

FUNCTION XV(XL, P)

XV returns the mass fraction of refrigerant in the vapor phase which is in equilibrium with a liquid at total pressure P and refrigerant mass fraction XL.

FUNCTION HL (XL, T, P)

HL returns the specific enthalpy of the liquid phase at temperature T, pressure P, and refrigerant mass fraction XL.

FUNCTION HV (XL, XV, T, P)

HV returns the specific enthalpy of the vapor phase at teperature T, pressure P, and refrigerant mass fraction XV. XL is the refrigerant mass fraction of the liquid phase in equilibrium with the vapor.

FUNCTION PSAT (XL, T, PIC)

PSAT returns the bubble point pressure of a mixture at temperature T and liquid refrigerant mass fraction XL. PIC is used as the first guess in the iterative solution.

FUNCTION TH(XL, H, P, TIC)

TH returns the temperature of a liquid at pressure P, refrigerant. Mass fraction XL, and specific enthalpy H. TIC is used as the first guess.

FUNCTION XLPT (P, T, XLIC)

XLPT returns the equilibrium refrigerant mass fraction of a liquid at pressure P and temperature T. XLIC is used as the first guess in the iterative solution.

FUNCTION TQ (X, P, H, Q)

TQ returns the temperature and quality (Q) of a two-phase mixture having an overall refrigerant mass fraction x and enthalpy H at a total pressure P.

The property data for ammonia-water mixtures used in this report were provided by the polynomial equations of Jain and Gable (3). If a property was not given explicitly by these equations, it was found iteratively using the Secant method.

APPENDIX B

COMPUTER PROGRAM LISTING

```
)348*ABS(1).MAIN(4)
          DIMENSION ITHL(16),TOBS(16)
          COMMON T(20),P(20),X(20),H(20),FL(20),Q(20)
          COMMON /LU/ LUR,LUW,LUF
          COMMON /TOLS/ TOL1,TOL2,ITERMX
         COMMON /DD/ IP
          LUR=5
          LUW=6
           WRITE(LUW,2)
    2      FORMAT(1X,'ENTER FILE NUMBER')
           READ(LUR,*) LUF
          ITERMX=100
          WRITE(LUW,3)
    3     FORMAT(1X,'ENTER TOL1,TOL2,TOLP  ')
          READ(LUR,*) TOL1,TOL2,TOLP
          ICY=0
          WRITE(LUW,5)
    5     FORMAT(1X,'ENTER IP,ITPCMX  ')
          READ(LUR,*) IP,ITPCMX
    6     FORMAT(4I5)
          READ(LUF,*) TAIR
          READ(LUF,*) NP
          DO 10 I=1,NP
          READ(LUF,*) T(I),P(I),X(I),H(I),FL(I),Q(I)
   10     CONTINUE
          DO 11 I=1,NP
          READ(LUF,*) ITHL(I),TOBS(I)
   11     CONTINUE
   12     FORMAT(I5,F10.0)
          D1C=0
  111      CPC=1.0
         PEVAP=P(9)
         FLOW=FL(1)
          WRITE(LUW,78)
   78     FORMAT(1X,'ENTER UA: RECT, ABSSC, LVHX, EVAP, ABSAC, COND  ')
          READ(LUR,*) UAREC,UAABSS,UALVHX,UAEVAP,UAABSA,UACOND
          WRITE(LUW,*) UAREC,UAABSS,UALVHX,UAEVAP,UAABSA,UACOND
          WRITE(LUW,80)
   80     FORMAT(1X,'ENTER GENERATOR HEAT INPUT')
          READ(LUR,*) QGEN
          WRITE(LUW,*) QGEN
          IF (QGEN.LT.0.0) STOP
          WRITE(LUW,81)
   81     FORMAT(1X,'ENTER GENERATOR HEAT TRANSFER EFFECTIVENESS-LIQUID')
          READ(LUR,*) EFFL
          WRITE(LUW,*) EFFL
          WRITE(LUW,83)
   83     FORMAT(1X,'ENTER GENERATOR HEAT TRANSFER EFFECTIVENESS-VAPOR')
          READ(LUR,*) EFFV
```

```
      WRITE(LUW,*) EFFV
7     FORMAT(6F10.0)
C.
1000  CONTINUE
      ITERPC=0
1005  CONTINUE
      ITERPC=ITERPC+1
      IF (ITERPC.LT.ITPCMX) GO TO 1006
      CALL DIAG(ITERPC,0)
      STOP 8
1006  CONTINUE
      ICY=ICY+1
C. CYCLE CALCULATIONS - NO PRESSURE DROPS
      CALL PUMP(1,2,P(5),FLOW)
      IF (IP.LT.0) CALL DIAG(ITERPC,1)
      CALL RECT(5,2,6,3,13,UAREC)
      IF (IP.LT.0) CALL DIAG(ITERPC,2)
      CALL ABSSC(11,14,12,3,4,UAABSS)
      IF (IP.LT.0) CALL DIAG(ITERPC,3)
      CALL CONABS(12,1,TAIR,UAABSA,QABS)
      IF (IP.LT.0) CALL DIAG(ITERPC,4)
      CALL GEN(4,13,14,5,QGEN,EFFL,EFFV)
      IF (IP.LT.0) CALL DIAG(ITERPC,5)
      CALL CONABS(6,7,TAIR,UACOND,QCOND)
      IF (IP.LT.0) CALL DIAG(ITERPC,6)
      CALL HXLV(10,7,11,8,UALVHX)
      IF (IP.LT.0) CALL DIAG(ITERPC,7)
      CALL THROT(8,9,PEVAP)
      IF (IP.LT.0) CALL DIAG(ITERPC,8)
      CALL EVAP(9,10,15,16,UAEVAP,CPC,QEVAP)
      IF (IP.LT.0) CALL DIAG(ITERPC,9)
1010  CONTINUE
      D1=QGEN+QEVAP+QCOND+QABS
      IF (ABS(D1-D1C).LT.TOLP*QGEN) GO TO 1008
      D1C=D1
      GO TO 1005
1008  CONTINUE
9000  CONTINUE
C. PRINT SUMMARY
      WRITE(LUW,9010) ITERPC
9010  FORMAT(/1X,'NO. OF ITERATIONS=',I3 )
      WRITE(LUW,9015) TAIR
9015  FORMAT(1X,'AIR TEMPERATURE=',F5.1)
      WRITE(LUW,9020) D1
9020  FORMAT(1X,'LACK OF CLOSURE=',F10.2)
      WRITE(LUW,9030) QGEN,QEVAP,QCOND,QABS
9030  FORMAT(1X,'QGEN=',F8.1,'   QEVAP=',F8.1,'   QCOND=',F8.1,
     1 '   QABS=',F8.1)
      COP=QEVAP/QGEN
      WRITE(LUW,9040) COP
9040  FORMAT(1X,'COOLING COP=',F8.4/)
      WRITE(LUW,9050)
9050  FORMAT(3X,'LOC.',3X,'T',7X,'P',8X,'X',6X,'H',7X,'M',8X,'Q',
     1 4X,'TC',4X,'TOBS')
      DO 9100 I=1,NP
```

```
9100    WRITE(LUW,9110) I,T(I),P(I),X(I),H(I),FL(I),Q(I);ITHL(I);TOBS(I)
9110    FORMAT(1X,I4,2F8.1,F8.3,2F8.1,F8.3,I5,F8.1)
99      CONTINUE
        WRITE(LUW,59)
59     FORMAT(/)
        D1C=1000000.
        GO TO 111
        END
```

0348*ABS(1).PUMP(0)

```
        SUBROUTINE PUMP(IL,JL,PHIGH,FLOW)
        COMMON T(20),P(20),X(20),H(20),FL(20),Q(20)
C. IL - LIQUID FROM AIR-COOLED ABSORBER
C. JL - LIQUID TO RECTIFIER HX
        T(JL)=T(IL)
        P(JL)=PHIGH
        FL(JL)=FLOW
        H(JL)=H(IL)
        X(JL)=X(IL)
        RETURN
        END
```

0348*ABS(1).RECT(6)

```
        SUBROUTINE RECT(IV,IL1,JV,JL1,JL2,UA)
        COMMON T(20),P(20),X(20),H(20),FL(20),Q(20)
        COMMON /TOLS/ TOL1,TOL2,ITERMX
        COMMON /DD/ ICK
        COMMON /LU/ LUR,LUW,LUF
C. IV - VAPOR FROM GENERATOR
C. IL1 - LIQUID FROM SOLUTION PUMP
C. JV - VAPOR TO CONDENSOR
C. JL1 - LIQUID TO ABSORBER HX
C. JL2 - LIQUID RETURN TO GENERATOR
C.
        P(JV)=P(IV)
        P(JL1)=P(IL1)
        FL(JL1)=FL(IL1)
        X(JL1)=X(IL1)
        ITER=0
       T1=(T(IV)+T(IL1))/2.
        TT=T1
        ASSIGN 100 TO IJP
        GO TO 800
```

```
100     F1=F
        T2=T1+2.
110     TT=T2
        ASSIGN 120 TO IJP
        GO TO 800
120     F2=F
        ITER=ITER+1
        IF (ITER.LT.ITERMX) GO TO 150
        WRITE(LUW,130) T(JL1),QX,DTLM,H(JL1),X(JV)
130     FORMAT(1X,'***RECT***',6F8.3)
        STOP
150     CONTINUE
        T(JV)=T2-F2/(F2-F1)*(T2-T1)
        IF (ABS(T(JV)-T2).LT.TOL1) RETURN
        F1=F2
        T1=T2
        T2=T(JV)
        GO TO 110
800     CONTINUE
        T(JL2)=TT
        X(JL2)=XLPT(P(JL2),T(JL2),X(JL2))
        X(JV)=XV(X(JL2),P(JV))
        FL(JV)=FL(IV)*(X(IV)-X(JL2))/(X(JV)-X(JL2))
        FL(JL2)=FL(IV)-FL(JV)
        H(JV)=HV(0.9,X(JV),TT,P(JV))
        H(JL2)=HL(X(JL2),T(JL2),P(JL2))
        QX=FL(JV)*H(JV)+FL(JL2)*H(JL2)-FL(IV)*H(IV)
        H(JL1)=H(IL1)-QX/FL(IL1)
        TM=TSAT(X(JL1),P(JL1))
        T(JL1)=TH(X(JL1),H(JL1),P(JL1),150.)
        Q(JL1)=0.
        IF (T(JL1).GT.TM) T(JL1)=TQ(X(JL1),P(JL1),H(JL1),Q(JL1))
        IF (ICK.EQ.7) WRITE(LUW,7) TT,T(IV),T(JL1),T(IL1),QX
 7      FORMAT(1X,5F10.3)
       ARG=(T(IV)-T(JL1))/(TT-T(IL1))
       IF (ARG.GT.0.0) GO TO 810
       DTLM=0.
       GO TO 820
810    CONTINUE
       DTLM=(T(IV)-T(JL1)-TT+T(IL1))/ALOG(ARG)
820    CONTINUE
        QXC=-UA*DTLM
        F=QX-QXC
        GO TO IJP
        END
```

```
0348*ABS(1).ABSSC(9)
        SUBROUTINE ABSSC(IV,IL1,JL1,IL2,JL2,UA)
        COMMON T(20),P(20),X(20),H(20),FL(20),Q(20)
```

```
            COMMON /TOLS/ TOL1,TOL2,ITERMX
            COMMON /DD/ICK
           COMMON /LU/ LUR,LUW,LUF
C. IV - VAPOR FROM EVAPORATOR HX
C. IL1 - LIQUID FROM GENERATOR
C. JL1 - LIQUID TO AIR-COOLED ABSORBER
C. IL2 - LIQUID FROM RECTIFIER
C. JL2 - LIQUID TO GENERATOR
C.
        X(JL2)=X(IL2)
        P(JL2)=P(IL2)
        FL(JL2)=FL(IL2)
        FL(JL1)=FL(IV)+FL(IL1)
        X(JL1)=(FL(IV)*X(IV)+FL(IL1)*X(IL1))/FL(JL1)
        P(JL1)=P(IV)
        ITER=0
        T1=T(IL2)+40.
        TT=T1
        ASSIGN 100 TO IJP
        GO TO 800
100     F1=F
        ASSIGN 120 TO IJP
        T2=T1+5.
110     TT=T2
        GO TO 800
120     F2=F
        ITER=ITER+1
        IF (ITER.LT.ITERMX) GO TO 150
        WRITE(LUW,130) T(JL2),QX,DTLM,XLIQ,XVAP,HLIQ,HVAP
130     FORMAT(1X,'***ABSSC***',8F8.3)
        RETURN
150     CONTINUE
        T(JL2)=T2-F2/(F2-F1)*(T2-T1)
        IF (ABS(T(JL2)-T2).LT.TOL1) RETURN
        F1=F2
        T1=T2
        T2=T(JL2)
        GO TO 110
800     CONTINUE
        TM=TSAT(X(JL2),P(JL2))
        IF (T(JL2).GT.TM) GO TO 820
810     CONTINUE
C. SUBCOOLED LIQUID
        H(JL2)=HL(X(JL2),TT,P(JL2))
        Q(JL2)=0.
        GO TO 830
820     CONTINUE
C. TWO-PHASE MIXTURE
        XLIQ=XLPT(P(JL2),TT,0.4)
        XVAP=XV(XLIQ,P(JL2))
        Q(JL2)=(X(JL2)-XLIQ)/(XVAP-XLIQ)
        HLIQ=HL(XLIQ,TT,P(JL2))
        HVAP=HV(XLIQ,XVAP,TT,P(JL2))
        H(JL2)=(1.0-Q(JL2))*HLIQ+Q(JL2)*HVAP
830     CONTINUE
```

```
      QX=FL(JL2)*(H(JL2)-H(IL2))
      H(JL1)=(FL(IV)*H(IV)+FL(IL1)*H(IL1)-QX)/FL(JL1)
      T(JL1)=TQ(X(JL1),P(JL1),H(JL1),Q(JL1))
      IF (ICK.EQ.9) WRITE(LUW,*) T(IL1),TT,T(JL1),T(IL2),QX
      ARG=(T(IL1)-TT)/(T(JL1)-T(IL2))
      IF (ARG.GT.0.) GO TO 840
      DTLM=0.
      GO TO 845
840   CONTINUE
      DTLM=(T(IL1)-TT-T(JL1)+T(IL2))/ALOG(ARG)
845   CONTINUE
      QXC=UA*DTLM
      F=QX-QXC
      GO TO IJP
      END
```

0348*ABS(1).GEN(6)

```
      SUBROUTINE GEN(IL1,IL2,JL1,JV,QGEN,EFFL,EFFV)
      COMMON T(20),P(20),X(20),H(20),FL(20),Q(20)
      COMMON /TOLS/ TOL1,TOL2,ITERMX
      COMMON /DD/ ICK
      COMMON /LU/ LUR,LUW,LUF
C. IL1 - LIQUID FROM ABSORBER
C. IL2 - LIQUID FROM RECTIFIER
C. JL1 - LIQUID TO SOLUTION-COOLED ABSORBER
C. JV - VAPOR TO RECTIFIER
      TGEN1=T(JV)
      P(JL1)=P(IL1)
      ITERG=0
      THOLD=TGEN1
      ASSIGN 10 TO IJP1
      GO TO 1000
10    F1=F
      TGEN2=TGEN1+5.
20    THOLD=TGEN2
      ITERG=ITERG+1
      IF (ITERG.LT.ITERMX) GO TO 25
      WRITE(LUW,24) TGEN,P(JL1),X(JL1),H(JL1),FL(JL1)
24    FORMAT(1X,'***GEN***',F10.3)
      RETURN
25    CONTINUE
      ASSIGN 30 TO IJP1
      GO TO 1000
30    F2=F
      TGEN=TGEN2-F2/(F2-F1)*(TGEN2-TGEN1)
      IF (ABS(TGEN-TGEN2).LT.TOL1) GO TO 60
      TGEN1=TGEN2
      F1=F2
      TGEN2=TGEN
```

```
            GO TO 20
      60    CONTINUE
            T(JL1)=TGEN-EFFL*(TGEN-T(IL1))
            RETURN
      1000  CONTINUE
            T(JL1)=THOLD-EFFL*(THOLD-T(IL1))
      C. EFFL IS A HEAT EXCHANGE EFFECTIVENESS BETWEEN THE EXITING WEAK
      C.  SOLUITION AND THE INCOMING STRONG SOLUTION
            X(JL1)=XLPT(P(JL1),THOLD,0.35)
            H(JL1)=HL(X(JL1),T(JL1),P(JL1))
      C. NOTE THAT LIQUID IS SUBCOOLED BEFORE LEAVING GENERATOR IF EFFL>0
      C.  EFFV IS A HEAT TRANSFER EFFECTIVENESS BETWEEN THE ENTERING LIQUID AND
      C.  THE EXITING VAPOR AT THE TOP OF THE GENERATOR
            T(JV)=THOLD-EFFV*(THOLD-T(IL1))
            P(JV)=P(JL1)
            XLTOP=XLPT(P(JV),T(JV),0.40)
            IF (ICK.EQ.10) WRITE(LUW,*) XLTOP,P(JV),T(JV),THOLD,X(JL1)
          1 ,T(JL1)
            X(JV)=XV(XLTOP,P(JV))
            FL(JL1)=(FL(IL1)*X(IL1)+FL(IL2)*X(IL2)-(FL(IL1)+FL(IL2))*X(JV))
          1 /(X(JL1)-X(JV))
            FL(JV)=FL(IL1)+FL(IL2)-FL(JL1)
            H(JV)=HV(XLTOP,X(JV),T(JV),P(JV))
            F=FL(IL1)*H(IL1)+FL(IL2)*H(IL2)-FL(JV)*H(JV)-FL(JL1)*H(JL1)+QGEN
            GO TO IJP1
            END
```

0348*ABS(1).CONABS(4)

```
            SUBROUTINE CONABS(IV,JL,TAIR,UA,QCOND)
            COMMON T(20),P(20),X(20),H(20),FL(20),Q(20)
            COMMON /TOLS/ TOL1,TOL2,ITERMX
            COMMON /LU/ LUR,LUW,LUF
            COMMON /DD/ ICK
      C. IV - VAPOR FROM RECTIFIER
      C. JL - LIQUID TO EVAPORATOR HX
            FL(JL)=FL(IV)
            P(JL)=P(IV)
            X(JL)=X(IV)
            ITER=0
            T1=TAIR+20.
            TT=T1
            ASSIGN 100 TO IJP
            GO TO 800
      100   F1=F
            T2=T1+5.
      110   TT=T2
            ASSIGN 200 TO IJP
            GO TO 800
      200   F2=F
```

```
      ITER=ITER+1
      IF (ITER.LT.ITERMX) GO TO 210
      WRITE(LUW,230) TT,H(JL),QCOND,T(IV),X(JL)
230   FORMAT(1X,'***COND***',5F10.3)
      STOP 230
210   CONTINUE
      T(JL)=T2-F2/(F2-F1)*(T2-T1)
      IF (ABS(T(JL)-T2).LT.TOL1) RETURN
      F1=F2
      T1=T2
      T2=T(JL)
      GO TO 110
800   CONTINUE
      ARG=(T(IV)-TAIR)/(TT-TAIR)
      IF (ARG.GT.0.0) GO TO 810
      DTLM=0.
      GO TO 820
810   CONTINUE
      DTLM=(T(IV)-TT)/ALOG(ARG)
820    CONTINUE
      QCOND=-UA*DTLM
      TMAX=TSAT(X(JL),P(JL))
      IF (TT.GT.TMAX) GO TO 850
C. SUBCOOLDED AT OUTLET
      Q(JL)=0.
      H(JL)=HL(X(JL),TT,P(JL))
      GO TO 880
850   CONTINUE
C. TWO-PHASE AT OUTLET
      XLIQ=XLPT(P(JL),TT,0.8)
      XVAP=XV(XLIQ,P(JL))
      HLIQ=HL(XLIQ,TT,P(JL))
      HVAP=HV(XLIQ,XVAP,TT,P(JL))
      Q(JL)=(X(JL)-XLIQ)/(XVAP-XLIQ)
      H(JL)=(1.-Q(JL))*HLIQ+Q(JL)*HVAP
880   CONTINUE
      QCONDC=(H(JL)-H(IV))*FL(JL)
      F=QCOND-QCONDC
      IF (ICK.EQ.6) WRITE(LUW,6) TT,TMAX,XLIQ,XVAP,HLIQ,HVAP,H(JL)
     1 ,Q(JL),T(IV)
6        FORMAT(1X,8F9.3)
      GO TO IJP
      END
```

0348*ABS(1).EVAP(6)

```
      SUBROUTINE EVAP(IL,JV,IW,JW,UA,CPC,QEVAP)
      COMMON T(20),P(20),X(20),H(20),FL(20),Q(20)
      COMMON /DD/ ICK
      COMMON /TOLS/ TOL1,TOL2,ITERMX
```

```
      COMMON /LU/ LUR,LUW,LUF
C. IL - LIQUID FROM THROTTLE
C. JV - VAPOR TO HEAT EXCHANGER
C. IW - COOLANT WATER INLET
C. JW - CHILLED WATER OUTLET
C.
      X(JV)=X(IL)
      FL(JV)=FL(IL)
      P(JV)=P(IL)
      P(JW)=P(IW)
      FL(JW)=FL(IW)
      X(JW)=0.
      TMAX=T(IW)-0.0001
      T1=TMAX
      TT=T1
      ASSIGN 110 TO IJP
      GO TO 800
110   F1=F
      ASSIGN 120 TO IJP
      T2=T1-2.
      ITER=0
115   TT=T2
      GO TO 800
120     F2=F
      ITER=ITER+1
      IF (ITER.LT.ITERMX) GO TO 5
      WRITE(LUW,3) T(JV),DTLM,QEVAP,H(JV),Q(JV)
      RETURN
 5     CONTINUE
      IF (ICK.EQ.4) WRITE(LUW,*) TT,T(IL),T(IW),FL(IL),Q(JV),DTLM,QEVAPC
      T(JV)=T2-F2/(F2-F1)*(T2-T1)
     IF (ABS(T(JV)-T2).LT.TOL1) RETURN
     T1=T2
     F1=F2
     T2=T(JV)
     GO TO 115
800    CONTINUE
      DTLM=(TT-T(IL))/ALOG((T(IW)-T(IL))/(T(IW)-TT))
      QEVAPC=UA*DTLM
      XLIQ=XLPT(P(JV),TT,0.8)
      XVAP=XV(XLIQ,P(JV))
      Q(JV)=(X(JV)-XLIQ)/(XVAP-XLIQ)
      HLIQ=HL(XLIQ,TT,P(JV))
      HVAP=HV(XLIQ,XVAP,TT,P(JV))
      H(JV)=(1.-Q(JV))*HLIQ+Q(JV)*HVAP
      QEVAP=(H(JV)-H(IL))*FL(IL)
      F=QEVAP-QEVAPC
      IF (ICK.EQ.4) WRITE(LUW,3) TT,DTLM,QEVAP,H(JV),Q(JV)
3     FORMAT(1X,6F10.2)
     H(JW)=H(IW)-QEVAPC/FL(IW)
     T(JW)=T(IW)+(H(JW)-H(IW))/CPC
      GO TO IJP
      END
```

```
0348*ABS(1).HXLV(7)
              SUBROUTINE HXLV(IV,IL,JV,JL,UA)
              COMMON T(20),P(20),X(20),H(20),FL(20),Q(20)
              COMMON /TOLS/ TOL1,TOL2,ITERMX
              COMMON /DD/ ICK
              COMMON /LU/ LUR,LUW,LUF
      C. IV - VAPOR FROM EVAPORATOR
      C. JV - VAPOR TO ABSORBER
      C. IL - LIQUID FROM CONDENSOR
      C. JL - LIQUID TO EVAPORATOR
      C.
              FL(JV)=FL(IL)
              FL(IV)=FL(IL)
              FL(JL)=FL(IL)
              X(JV)=X(IV)
              X(JL)=X(IL)
              P(JV)=P(IV)
              P(JL)=P(IL)
              ITER=0
              T1=T(IL)-15.
              TT=T1
              ASSIGN 100 TO IJP
              GO TO 800
      100     F1=F
              T2=T1+5.
              ASSIGN 120 TO IJP
      110     TT=T2
              GO TO 800
      120     F2=F
              ITER=ITER+1
              IF (ITER.LT.ITERMX) GO TO 150
              WRITE(LUW,130) T(JL),QX,DTLM,QX1,H(JL),H(JV)
      130     FORMAT(1X,'***HXLV***',6F8.3)
              STOP
      150     CONTINUE
              T(JL)=T2-F2/(F2-F1)*(T2-T1)
              IF (ABS(T(JL)-T2).LT.TOL1) RETURN
              F1=F2
              T1=T2
              T2=T(JL)
              GO TO 110
      800     CONTINUE
              Q(JL)=0.
              H(JL)=HL(X(JL),TT,P(JL))
              QX=(H(IL)-H(JL))*FL(IL)
              H(JV)=H(IV)+QX/FL(IV)
              T(JV)=TQ(X(JV),P(JV),H(JV),Q(JV))
             IF (T(JV).GE.T(IL)) T(JV)=T(IL)-0.1
              ARG=(TT-T(IV))/(T(IL)-T(JV))
             IF (ICK.EQ.11) WRITE(LUW,*) TT,T(IV),T(IL),T(JV),QX,Q(JV)
```

```
      DTLM=(TT-T(IV)-T(IL)+T(JV))/ALOG(ARG)
      QX1=UA*DTLM
      F=QX-QX1
      GO TO IJP
      END
```

0348*ABS(1).THROT(0)

```
      SUBROUTINE THROT(IL,JL,PSAT)
      COMMON T(20),P(20),X(20),H(20),FL(20),Q(20)
      P(JL)=PSAT
      ITERT=0
      XF=X(IL)
      H(JL)=H(IL)
      X(JL)=X(IL)
      FL(JL)=FL(IL)
      T(JL)=TQ(X(JL),P(JL),H(JL),Q(JL))
      RETURN
      END
```

END PRT

0348*ABS(1).DIAG(1)

```
      SUBROUTINE DIAG(ITERPC,LOC)
      COMMON T(20),P(20),X(20),H(20),FL(20),Q(20)
      COMMON /LU/ LUR,LUW,LUF
      WRITE(LUW,23) ITERPC,LOC
23    FORMAT(1X,' ITERATION=',I5,'   LOCATION=',I5)
      DO 24 I=1,16
24    WRITE(LUW,9110) I,T(I),P(I),X(I),H(I),FL(I),Q(I)
9110  FORMAT(1X,I5,6F10.3)
      RETURN
      END
```

0348*ABS(1).TSAT(0)

```
      FUNCTION TSAT(XL,P)
C.  THESE CURVE FITS ARE TAKEN FROM:
C.  EQUILIBRIUM PROPERTY DATA EQUATIONS FOR AQUA-AMMONIA MIXTURES
C.  P.C. JAIN AND G.K.GABLE   ASHRAE (1971)
```

```
      C.  ENGLISH UNITS: T=F, X=LB/LB SOLN, P=PSIA
          IF (P.LT.200.) GO TO 100
          TSAT=((((-240.11*XL+346.31)*XL-27.120)*XL+166.94)*XL-535.76)*XL
         1 +(0.038839-0.18053E-03*P)*XL*P+305.04+(0.44631-
         2 0.24284E-03*P)*P
          RETURN
      100 CONTINUE
          TSAT=((((-692.82*XL+1673.3)*XL-1424.98)*XL+787.79)*XL-584.78)*XL
         1 +(-0.34428+0.00011334*P)*P*XL+203.80+(1.8362-0.0060111*P)*P
          RETURN
          END
```

```
0348*ABS(1).XV(1)
          FUNCTION XV(XL,P)
          IF (XL.GT.0.9999) XL=0.9999
          IF (P.LT.200.) GO TO 100
          R=((((10.749*XL-17.869)*XL+4.0279)*XL-1.3086)*XL+2.5622E-03
         1 *P*XL)*XL-4.256E-03*P+7.1588
          GO TO 200
      100 CONTINUE
          R=((((108.485*XL-229.009)*XL+155.247)*XL-41.0442)*XL)*XL
         1 +11.2925-0.031256*P+0.0213337*P*XL**2
      200 XV=1.-(1.-XL)**R
          RETURN
          END
```

```
0348*ABS(1).HL(0)
          FUNCTION HL(XL,T,P)
          IF (P.LT.200.) GO TO 100
          HL=((((561.86*XL-1929.6)*XL+2343.3)*XL-828.41)*XL-103.48)*XL
         1 -76.824+1.12703*T
          RETURN
      100 CONTINUE
          HL=(((-656.458*XL+1358.93)*XL-498.318)*XL-182.534)*XL
         1 -57.1775+1.09174*T
          RETURN
          END
```

```
0348*ABS(1).HV(0)
      FUNCTION HV(XL,XV,T,P)
      IF (P.LT.200.) GO TO 100
      XVT=ALOG(0.00004)
      IF (XV.LE.0.99996) XVT=ALOG(1.-XV)
      IF (XL.LT.0.36) GO TO 50
      HV=(((0.068765*XVT+2.0794)*XVT+24.839)*XVT+144.63)*XVT+911.73
     1 +8.370E-09*T**4+(((-3.7752E-05*T+0.027252)*T-5.9429)
     2 *T)*(1.-XV)+0.54663*T*(1.-XV)**2-3.1313
      RETURN
   50 CONTINUE
      HV=(((-1342.65*XV+2954.4)*XV-2485.3)*XV+391.37)*XV+1080.9
     1 +8.370E-09*T**4+(((-3.7752E-05*T+0.027252)*T-
     2 5.9429)*T)*(1.-XV)+0.54663*T*(1.-XV)**2-3.1313
      RETURN
  100 CONTINUE
      HV=((-4.948E-06*T+1.49518E-03)*T+0.415871)*T+530.976
     1 +(((4.05554E-05*T-0.0290022)*T+6.79126)*T)*(1.-XV)
      RETURN
      END
```

```
0348*ABS(1).PSAT(1)
      FUNCTION PSAT(XL,T,P1C)
      COMMON /TOLS/ TOL1,TOL2,ITERMX
      COMMON /LU/ LUR,LUW,LUF
      ITER=0
      P1=P1C
      F1=T-TSAT(XL,P1)
      P2=P1+1.
   10 CONTINUE
      ITER=ITER+1
      IF (ITER.LT.ITERMX) GO TO 15
      WRITE(LUW,16) XL,T,PSAT,ITER
   16 FORMAT(/1X,'***PSAT*** ',3F10.3,I5)
      RETURN
   15 CONTINUE
      F2=T-TSAT(XL,P2)
      PSAT=P2-F2/(F2-F1)*(P2-P1)
      IF (ABS(PSAT-P2).LT.TOL1) RETURN
      P1=P2
      F1=F2
      P2=PSAT
      GO TO 10
      END
```

0348*ABS(1).TH(1)

```
      FUNCTION TH(XL,H,P,T1C)
      COMMON /TOLS/ TOL1,TOL2,ITERMX
      COMMON /LU/ LUR,LUW,LUF
      ITER=0
      T1=T1C
      F1=H-HL(XL,T1,P)
      T2=T1+2.
   10 CONTINUE
      ITER=ITER+1
      IF (ITER.LT.ITERMX) GO TO 15
      WRITE(LUW,16) XL,H,TH,P
   16 FORMAT(/1X,'***TH*** ',4F10.3)
      RETURN
   15 CONTINUE
      F2=H-HL(XL,T2,P)
      TH=T2-F2/(F2-F1)*(T2-T1)
      IF (ABS(TH-T2).LT.0.01) RETURN
      T1=T2
      F1=F2
      T2=TH
      GO TO 10
      END
```

0348*ABS(1).XLPT(0)

```
      FUNCTION XLPT(P,T,XL1C)
      COMMON /TOLS/ TOL1,TOL2,ITERMX
      COMMON /LU/ LUR,LUW,LUF
      ITER=0
      XL1=XL1C
      F1=T-TSAT(XL1,P)
      XL2=XL1+0.01
   10 CONTINUE
      ITER=ITER+1
      IF (ITER.LT.ITERMX) GO TO 15
      WRITE(LUW,16) P,T,XLPT
   16 FORMAT(/1X,'***XLPT***  ',3F10.3)
      RETURN
   15 CONTINUE
      F2=T-TSAT(XL2,P)
      XLPT=XL2-F2/(F2-F1)*(XL2-XL1)
      IF (ABS(XLPT-XL2).LT.TOL2) RETURN
      XL1=XL2
      F1=F2
      XL2=XLPT
      GO TO 10
```

```
      END
```

0348*ABS(1).THV(1)

```
      FUNCTION THV(XL,XV,H,P,T1C)
      COMMON /TOLS/ TOL1,TOL2,ITERMX
      COMMON /LU/ LUR,LUW,LUF
      ITER=0
      T1=T1C
      F1=H-HV(XL,XV,T1,P)
      T2=T1+2.
   10 CONTINUE
      ITER=ITER+1
      IF (ITER.LT.ITERMX) GO TO 15
      WRITE(LUW,16) XL,XV,H,P,THV
   16 FORMAT(1X,'***TH***  ',5F9.3)
      RETURN
   15 CONTINUE
      F2=H-HV(XL,XV,T2,P)
      THV=T2-F2/(F2-F1)*(T2-T1)
      IF (ABS(THV-T2).LT.TOL1) RETURN
      T1=T2
      F1=F2
      T2=THV
      GO TO 10
      END
```

0348*ABS(1).TQ(0)

```
      FUNCTION TQ(X,P,H,Q)
      COMMON /TOLS/ TOL1,TOL2,ITERMX
      COMMON /LU/ LUR,LUW,LUF
      COMMON /DD/ ICK
      ITERT=0
      T1=TSAT(X,P)
      TT=T1
      ASSIGN 670 TO IJPT
      GO TO 800
  670 CONTINUE
      FT1=FT
      T2=T1+2.
      ASSIGN 675 TO IJPT
  673 TT=T2
      GO TO 800
  675 FT2=FT
```

```
            ITERT=ITERT+1
            IF (ITERT.LT.ITERMX) GO TO 680
            WRITE(LUW,681) TT,P,XF,XG,HF,HG,Q,H
    681     FORMAT(1X,'***TQ***',8F7.2)
            RETURN
    680     CONTINUE
            IF (ICK.EQ.5) WRITE(LUW,681) TT,P,XF,XG,HF,HG,Q,H
            TT=T2-FT2/(FT2-FT1)*(T2-T1)
            IF (ABS(TT-T2).LT.TOL1) GO TO 690
            T1=T2
            FT1=FT2
            T2=TT
            GO TO 673
    690     CONTINUE
            TQ=TT
            RETURN
    800     CONTINUE
            XF=XLPT(P,TT,X)
            IF (XF.GT.0.9999) XF=0.9999
            HF=HL(XF,TT,P)
            XG=XV(XF,P)
            Q=(X-XF)/(XG-XF)
            HG=HV(XF,XG,TT,P)
            FT=Q*HG+(1.-Q)*HF-H
            GO TO IJPT
            END
END PRT
>
```

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10. SUPPLEMENTARY NOTES

☐ Document describes a computer program; SF-185, FIPS Software Summary, is attached.

11. ABSTRACT *(A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here)*

A mathematical model of the steady-state performance of an absorption heat pump is described. The model is compared with experimental data from a residential-sized water chiller. It is also used to determine the sensitivity of the heat pump performance to its design variables.

12. KEY WORDS *(Six to twelve entries; alphabetical order; capitalize only proper names; and separate key words by semicolons)*

absorption heat pump; ammonia-water; ARKLA water chiller; experimental performance; mathematical model; steady-state performance

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