Simulation Model of a Single-Stage Lithium Bromide-Water Absorption Cooling Unit

David Miao

AUGUST 1978
Simulation Model of
a Single-Stage Lithium Bromide -
Water Absorption Cooling Unit

David Miao
Lewis Research Center
Cleveland, Ohio
SIMULATION MODEL OF A SINGLE-STAGE LITHIUM BROMIDE - WATER ABSORPTION COOLING UNIT

by David Miao

Lewis Research Center

SUMMARY

The performance and load capability of a given LiBr-H$_2$O absorption chiller operating with a hot-water heat source depends on six quantities: the inlet temperatures and flow rates of the hot-water source, the cooling-tower water, and the return chiller water. Based on this, a computer model for a single-stage absorption cooling machine has been developed which does not require data relative to the interior characteristics of the machine (heat-transfer rates and surfaces). The model considers both heat-transfer and thermodynamic processes. It consists of two algorithms, one for the design, or reference conditions, and the other for the off-design analysis. It is constructed from the steady-state equations but may also be used for the transient analysis of a cooling system.

The program can be used in an independent mode or as a subroutine, as for example, with TRNSYS'S, for the analysis of a cooling system. For a given size of machine the model can be used to predict off-design cooling-system performance, the only input requirements being a set of reference or rated conditions for the machine.

INTRODUCTION

The LiBr-H$_2$O absorption liquid chiller has been used in the refrigeration and air-conditioning industry for some time. One of the primary reasons for using this type of machine is that steam or hot water, whichever is available, can be directly used as an energy source to power the machine. This characteristic is particularly attractive for solar-cooling applications.

In a typical solar-cooling application, water heated through the passages of a bank of solar energy collectors is used to power an absorption machine to provide chilled water which in turn is used to air condition the building.
Typically an absorption chiller is designed to handle the maximum expected load of the building. The design point thus represents a set of fixed operating conditions. However, the actual load varies with building heat-transfer characteristics as well as local weather conditions. The design load may seldom be experienced. Since the chilled-water temperature is likely to increase with decreasing heat load (part-load operation), the chiller may be incapable of dehumidification.

An approach is, prior to machine selection, to simulate various loading conditions through a computer model of the machine. The typical models available today are either empirical (ref. 1), or based upon a thermodynamic approach. The former generally represents a specific machine, and therefore its usefulness is limited; the latter is useful for providing a set of design conditions to the machine manufacturer to determine the size of an absorption machine.

A thermodynamic approach can be used for simulating various operating conditions but such a model does not recognize limitations of the heat-transfer processes. A better approach is to take both heat-transfer and thermodynamic processes into consideration. Furthermore, if an existing machine is selected for a specific job, the heat-transfer surfaces in the machine are fixed but often not known, and therefore it will be difficult to determine the capability of the machine over a range of operating conditions. The only known inputs are three sets of inlet flows and temperatures to the machine: namely, the flow rates and corresponding temperatures of the return chilled water, the cooling water, and the incoming hot water. The unknowns required to be established are the corresponding outlet temperatures of the three flow streams. A computer model for handling this type of problem is not generally available. The purpose of this report is to document a method for modeling the system.

THERMODYNAMIC CYCLE

The thermodynamic cycle of the absorption machine is well known (refs. 2 to 4). Figure 1 represents a typical arrangement of a single-stage machine. The machine basically consists of five heat exchangers called a generator, a condenser, an evaporator, an absorber, and a solution heat exchanger. For a heat load imposed on the evaporator E, the LiBr-H₂O strong solution is pumped through the solution heat exchanger X to the generator G. Heat energy is added in the generator G to drive out the refrigerant (in this case, water is the refrigerant). The remaining solution is called the weak solution. A portion of the weak solution is forced through the solution exchanger X and a pressure reducing valve V1 back to the absorber A for the next cycle. To make a strong solution in the absorber A, the refrigerant leaving the generator G must also be brought back to the absorber through a condenser-evaporator path. In the process, the refrigerant is first condensed by removing its latent heat in the condenser C; then in passing
through an expansion valve \( V2 \), the pressure and the temperature of the refrigerant are reduced. The refrigerant is evaporated due to heat load addition in the evaporator \( E \). The refrigerant vapor is then brought to the absorber \( A \) to be absorbed. When the heat of absorption is removed, the strong solution is restored and the new cycle begins.

Most commercial machines are built on this basis. To simulate machine performance, a thermodynamic cycle analysis is used to perform heat balance calculations in order to establish heat input and cooling requirement for a refrigeration load. Heat inputs and outputs of the machine are marked in figure 1. The solid arrow lines indicate the direction of heat flow as well as fluid flow interior to the machine while the dashed lines indicate heat or fluid flowing into or out of the machine. Figure 1 is used to construct the thermodynamic portion of the machine model.

To perform such calculations, thermodynamic properties of the water and the LiBr-H\(_2\)O solution are also needed. Such information is readily found in reference 2 (the formulas used may be found in appendix A).

HEAT-TRANSFER CONSIDERATIONS

The thermodynamic analysis determines the cycle temperatures and the required heat flows for the five heat exchangers in the absorption machine as shown in figure 1. For a given refrigeration load, the heat exchangers must be designed to satisfy the aforementioned requirements. Once the heat exchangers are designed, the heat-transfer surfaces are fixed and heat transfer is limited by the surfaces provided in the machine. Therefore, for all operating loads the performance of the machine is determined from the actual heat-transfer surface areas.

In terms of heat-transfer processes on the LiBr-H\(_2\)O solution side, the heat exchangers may be classified into two types: The solution heat exchanger \( X \) (fig. 1) which deals strictly with sensible heat transfer is one type - and the other four: \( G, A, E, \) and \( C \) (fig. 1) which involve latent heat are another type. Heat exchangers \( G \) and \( A \) deal also with the heat of absorption, but since their heat-transfer coefficients are high and their temperature profiles are fairly constant, the heat-transfer analysis is treated in the same manner as those of exchangers \( E \) and \( C \). The following equations (ref. 5) are used for these four exchangers:

\[
EFFN = \frac{T_1 - T_2}{T_1 - T} \tag{1}
\]
where

\( \text{EFFN} \)  \( \text{temperature effectiveness of heat exchanger} \)

\( T_1 \)  \( \text{inlet temperature of heating or cooling medium} \)

\( T_2 \)  \( \text{outlet temperature of heating or cooling medium} \)

\( T \)  \( \text{temperature of LiBr-H}_2\text{O solution of refrigerant (water) undergoing evaporation, absorption, or condensation process} \)

To relate the temperature field to the heat transfer, \( \text{EFFN} \) is rewritten as

\[
\text{EFFN} = 1 - e^{-\frac{U_A}{G C_p}}
\]

(2)

where

\( U \)  \( \text{overall heat-transfer coefficient of heat exchanger} \)

\( A \)  \( \text{total heat-transfer surface} \)

\( G \)  \( \text{flow rate of heating or cooling medium} \)

\( C_p \)  \( \text{heat capacity of medium} \)

Equations (1) and (2) are used to solve for the required outlet temperatures \( T_2 \) 's of the four heat exchangers involving external fluid flows. Ideally, if all temperatures and flow rates are given at the design load, equations (1) and (2) should resolve four \( U_A \) 's for that machine.

To simulate various heat loads other than the design, the corresponding \( U_A \) 's must be calculated from additional equations so that equations (1) and (2) can be used to obtain the various outlet temperatures \( T_2 \) 's. However, the information about the heat-transfer surface is usually not available and the \( U_A \) terms are inseparable. Therefore, the next equations are derived on the \( U_A \) term basis.

Heat exchangers of this kind are typical shell-tube type. The cooling or heating medium is usually on the tube side, and the refrigerant (water) or LiBr-H\(_2\)O solution is on the shell side. The heat-transfer process is governed by the mechanism of the fluid flow on both sides and the tube wall thermal resistance. By definition, \( U \) is written as

\[
\frac{1}{U} = \frac{1}{h_w} + \frac{1}{h} + R_t + F
\]

(3a)

where

\( h_w \)  \( \text{tube-side coefficient due to forced convection} \)

\( h \)  \( \text{shell-side coefficient} \)
$R_t$  tube wall resistance

$F$  sum of fouling factors on both sides

The $R_t$ term in formula (3a) is a function of the tube wall thickness and the material of construction. Typically, its magnitude is very small because of low pressure operation and the use of high conductivity copper based tube material.

The design or selected fouling factor $F$ (ref. 6) is also rather small. The true fouling factor varies with water conditioning and plant operation and cannot be established without test data. Both $R_t$ and $F$ may be considered constant throughout machine operation.

The $h$ term, due to latent heat transfer, is very high for a good cost effective heat exchanger design. The $h$ value for boiling water or steam condensation may be on the order of two to six times the forced convection coefficient $h_w$ (ref. 7). Therefore, it is not a strong factor on the overall heat-transfer coefficient $U$, which may be conveniently written as

$$U = h_w \left( \frac{1}{1 + h_w R} \right)$$

(3b)

where $R$ is the sum of the resistances $(1/h) + R_t + F$.

Equation (3b) implies that $U$ can be found if $h_w$ is known.

To find $h_w$ on the tube side, the following forced convection formula for turbulent flow (ref. 7) is used:

$$h_w \frac{D}{k} = (0.23) \left( \frac{DG}{\mu A_c} \right)^{0.8} \left( \frac{C_p \mu}{K} \right)^{0.4}$$

(4a)

where

$D$  inside diameter of tube

$K$  thermal conductivity

$\mu$  viscosity

$A_c$  flow area

Equation (4a) indicates that the change of $h_w$ is sensitive to the changes of the flow rate $G$ (eight-tenth power function) but less dependent on the heat transport properties. Furthermore, the fluid temperature variations for an absorption machine are rather small, especially in a solar application; thus these temperature dependent properties remain practically constant. Therefore equation (4a) may be rewritten as
\[ h_w \propto G^{0.8} \]

Since proportionality can be established, \( h_w \) may be written as follows in terms of a reference condition with the subscript 0:

\[ h_w = \left( \frac{G}{G_0} \right)^{0.8} h_{w0} \] (4b)

\[ U_0 = h_{w0} \left( \frac{1}{1 + h_{w0}R_0} \right) \] (3)

By combining equations (3b), (3c), and (4b) and solving for \( U \), we obtain

\[ U = \left( \frac{G}{G_0} \right)^{0.8} \left( \frac{1 + h_{w0}R_0}{1 + h_wR} \right) U_0 \] (3d)

As long as the term \( h_wR \) is not substantially different from \( h_{w0}R_0 \), the factor \( (1 + h_{w0}R_0)/(1 + h_wR) \) is approaching unity. If this term is assumed to be one, the expected error in \( U \) is 5 to 10 percent. Under the worst conditions, the error may be as high as 20 percent. Therefore, equation (3d) may be reduced to

\[ U = \left( \frac{G}{G_0} \right)^{0.8} U_0 \]

or

\[ UA = \left( \frac{G}{G_0} \right)^{0.8} (UA)_0 \] (3e)

Equation (3e) implies that, if a reference condition is known, the \( UA \) term at other conditions can be found given the right flow proportions. To find a reference \( UA \), equations (1) and (2) must be used and flow rates are referred to the reference condition. Using actual measured values in the aforementioned formulas instead of the machine design values for the reference point is desirable wherever possible.

The second type of heat exchanger in the absorption machine is a liquid to liquid exchanger (exchanger X in fig. 1). This exchanger is placed in the absorption circuit to improve cycle efficiency. It is also typical of a shell-tube type with a true counterflow arrangement for better heat recovery. The strong solution (rich with water refrigerant)
is pumped through the tubes and the weak solution flows across the tube bundles, with flow deflecting baffles. As was pointed out previously, the heat-transfer rate is a strong function of the flow rate. The strong solution flow rate is greater than that of the weak one. To achieve a high heat-transfer coefficient on the tube side, it is natural for the heat exchanger designer to place the strong solution in the tubes. In addition, the better heat transport properties of the strong solution (more water content) aid in achieving a high coefficient. The lower shell-side coefficient of the weak solution can be improved by using spaced baffles.

Equation (4a) is used to calculate the tube-side coefficient. Equation (4b) is also applicable if the heat transport properties remain practically constant.

As indicated previously, equation (4a) or (4b) is applicable for turbulent flow. For a true counterflow type of heat exchanger, or single-tube pass arrangement, the velocity in the tubes may be reduced under some part load operation. It is possible the flow pattern may shift into the laminar region. Then equation (4a) or (4b) would not be applicable, and the formula for laminar flow (ref. 7) would have to be used.

Since this report is concerned with the simulation of a previously designed machine without knowing the interior arrangement of the heat-transfer surface areas, the laminar formula, even if it is available, is probably not useful for model construction. However, it is reasonable to assume that the turbulent flow formula is used for calculating the tube-side heat-transfer coefficient. In these machines, the heat exchanger with longer tube lengths (thus small flow area and high velocity in the tube) is commonly seen in commercial machines.

The formula for the shell-side coefficient (ref. 7) may be written as follows because the heat transport properties remain practically constant:

\[
\frac{h_{gw} D_e}{K} = 0.33 \left( \frac{D_e G_w}{\mu A_{cross}} \right)^{0.6} \left( \frac{C_p}{K} \right)^{0.3}
\]

or

\[
h_{gw} \propto G_w^{0.6}
\]

where

- \( D_e \) equivalent diameter
- \( h_{gw} \) coefficient of weak solution flow rate
- \( A_{cross} \) flow passage area measured along shell inside diameter

Unlike the tube-side formula, equation (5) is not restricted by the turbulent flow. The shell-side coefficient can be increased by means of closer baffle spacings.
Therefore, it is reasonable to assume that the weak solution with less flow rate is on the shell side.

The relation between the overall heat transfer and the individual coefficients is the same as that of equation (3a). In this case the controlling resistance is on the tube side because of the single tube pass arrangement. The magnitude may be on the order of the shell-side coefficient. Since heat-transfer coefficients on both sides are poor, the magnitude of \((1/h_w) + (1/h)\) in equation (3a) is much larger than that of \(R_t\) and \(F\) (perhaps 10 times larger); therefore, \(R_t\) and \(F\) are neglected and equation (3a) may be re-written as

\[
\frac{1}{U_x} = \frac{1}{h_{gs}} + \frac{1}{h_{gw}}
\]  

(6a)

where

\(x\) refers to solution exchanger
\(gs\) refers to strong solution
\(gw\) refers to weak solution

For a referenced condition, equation (6a) becomes

\[
\frac{1}{U_{x0}} = \frac{1}{h_{gs0}} + \frac{1}{h_{gw0}}
\]  

(6b)

Once again for a given machine, where the interior construction of the machine is not known, equation (6b) cannot be solved without making assumptions. If \(h_{gs0}\) and \(h_{gw0}\) are assumed equal, equation (6b) becomes

\[
h_{gs0} = h_{gw0} = 2U_{x0}
\]  

(7)

By combining equations (7) and (4a) or (5), \(h_{gs}\) and \(h_{gw}\) can be obtained for other simulated conditions; specifically

\[
h_{gs} = \left(\frac{G_s}{G_{s0}}\right)^{0.8} h_{gs0} = \left(\frac{G_s}{G_{s0}}\right)^{0.8} (2U_0)
\]  

(8)

\[
h_{gw} = \left(\frac{G_w}{G_{w0}}\right)^{0.6} h_{gw0} = \left(\frac{G_w}{G_{w0}}\right)^{0.6} (2U_0)
\]  

(9)
Then substituting equations (8) and (9) into equation (6a) and rearranging the terms yield

\[ U_x = (2U_{x0}) \left[ \frac{1}{\left( \frac{G_{s0}}{G_s} \right)^{0.8} + \left( \frac{G_{w0}}{G_w} \right)^{0.6}} \right] \]  

(10)

Since the heat-transfer surface area is fixed, equation (10) may be written as

\[ (UA)_x = 2(UA)_{x0} \left[ \frac{1}{\left( \frac{G_{s0}}{G_s} \right)^{0.8} + \left( \frac{G_{w0}}{G_w} \right)^{0.6}} \right] \]  

(11a)

Equation (11a) again shows that the overall heat-transfer rate at any other condition can be established through a known reference condition (design or test). Equations (8) and (9) can also be extended to include the property corrections if better accuracy is desired. The heat transport properties except thermal conductivity may be found in reference 3. For thermal conductivity values for various LiBr-H₂O solutions, a fraction of water conductivity proportional to water concentration are suggested. In general these effects on heat-transfer coefficients are small and will not be taken into consideration at this time.

The aforementioned equations were derived on the assumption that \( h_{gs0} = h_{gw0} \); the assumption appears valid because (1) the fluid properties on both shell and tube sides are similar and (2) the flow rates are not substantially different within the operating range of the solution concentration. However, if \( h_{gs0} \) is substantially different from \( h_{gw0} \), equation (11a) may be generalized as

\[ (UA)_x = (F1)(UA)_{x0} \left[ \frac{1}{\left( \frac{G_{s0}}{G_s} \right)^{0.8} + (F2) \left( \frac{G_{w0}}{G_w} \right)^{0.6}} \right] \]  

(11b)
where, for example,

\[ h_{gs0} = h_{gw0} \quad F1 = 2 \quad \text{and} \quad F2 = 1 \]

\[ h_{gs0} \ll h_{gw0} \quad F1 = 1 \quad \text{and} \quad F2 = 0 \]

\[ h_{gs0} = 1.5 h_{w0} \quad F1 = 2.5 \quad \text{and} \quad F2 = 2/3 \]

Equation (11b) may be useful to experimentally determine the actual values of \( F1 \) and \( F2 \) for use in the program for a given machine.

Equation (11a) or (11b) can be solved if \((UA)_{x0}\) is known or may be found from a given set of the design temperatures. The effectiveness is given in terms of the temperatures (refs. 1 and 4) as

\[ \text{EFFNX} = \frac{T_g - T_5}{T_g - T_a} \]  \hspace{1cm} (12)

where

\( T_g \)  temperature of generator

\( T_5 \)  outlet temperature of weak solution

\( T_a \)  temperature of absorber

In general the exchanger is designed with the effectiveness \( \text{EFFNX}_0 \) = 0.7 to 0.8.

If \( T_5 \) in equation (12) for the design load is not known, the relation between \( \text{EFFNX}_0 \) and \( T_{50} \) may be established by heat balance (ref. 1).

When \( \text{EFFNX}_0 \) is found together with flow rates \( G_{s0} \) and \( G_{w0} \) and the solution heat capacities \( C_{s0} \) and \( C_{w0} \), then \((UA)_{x0}\) is calculated from the following equation:

\[ \text{EFFNX} = \frac{1 - e^{-NTU_x \left[ \frac{1 - (C_{min}/C_{max})}{1 - \left( \frac{C_{min}}{C_{max}} \right) e^{-NTU_x \left[ 1 - (C_{min}/C_{max}) \right]} \right]}}}{1 - \left( \frac{C_{min}}{C_{max}} \right) e^{-NTU_x \left[ 1 - (C_{min}/C_{max}) \right]}} \]  \hspace{1cm} (13)

where

\( NTU_x \)  \((UA)_{x0}/C_{min}\)

\( C_{min} \)  \( G_w C_w \)

\( C_{max} \)  \( G_s C_s \)
\( C_w \) heat capacity of weak solution
\( C_s \) heat capacity of strong solution

The subscript 0 used previously has been deliberately omitted in equations (12) and (13) for the purpose of generalization. Then \( C_{\text{min}} = (G_w C_w) \) and \( C_{\text{max}} = (G_s C_s) \) because \( G_w < C_s \) and \( C_w < C_s \) for LiBr-H\(_2\)O absorption machine. The \( (UA)_{x0} \) is solved implicitly in equation (13).

OTHER CONSIDERATIONS

The equations derived in the previous section together with the thermodynamic equations discussed in the section THERMODYNAMIC CYCLE are the working formulas for the five heat exchangers to be used in the construction of the simulation model. In addition to these formulas, heat losses, pump capacity, operating range of the solution concentrations, and operating temperature limits should be included. Unfortunately machine construction does vary with the design approach of different manufacturers, and the construction information is usually not available. It is difficult to generalize all the limitations to be accommodated by the model. Nevertheless, some of the important considerations that should be taken into account follow.

Heat Losses

The heat losses vary with the specific design and the ambient environment in which the machine is installed. Heat may leak out of or into the machine, and between the partition shells separating the heat exchangers in the machine. The result is that additional heat supply is required to accommodate these losses. To account for these losses, a simplistic approach is to add a fixed percentage to the heat supply. A few percent may be sufficient for the type machine considered herein. The thermodynamic equations (appendix A) may be modified as follows:

\[
Q_G = (G_w H_5 - G_s H_1 + G_R H_7)(F_G Q) \quad (14)
\]

\[
Q_C = G_R (H_7 - H_8) \left[ 1 + \left( \frac{Q_G}{Q_G + Q_E} \right) (1 - F_{QG}) \right] \quad (15)
\]

\[
Q_A = (G_w H_5 - G_s H_1 + G_R H_{10}) \left[ 1 + \left( \frac{Q_G}{Q_G + Q_E} \right) (1 - F_{QG}) \right] \quad (16)
\]
where
\[ F_{QG} \text{ multiplication factor} \]
\[ F_{QG} = 1 \text{ (no heat loss considered)} \]
\[ F_{QG} = 1.02 \text{ (equivalent 2 percent loss)} \]

Solution Pump Capacity

Normally the pump capacity is chosen to meet the design load. For part load operation, the required flow rate may or may not exceed the maximum capacity. For a particular load demand, if heat source temperature is low and/or the cooling water temperature is high, the machine, based on the thermodynamic cycle analysis, tends to demand more solution flow. Since the flow control is not known and varies somewhat with different machines, it is assumed that the solution flow rate cannot exceed the capacity of the design point.

Concentration of the LiBr-H\textsubscript{2}O Solution

For an absorption process to exist in operation, there are limits on the solution concentrations. If the concentration is too rich, crystallization will occur. If the concentration is too lean, no absorption process will occur. Reference 2 suggested that the concentrations should be kept within 0.5 to 0.65 range. For this model a range from 0.4 to 0.68 has been used.

Temperatures and Temperature Differences

The temperature limitations, like the solution concentrations, are set for the operable absorption process. Usually these are the outlet temperatures of the external fluids in heat exchanger G, C, A, and E (fig. 1). The limits of these temperatures have been placed in the program (see appendix B).

In addition to the temperature limits, the temperature difference across the heat exchanger surfaces are also limited by the heat-transfer processes. In general the temperature differences between the two heat exchange mediums at outlet condition will be used for setting the limits (see appendix B).

When the aforementioned limits and the concentration limits are properly set, the solution heat exchanger temperature as well as the pressure limits may be neglected.
MODELING ALGORITHM

With the necessary equations and the limiting conditions established, the next step is to formulate an algorithm for computer operation. The desired solution for a given set of inputs is the one that achieves the lowest possible outlet temperature of the chilled water. The heat balance is not only required to satisfy the thermodynamic analysis but also simultaneously satisfy the heat exchanger equations.

The model consists of two different algorithms. One part is used to solve for the reference or design conditions. Another part is used to solve for the off-design condition based on the established reference condition. The second part is simply to perform an internal heat balance to establish the corresponding outlet temperatures of the three flow streams, namely hot water GH, cooling water GC, and chilled water GE. The calculation sequence for this part is first outlined as follows:

1. Input GH, GC, GE, TH1, TA1, TE1 and an off-design tonnage, (see fig. 1).
2. Calculate flow rate per ton for flow GH, GC, and GE.
3. Calculate effectiveness (eq. (2)) for exchanger G, C, A, and E.
4. Calculate TE2, TE, TH2, and TG.
5. Calculate TC2 with an assumed COP.
6. Assume TA.
7. Calculate TC.
8. If TA or TC exceed limits, change tonnage.
9. Calculate TG, TC, TA, and TE with newly assumed tonnage.
10. Calculate solution concentration.
11. If X1 or X4 exceeds limits, change tonnage.
12. Calculate enthalpies H8 and H10 of refrigerant at outlets of condenser C and evaporator E, respectively.
13. Calculate refrigerant flow GR and solution flows GS and GW, respectively.
15. Calculate two outlet temperatures T3 and T5 of solution exchanger.
16. Calculate refrigerant enthalpy H7 at outlet of generator G, weak solution enthalpy H5 at outlet of solution exchanger X, and strong solution enthalpy H, at outlet of absorber A.
17. Calculate generator heat QG, condenser heat QC, and absorber heat QA.
18. Calculate COP.
19. If TA is not agreeable with assumed value, adjust TA to suit.
20. If COP is not agreeable with assumed value, adjust COP to suit.
21. Check temperature difference limits.
22. Check pumping rate limits.
23. Check concentration limits.
24. Force tonnage to maximum.
(25) Check chilled water outlet temperature TE2 at set point.
(26) Calculate pressure PE and PC.

To establish the reference conditions, several of the aforementioned indicated steps are repeated. The algorithm used depends upon the information available.

If all the design or reference temperatures are given but the flow rates are not, steps (10) to (18) and step (26) are repeated. The flow rates and all reference (UA)'s are the calculated outputs. The effectiveness of the solution heat exchanger can be calculated from the known temperatures (eq. (12) as an input to the program).

If all three external flow rates are known instead of their outlet temperatures, steps (1) to (18) and step (26) are repeated. In this case the corresponding outlet temperatures are determined.

If the outlet temperature of the solution heat exchanger T5 or its effectiveness is not given, an assumed effectiveness must be used as an input until a rated reference tonnage is found.

PROGRAM DESCRIPTION

The computer program was written in FORTRAN IV language. It can be used as a subroutine to simulate the absorption machine performance in a cooling system. Although the equations derived are steady-state type, no restriction is imposed for use in the transient analysis of a cooling system.

When used as a subroutine, the program may have to be modified to accept a set of the design or the test conditions. The flow rates and the inlet and outlet temperatures of the three external fluid streams are system connected to run the simulation. If additional outputs such as heat loads, COP, and operating pressures are required, they may be system linked or printed out for analysis.

When used as an independent program, the first case is treated as the reference case. The program calculates additionally needed reference values and stores these values automatically in the program. Starting with the next case, the user inputs as many off-design cases as are desired. NAMELIST input is used in the program.

All tolerances for the limitation conditions discussed previously have been prestored in the program but can be changed as desired. The units system used to perform the calculation is metric but provision to use British units for inputs and outputs is included. Changing either the units or the tolerances shall be discussed in the next section and appendix B.
OPERATION OF THE PROGRAM

Use as a Subroutine

If the program is used as a subroutine, the reference data and program controls must be inserted as data statements or their equivalent by the user. The required data are UAG0, UAC0, UAE0, UAA0, UAX0, GS0, GW0, and TON0. The controls are FQG, METRIC, KLBHR, and JWRITE (see appendixes B and C).

The input variables are currently placed in an array called XIN. These variables (listed in order), are GHT, GCT, GET, TH1, GA1, TE1, and TONX (see appendixes B and C).

The output variables are arranged in an array OUT. These variables are GHT, GCT, GET, TH2, TC2, TE2, and TON. If additional outputs such as COP, PC, and PE are required, the user may place these variables in the additional locations of array OUT (see appendixes B and C).

Use as the Main Program

If the program is used independently, the reference data must be calculated from this program based upon the available design or experimental informations. The input variables in this case will be TH2, TA2, or TC2, TE2, TH1, TA1, TE1, TG, TC, TA, TE, TON0, TONX, KLBHR, METRIC, and JREF (see appendix B and fig. 1). TONX is the initial guess of the actual load. The data are entered via a NAMELIST read and are for reference case. The NAMELIST name is REF. The first tabulated output will be the results of the design conditions and the table is identified with a case marked 0.

To run other cases with fixed heat-transfer surfaces (the same machine), additional cases are placed in the run stream with a NAMELIST name of VAR. As many cases as desired can be run. The input for these cases are GH, CC, GE, TH1, TA1, TE1, and TONX (see appendixes B and C). The outputs are tabulated as before, and the case is identified with a case number greater than 0.

The convergence is controlled by KTA, KCOP, KTONI, and KTON2. If the number of the iterations is excessive, the output may be incorrect. The user must examine the results to decide whether he should increase the number of iterations, or discontinue his run because of exceeding machine operating constraints.

The tolerance controls for the temperatures and concentrations are currently pre-stored in the program (see appendixes B and C). The values may be changed to suit the user's purpose.
SAMPLE CALCULATIONS

Two sample computer printouts are included to demonstrate the use of the program in appendix C.

Sample 1 shows that, for a given set of the design conditions, the program not only finds the correct design load but generates the results for the off-design loads as well.

The absorption machine used in the sample calculations is a TRANE model C1H (ref. 6). This model was designed for a nominal rated tonnage at 174 tons. The printout table (case 0) shows that the calculated tonnages agree with the design load. The output of this case is then stored in the program as the reference data of the machine to be used for the off-design runs.

A total of 130 off-design cases (the off-design loads and operating conditions in table 2C1H of ref. 6), have been run with the program. Most of the calculated tonnages agree with the data in reference 6 within 2 percent and generally are slightly greater than the table values (two typical cases are shown in appendix C). In some of the cases, however, the calculated values are high by 9 percent. These cases usually are associated with the extremely high or low outlet temperature of the chilled water. All cases were run on the assumption that the nominal design flow rates were chosen to establish the rated table values. If these flow rates are not nominal but varied within the design range, the program calculated tonnages can be brought to agreement with those tables indicated.

Sample 2 was intended to show that, with minor changes, the program can be used as a subroutine in a system program. In this case the system program is TRNSYS (ref. 1). Sample 2 is a solar assisted building cooling system modeled with TRNSYS program (see appendix C).

CONCLUDING REMARKS

A computer model of a LiBr-H_2O single-stage absorption machine has been developed. By utilizing a given set of design data but without knowing the interior characteristics of the machine, the off-design performance can be simulated or evaluated. Although the model is not validated experimentally, it can be a useful tool for analyzing the capability of a given machine, or for studying the machine performance in a cooling system.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, May 16, 1978,
776-22.
APPENDIX A

SYMBOL LIST, THERMODYNAMIC FORMULAS, AND EQUATIONS FROM REFERENCE 1

Strong concentration ($X_1 > 0.5$):

$$X_1 = \frac{(49.04 + 1.125 TA - TE)}{(134.65 + 0.47 TA)} \text{ kg LiBr} \quad \text{kg solution}$$

Weak concentration ($X_4 < 0.65$):

$$X_4 = \frac{(49.04 + 1.125 TG - TC)}{(134.65 + 0.47 TG)} \text{ kg LiBr} \quad \text{kg solution}$$

Enthalpy of condenser outlet:

$$H_8 = (TC - 25) \quad \text{kcal/kg}$$

Enthalpy of evaporator outlet:

$$H_{10} = (572.8 + 0.417 TE) \quad \text{kcal/kg}$$

Refrigerant flow:

$$GR = \frac{QE}{(H_{10} - H_8)} \quad \text{kg hr}$$

Strong solution flow:

$$GS = GR \frac{X_4}{(X_4 - X_1)} \quad \text{kg hr}$$

Weak solution flow:

$$GN = GR \frac{X_1}{(X_4 - X_1)} \quad \text{kg hr}$$
Heat capacity of strong solution:

$$CX_1 = 1.01 - 1.23(X_1) + 0.48(X_1)^2 \quad \text{kcal/(kg)(°C)}$$

Heat capacity of weak solution:

$$CX_4 = 1.01 - 1.23(X_4) + 0.48(X_4)^2 \quad \text{kcal/(kg)(°C)}$$

Outlet temperature of weak solution

$$T_5 = T_G - (EFFNX)(T_G - T_A) \quad ^\circ \text{C}$$

Outlet temperature of strong solution:

$$T_3 = T_A + (EFFNX)\left(\frac{X_1}{X_4}\right)\left(\frac{CX_4}{CX_1}\right)(T_G - T_A) \quad ^\circ \text{C}$$

Enthalpy of absorber outlet:

$$H_1 = \left[42.81 - 425.92(X_1) + 404.67(X_1)^2\right] + \left[1.01 - 1.23(X_1) + 0.48(X_1)^2\right](T_A) \quad \text{kcal/kg}$$

Enthalpy of weak solution at heat exchanger outlet:

$$H_5 = \left[42.81 - 425.92(X_4) + 404.67(X_4)^2\right] + \left[1.01 - 1.23(X_4) + 0.48(X_4)^2\right](T_5) \quad \text{kcal/kg}$$

Enthalpy of refrigerant at generator outlet:

$$H_7 = (572.8 + 0.46 T_G - 0.043 T_C) \quad \text{kcal/kg}$$

Condenser heat load:

$$QC = (GR)(H_7 - H_8) \quad \text{kcal/hr}$$

Generator heat load:

$$QG = (GW)(H_5) + (GR)(H_7) - (GS)(H_1) \quad \text{kcal/hr}$$
Absorber heat load:

\[ QA = (GW)(H5) + (GR)(H10) - (GS)(H1) \text{ kcal/hr} \]

Coefficient of performance:

\[ \text{COP} = \frac{QE}{QG} \]

Evaporator heat load:

\[ QE = 3024.0 \text{ kcal/hr} \]

Evaporator pressure:

\[ PE = \text{antilog}_{10} \left( 7.8553 - \frac{1555}{TE + 273.15} - \frac{11.2414 \times 10^4}{(TE + 273.15)^2} \right) \text{ mm Hg} \]

Condenser pressure:

\[ PC = \text{antilog}_{10} \left( 7.8553 - \frac{1555}{TC + 273.15} - \frac{11.2414 \times 10^4}{(TC + 273.15)^2} \right) \text{ mm Hg} \]
SYMBOL LIST FOR HEAT-TRANSFER CALCULATIONS IN COMPUTER PROGRAM

Flow rates, gal/min, lb/hr, kg/hr

- GH (Hot water supply)
- GC (Cooling water supply)
- GA (Cooling Water supply)
- GE (Returning chilled water)
- GR (Refrigerant - water)
- GW (Weak solution)
- GS (Strong solution)

Temperatures, °F, °C

- TH1, TH2 (Inlet and outlet conditions of GH)
- TC1, TC12, or TA2 (Inlet and outlet conditions of GA)
- TC12, TC2 (Inlet and outlet conditions of GC)
- TE1, TE2 (Inlet and outlet conditions of GE)
- TG (Generator)
- TC (Condenser)
- TA (Absorber)
- TE (Evaporator)

Heat-transfer rates, Btu/(hr)(°F), cal/(hr)(°C)

- UAG (Generator)
- UAC (Condenser)
- UAA (Absorber)
- UAE (Evaporator)
- UAX (Heat exchanger)

First digit: Overall heat-transfer coefficient
Second digit: Overall heat-transfer surface
Third digit: Component symbol
Number of heat-transfer units

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>NTUG</td>
<td>(Generator)</td>
</tr>
<tr>
<td>NTUC</td>
<td>(Condenser)</td>
</tr>
<tr>
<td>NTUA</td>
<td>(Absorber)</td>
</tr>
<tr>
<td>NTUE</td>
<td>(Evaporator)</td>
</tr>
<tr>
<td>NTUX</td>
<td>(Heat exchanger)</td>
</tr>
</tbody>
</table>

Heat-transfer effectiveness

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>EFFNG</td>
<td>(Generator)</td>
</tr>
<tr>
<td>EFFNC</td>
<td>(Condenser)</td>
</tr>
<tr>
<td>EFFNA</td>
<td>(Absorber)</td>
</tr>
<tr>
<td>EFFNE</td>
<td>(Evaporator)</td>
</tr>
<tr>
<td>EFFNX</td>
<td>(Heat exchanger)</td>
</tr>
</tbody>
</table>

(A digit 0 following aforementioned symbols signifies a reference or a design condition being used. A digit T following aforementioned symbols and symbols in appendix A signifies total quantities.)

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>TON0</td>
<td>(Reference refrigerant tonnage)</td>
</tr>
<tr>
<td>TON</td>
<td>(Tonnage calculated)</td>
</tr>
<tr>
<td>TONX</td>
<td>(Tonnage variable)</td>
</tr>
<tr>
<td>COPX</td>
<td>(COP variable)</td>
</tr>
<tr>
<td>TAX</td>
<td>(TA variable)</td>
</tr>
<tr>
<td>GSC1</td>
<td>(Product of strong solution flow and heat capacity)</td>
</tr>
<tr>
<td>GWC4</td>
<td>(Product of weak solution flow and heat capacity)</td>
</tr>
<tr>
<td>CRATIO</td>
<td>=GWC4/GSC4</td>
</tr>
<tr>
<td>EXPX</td>
<td>(Exponential function for heat exchanger)</td>
</tr>
</tbody>
</table>

Controls and limits

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>METRIC</td>
<td>(Input to be metric units &gt; 0)</td>
</tr>
<tr>
<td>KLBHR</td>
<td>(Input to be lb/hr &gt; 0)</td>
</tr>
<tr>
<td>JWRITE</td>
<td>(Write output &gt; 0)</td>
</tr>
<tr>
<td>KTA</td>
<td>(TA converging cycle = 50)</td>
</tr>
<tr>
<td>KCOP</td>
<td>(COP converging cycle = 50)</td>
</tr>
</tbody>
</table>
KTON1 and KTON2  (TONX converging cycle = 100)

ACONST = 1.0°C  Limits of (TE2 - TE)
BCONST = 1.296°C  Limits of (TA - TC12)
CCONST = 1.425°C  Limits of (TC - TC2)
DCONST = 1.919°C  Limits of (TH1 - TG)

TELO - 2.22°C  (Lowest temperature limits of TE)
TE2SET = 4.43°C  (Lowest temperature limits of TE2)

COPHI = 0.93  (Highest limits of COP)
COPLO = 0.60  (Lowest limits of COP)

FQG = 1.0  (No heat loss added)

EFFNX = 0.71428  (Initialization of EFFNX)
APPENDIX C
SAMPLES 1 AND 2 WITH PROGRAM LISTINGS

Sample 1: LiBr-H\textsubscript{2}O Single-Stage Absorption Machine Used as a Main Program

```plaintext
1* C USE THIS TO EVALUATE OUTPUT OF AN ABSORPTION MACHINE WITH FIXED --UA--
2* C ALL WATER SPECIFIC HEAT & DENSITY ASSUMED TO BE --1.0-- EXCEPT HOT WATER
3* DIMENSION XIN(10),PAR(13),XNTU(6),EFFN(6)
4* DIMENSION X(6),Y(6),GIN(3)
5* DIMENSION TONGVH(160),TONGCAL(160)
6* C
7* C --METRIC=0,BRITISH UNITS USED.--------JWRITE=1 WRITE ALL,JWRITE=0 NO WRITE
8* C KLBHR=0,GPH FOR FLOW INPUT.--------KLBHR=1, LBS/HR INPUT
9* DATA METRIC/0/,KLBHR/0/,JWRITE/1/
10* DATA PMCR/4536/
11* DATA TFC1/32.0,TFC2/1.8/
12* DATA CALBTU/3.96831/
13* C
14* C
15* C
16* C
17* C
18* DATA METRIC/0/,KLBHR/0/,JWRITE/1/
19* DATA ACONST/1.0/,ACONST/1.296/,CCONST/1.423/,DCONST/1.919/
20* C
21* C
22* DATA TELO/2.22/,TE2SET/4.43/
23* C
24* DATA METRIC/0/,KLBHR/0/,JWRITE/1/
25* DATA COPH/0.93/,COPLO/0.66/,FQG/1.0/
26* DATA EFFNX=0.71426 FOR T5=135 F EFFNX=(TG-T5)/(TG-TE)
27* DATA EFFNX=0.71428/
28* C
29* C
30* C
31* C
32* DATA XRSV(1)=TH2
33* DATA XRSV(2)=TA2
34* DATA XRSV(3)=TA1
35* DATA XRSV(4)=TG
36* DATA XRSV(5)=TC
37* DATA XRSV(6)=TE
38* DATA XRSV(7)=T5
39* C
40* C
41* C
42* C
43* C
44* C
45* DATA XIN(1)=TH2
46* DATA XIN(2)=TA2
47* DATA XIN(3)=TE
48* DATA XIN(4)=TA1
49* DATA XIN(5)=TA1
50* DATA XIN(6)=TE1
51* DATA XIN(7)=TG
52* DATA XIN(8)=TC
53* DATA XIN(9)=TA
54* DATA XIN(10)=TE
55* DATA XINRSV(1)=XIN(1)
56* CONTINUE
57* C
58* C
59* C
60* C
61* C
62* C
63* C
```
IF(METRIC.GT.1) TFTC1=0.0
IF(METRIC.GT.0) TFTC2=1.0
IF(METRIC.GT.0) PDKG=1.0
IF(METRIC.GT.0) CALBTU=1.0
IF(METRIC.GT.C) BPH=1.0
IF(KLBHR.GT.0) BPH=1.0
IF(REF.GT.0) GO TO 19
7 CONTINUE

UA VALUES ARE PER TON BASIS--------
TON=TON
GSO=GS*PDKG
GHO=GH*PDKG
GEO=GCT
PAR(1)=UAG*PDKG
PAR(2)=UAC*PDKG
PAR(3)=UA*PDKG
PAR(4)=UAX*PDKG
PAR(5)=UAX*PDKG
UA=PAR(5)

INPUT=1
C
990 CONTINUE
JREF=0
READ(5,VAR,END=999)
XIN(1)=GH
XIN(2)=GC
XIN(3)=GE
XIN(4)=TH1
XIN(5)=TA1
TC1=TA1
XIN(6)=TE1
XIN(7)=TONX
GESAV=XIN(3)
TE2SAV=TE2
TH2SAV=TH2
KERROR=0
TON IS AN ASSUMED VALUE TO START
TON=TONX
TONREF=TONX*0.5
TONMIN=TONX*0.1
TONMAX=TONX*1.2
IF(XIN(7).LE.TONREF) TONX=TONREF

19 CONTINUE
KGC=0

9 CONTINUE

FLOW RATES ARE PER TON BASIS--------
QE=3223*9573
H1=0.
H5=0.
H7=G.
H8=G.
H10=0.
TON=0.0
KTON1=0
KTON2=0
KTON3=-1

1310 X4=0.67
1320 X1=0.41
1330 11 WRITE=0
1340 11 CONTINUE
1350 11 IF(COPX*.LE.*COPLO.*OR.*COPX*.GE.*COPHI) COPX=0.722
1370 11 CONTINUE
1390 GH=XIN(1)/TONX*PDKG*BPH*Q.975
1400 GC=XIN(2)/TONX*PDKG*BPH
1410 GE=XIN(3)/TONX*PDKG*BPH
1420 TH1=(XIN(4)-TFTC1)/TFTC2
1430 TC1=(XIN(5)-TFTC1)/TFTC2
1440 TE1=(XIN(6)-TFTC1)/TFTC2
1450 TE2SVM=TF2SAV-TFTC1)/TFTC2
1460 TH2SVM=(TH2SAV-TFTC1)/TFTC2
1470 11 IF(JREF.EQ.0) GO TO 20
1480 TH2=(XIN(1)-TFTC1)/TFTC2
1490 TA2=(XIN(2)-TFTC1)/TFTC2
1500 IF(JREF.EQ.2) TC12=TAF2
1510 IF(JREF.EQ.1) TC2=TA2
1520 TE2=(XIN(3)-TFTC1)/TFTC2
1530 TG=(XIN(7)-TFTC1)/TFTC2
1540 TC=(XIN(8)-TFTC1)/TFTC2
1550 TA=(XIN(9)-TFTC1)/TFTC2
1560 TE=(XIN(10)-TFTC1)/TFTC2
1570 11 CONTINUE
1580 11 GO TO 21
1590 11 CONTINUE
1600 11 C ASSUME TUBE SIDE WATER FILM COEF.* IS CONTROLLING
1610 GH1=(XIN(1)/GHI)**0.8
1620 GC1=(XIN(2)/GEC)**0.8
1630 GE1=(XIN(3)/GEC)**0.8
1640 C
1650 C TOTAL BASIS IN METRIC UNITS
1660 SIN(1)=GH/GH1*TONX
1670 SIN(2)=GC/GC1*TONX
1680 SIN(3)=GE/GE1*TONX
1690 11 C
1700 DO 10 I=1,4
1710 XNTU(I)=PAR(I)/GIN(I)*1.D0*TONC
1720 IF(I.EQ.4) XNTU(I)=PAR(I)/GIN(2)*TONC
1730 11 IF(XNTU(I).GE.1.F*) GO TO 5
1740 EFFN(I)=3.*EXP(-XNTU(I))
1750 GO TO 10
1760 11 CONTINUE
1770 EFFN(I)=0.999
1780 11 CONTINUE
1790 C
1800 C
1810 TE2=TE1-QE/GE
1820 TH2=TH1-10E/COPX)/GH
1830 C
1840 C FOR CHECKING TRANE TABLE FIGURES ONLY TE2 *TH2 KNOWN INSTEAD
1850 IF(JTF2*LF.*C ) GO TO 603
1860 TE2=TE1
1870 TH2=TH1
1880 C TE2=TF2SAV
1890 C TH2=TH2SAV
1900 C TE2=TF2SVM
1910 C TH2=TH2SVM
1920 C TE1=TE2*QF/GE
1930 TH1=TH2*10E/COPX)/GH
1940 C 603 CONTINUE
1950 C
1960 C
1970 C TE=TE1-(TF1-TE2/EFFN(3)
TG = TH1 - (TH1 - TH2) / EFFN(1)
TC = TC1 + (1.0 + 1.0/COPX) * QE/6C

C ASSUMED A VALUE FOR TA
TA2 = (TC1 + TC2) * 0.5
TA = TC1 - (TC1 - TA2) / EFFN(4)
KTA = 0

15 CONTINUE
TC = TC2 / EFFN(2) - (1.0 / EFFN(2) - 1.0) * (TC1 + EFFN(4) * (TA - TC1))

16 IF(TC.LT.TA) GO TO 40
17 IF(TC.GE.TA) GO TO 41
18 IF(TC.GE.TG) GO TO 41

21 CONTINUE
X1 = (49.04 + 1.125 * TA - TE) / (134.65 + 3.47 * TA)
X4 = (49.04 + 1.125 * TG - TC) / (134.65 + 0.47 * TG)

19 IF(X1.LT.CONST1) GO TO 45
20 IF(X4.LT.X1) GO TO 43

C ASSUMING ORIGINAL FILM COEF. EQUAL ON BOTH SIDES. --GW ON SHELL SIDE

F1 = 2.0
F2 = 1.0

C
23 F1 = 2.0, F2 = 1.0
HGS = HGW.

24 IF(XTUX.GT.X1) GO TO 17

25 CONTINUE
XNTUX = UAX / GW4 * (TONX / TONX)

26 CONTINUE
C
28 IF(ABS(1.0 - CRATIO).LT.0.1) GO TO 13
29 IF((XNTUX - CRATIO).GE.10.) GO TO 12
30 EXPX = EXP(-XNTUX * (1.0 - CRATIO))
31 EFFNX = (1.0 - EXPX) / (1.0 - CRATIO * FXPX)
32 GO TO 14
33 C
34 C
35 C
36 C
37 C
38 C
39 C
40 C
41 C
42 C
43 C
44 C
45 C
46 C
47 C
48 C
49 C
50 C
51 C
52 C
53 C
54 C
55 C
56 C
57 C
58 C
59 C
60 C
61 C
62 C
63 C
64 C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C

C
IFIC(JREF.GT.0) GO TO 60

TC1Z=TC1+QA/GC

TAX=TC1-(TC1-TC1Z)/EFFF(4)

IFIC((TAX+Tb)/2 0.5) GO TO 15

IFIC(ABS(COPX-COP).LT.0.0001) GO TO 42

IFIC(KCOP.EQ.50) GO TO 42

IFIC((COPX+COP)*50.5) KCOP=KCOP+1

IFIC(I1.I.T.X4.EQ.1) 44.65,65

IFIC(X1.GT.CONST1.AND.X4.LT.CONST4.AND.X4.GT.X1) GO TO 46

IFIC(KTON2-100) 49,43,43

IFIC(X1.6T.CONST1.AND.X4.6T.CONST4.AND.X4.GT.X1) GO TO 46

IFIC(X1.GT.CONST1.AND.X4.LT.CONST4.AND.X4.GT.X1) GO TO 46

IFIC(X1.LT.Y1.AND.KONX.EQ.1) GO TO 60

IFIC(X1.LT.Y1) GO TO 45

IFIC(I1.I.T.X4.EQ.1) 47 CONTINUE

IFIC(X1.GT.CONST1.AND.X4.LT.CONST4.AND.X4.GT.X1) GO TO 46

IFIC(KTON2-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43

IFIC(KTON1-100) 49,43,43
KTON1=KTON1+1
GOTO 11

C
46 CONTINUE
IF(KTONX.EQ.1.0 OR TONX.LE.0.0) GOTO 60
IF(TONX.GE.TONX) GOTO 50

C
CHECK MAX. STRONG SOLUTION PUMP RATE -----------------
GST0=GST0+TONX
GSPUMP=GST0*TONX
IF(GSPUMP.GT.GST0) GOTO 43

C
TONX=TONX
IF(TE2.LT.TE2SET.OR.TE.LE.TELE) GOTO 48
GOTO 49

C
60 CONTINUE
65 IF(JREF.GT.0.0 OR KTONX.EQ.1.0 OR KGC.GE.30) GOTO 66
KGC=KGC+1
XIN(3)=XIN(3)*T0NX-1.0)/TONX
GOTO 9

C
C
C
31 CONTINUE

C
35 CONTINUE
UAG=XNTU(1)*GH
UAC=XNTU(2)+GC
UAE=XNTU(3)+GE
UAA=XNTU(4)+GC
UAX=XNTU(6)+GC

C
A=ALOG(10.0)
B=1555.0/(TE+273.15)
C=11.2414/((TE+273.15)**2
PE=EXP(A*7.8553-8-C)
F=1555.0/(TC+273.15)
C=11.2414/((TC+273.15)**2
PC=EXP(A*7.8553-8-C)
DG=DQ*CALPTU
QC=DQ*CALPTU
ID=H1*CALBTU

T3=T3*TFTC2*TFTC1
T5=T5*TFTC2*TFTC1

TH1=TH1*TFTC2*TFTC1
TH2=TH2*TFTC2*TFTC1
TC1=TC1*TFTC2*TFTC1
TC12=TC12*TFTC2*TFTC1
TA2=TA2

TC2=TC2*TFTC2*TFTC1
TE1=TE1*TFTC2*TFTC1
TE2=TE2*TFTC2*TFTC1
TE =TE *TFTC2*TFTC1
TA =TA *TFTC2*TFTC1
TC =TC *TFTC2*TFTC1
TG =TG *TFTC2*TFTC1

UAG=UAG/PDKG
UAC=UAC/PDKG
UAE=UAE/PDKG
UAA=UAA/PDKG
UAX=UAX/PDKG

G=GH/PDKG
GC=GC/PDKG
GA=GC
GE=GE/PDKG
GR=GR/PDKG
GW=GW/PDKG
GS=GS/PDKG

GS1=GS1/PDKG
GWC4=GWC4/PDKG

QC=GC*TON
QCT=QC*TON
QET=TON

QAT=QA*TON
UAGT=UAG*TON
UACT=UAC*TON
UAE=UAE*TON
UAAT=UAAT*TON

GHT=GH*TON/BPH/C.975
GCT=GC*TON/BPH
GAT=GCT
GET=GT*TON/BPH

DT12E=TE1-TE2
DT12A=TA2-TC1
DT12C=TC2-TA2
DT12G=TH1-TH2

DE2=TE2-TE
DTE2=TA2-TE
DTC2=TC2-TC1

DTG=TH2-TG

IF(KTA,G,E.50 .OR., KGE,P*GE.50) GO TO 58
IF(KTON2,G,E.100 .OR., KTON1,G,E.100) GO TO 58
IF(JWRITE,EQ.,0) GO TO 59
GOTO 30
58 CONTINUE
KPROP=1
C
464* 465* 466* 467* 468* 469* 470* 471* 472* 473* 474* 475* 476* 477* 478* 479* 480* 481* 482* 483* 484* 485* 486* 487* 488* 489* 490* 491* 492* 493* 494* 495* 496* 497* 498* 499* 500* 501* 502* 503* 504* 505* 506* 507* 508* 509* 510* 511* 512* 513* 514* 515* 516* 517* 518* 519* 520* 521* 522* 523* 524* 525* 526* 527* 528* 529* 530* 531* 532*
X TA2 DT12C DTC2 TC2 DT12G DTG2 TH2

535* 97 FORMAT(1X,4HTEMP,12F10.3///)
536* 98 FORMAT(5X,4(I6,2X),8F10.3,3X,I3///)
537* C
538* C 59 CONTINUE
539* IF(JREF.GT.0) GO TO 7
540* C
541* C 61 CONTINUE
542* C
543* C TONGVN(INPUT)=XIN(7)
544* TONCAL(INPUT)=TON
545* INPUT=1+INPUT
546* GH=XIN(1)
547* GC=XIN(2)
548* GE=GESAV
549* TH1=XIN(4)
550* TA1=XIN(5)
551* TE1=XIN(6)
552* TONX=XIN(7)
553* TE2=TF2SAV
554* TH2=TH2SAV
555* IF(LVAR.EQ.0) GO TO 99D
556* DO 301 I=1,11
557* XINRSV(I)=XIN(I)
558* 301 CONTINUE
559* TC2=TC2RSV
560* TA2=TA2RSV
561* GO TO 99D
562* 999 CONTINUE
563* WRITE(6,601)
564* C WRITE(6,602) (TONGVN(M),TONCAL(M),M=1,160)
565* WRITE(6,602) (M,TONGVN(M),TONCAL(M),M=1,160)
566* 601 FORMAT(1X,11D10) THE FOLLOWING ARE KNOWN TON VS CALCULATED FOR TRANS
567* 6C2 FOR?AT(1X,6(4X,7F8.2)/
568* 6C2 FOR?AT(1X,6(4X,7F8.2)/
569* C STOP
570* C END
Sample 2: LiBr-H₂O Single-Stage Absorption Machine Used as a Subroutine in TRNSYS

```
1* SUBROUTINE TYPE17(TIME,XIN,OUT,T,DTI,DT2,PAR)
2* COMMON /PR2/ TIME,TIME2,TFINAL,DEL
3* C USE THIS TO EVALUATE OUTPUT OF AN ABSORPTION MACHINE WITH FIXED --UA--
4* C ALL WATER SPECIFIC HEAT & DENSITY ASSUMED TO BE --1.0-- EXCEPT HOT WATER
5* DIMENSION PAR(10),XIN(10),OUT(10)
6* DIMENSION X(6),Y(6),GIN(3),XNTU(6),EFFN(6),PRR(5)
7* C
8* C--METRIC=U,BRITISH UNITS USED,----------JWRITE=1 WRITE ALL,JWRITE=0 NO WRITE
9* C KLBHN=J6PM FOR FLOW INPUT,-----KLBHR=1, LBS/HR INPUT
10* DATA METRIC/G/,YLRHR/l/,JWRITE/l/
11* DATA METRIC/?/, KLHR/1/, JWRITE/0/
12* DATA PDKG/4536/
13* DATA TFC1/32.,TFC2/1.8/
14* DATA CALBTU/3.96831/
15* C
16* C CONST1 & CONST4 ARE CONCENTRATION LIMITS
17* DATA CONST1/0.4/,CONST4/0.6/
18* C A-B-C-D-CONST ARE LIMITS FOR EVAP., ABSORP., COND., & GENERATOR
19* DATA ACONST/1.,BCONST/1.296,CCONST/1.423,DCONST/1.919/
20* DATA TEL0/2.22,TE2SET/4.43/
21* C COP LIMITS --HEAT LOSS FACTOR
22* DATA COPHI/0.93/,COPI/0.66/,FOG/1.0/
23* C EFFNX=0.71428 FOR T5=135 F EFFNX=(TG-T5)/(TG-TA)
24* DATA FFFNX/0.71428/
25* C
26* C
27* C
28* C
29* INPUT=1
30* CONTINUE
31* H1=C.
32* HS=C.
33* H2=C.
34* H8=C.
35* H10=O.
36* C
37* C -------UA VALUES ARE PER TON BASIS-------
38* UAG=456.981*PDKG
39* UAC=1011.869*PDKG
40* UAE=1503.294*PDKG
41* UAA=1102.8*PDKG
42* UAX=118.929*PDKG
43* UAX0=118.929*PDKG
44* GSO=144.077*PDKG
45* GWC=131.323*PDKG
46* GH0=150.786*PD00.*3.975
47* GEO=417.600 *PD00.
48* GCC=553.674*PD00.
49* TONC=174.
50* C
51* C
52* C
53* C
54* PRR(1)=UAG
55* PRR(2)=UAC
56* PRR(3)=UAE
57* PRR(4)=UAA
58* PRR(5)=UAX
59* C
60* JREF=0.
61* CONTINUE
62* TONX=XIN(7)/12000.
63* TOF=TONX*0.5
64* TONP=TONX*0.41
65* TONH=TONX*1.2
```

34
133* IF (TC1 LE TE2SET) S = 0.0
134* TE2 = TF1-0.05*HS
135* TH2 = TH1-0.05*HS
136* TC2 = TC1 + (1.0 + 0.05*HS)*HS
137* IF (S LE 0.001) GO TO 80
138* C
139* TE = TE1-(TE1-TE2)/EFFN(3)
140* TG = TH1-(TH1-TH2)/EFFN(1)
141* C ASSUMING A VALUE FOR TA
142* TA2 = (TC1+TC2)*0.5
143* TA = TC1-(TC1-TC2)/EFFN(4)
144* KTA = C
145* 15 CONTINUE
146* C IF (TA LE (TC1+1.0)) GO TO 43
147* TC = TC2-EFFN(2)-(1.0*EFFN(2)-1.0)*(TC1+EFFN(4)*(TA-TC1))
148* C
149* C
150* IF (TC LE TA) GO TO 40
151* IF (TE GE TA) GO TO 41
152* IF (TG GE TA) GO TO 41
153* C
154* X1 = (49.04 + 1.25*TA-TE)/(134.65 + 0.47*TA)
155* X4 = (49.04 + 1.25*TG-TC1)/(134.65 + 0.47*TG)
156* IF (X1 LT CONST1) GO TO 45
157* IF (X4 LE X1) GO TO 43
158* C
159* HF = TC-25.0
160* H1C = 572.8 + 0.41*TE
161* GR = 0.0/TH1-TH2
162* GS = GR*X4/(X4-X1)
163* GW = GS*(X1/X4)
164* C
165* CX1 = 1.01-1.23*X1+0.48*X1**2
166* CX4 = 1.01-1.23*X4+0.48*X4**2
167* GSC1 = GS*CX1
168* GWC4 = GW*CX4
169* CRATIO = GWC4/GSC1
170* C
171* C ASSUMING ORIGINAL FILM COEF. EQUAL ON BOTH SIDES.--GW ON SHELL SIDE
172* F1 = 2.0
173* F2 = 1.0
174* C F1 = 2.0, F2 = 1.0, HGS = HGW, --F1 = 2.5, F2 = 2.73, HGS = 1.5 HGW, FOR UAX
175* RCS = GS/GS*(TONO/TNX)**G.8
176* RGW = (GW/GW)*(TONO/TNX)**G.8
177* UAX = F1*UAX/(1.0/(RGS+F2*RGW))
178* C
179* IF (GWC4 GT GSC1) GO TO 17
180* XNTUX = UAX/GWC4*(TONO/TNX)
181* GO TO 18
182* C
183* 17 CONTINUE
184* XNTUX = UAX/GSC1*(TONO/TNX)
185* CRATIO = GSC1/GWC4
186* 18 CONTINUE
187* C
188* IF (ABS(1.0-CRATIO) GT 0.01) GO TO 13
189* IF (XNTUX*CRATIO GE 1.0) GO TO 12
190* EXPX = EXP1-XNTUX*(1.0-CRATIO)
191* EFFNX = (1.0-EXPX)/(1.0-CRATIO*EXPX)
192* GO TO 14
193* C
194* 12 CONTINUE
195* EFFNX = L.999
196* GO TO 14
197* C
198* EFFNX = XNTUX*(1.0-XNTUX)
199* 14 CONTINUE
TS=TF-EFFNX*(TG-TA)
T3=TA+EFFNX*CRATTO*(TG-TA)

H1=(45.81-425.92*X1+404.67*X1*X2)+CX1*TA
H5=(43.81-425.92*X4+404.67*X4*X5)+CX4*T5

H7=ST2.8+0.46*TF-0.043*TC

FOG=1.0, NO HEAT LOSS, --FOG>1.0, C<2.0,% HEAT LOSS
IF(FOG<LE.1.0) FOG=1.0

QG=(GW*HS-GS*H1*GP*H7)*FOG

QC=GR*(H7-H8)*(1.0*GQ/QG*(1.0-FOG))

QA=(GW*HS-GS*H1*GR*H1*(1.0*GQ/QG)+(1.0-FOG))

COP=GF/GQ

TC12=TC1+QA/GC

IF(FOG.LE.1.9) FOG=1.0

OC=GR9*(H7-H8)*1.0

QA=IGU*H5*G12*H1*GR9*H1*1.0

COP=OC/OF

TC14=TC1+RA/GC

IF(TC12.GE.TC14) TC12=TC1+1.0

IF(ABS(TC12-TA).LT.0.0001) GO TO 41

IF(KTA.EQ.0.5°) GO TO 41

TAS(TA)+0.5

KTA=KTA+1

GO TO 15

41 CONTINUE

IFAPS(CDPX-COP).LT.0.0001) GO TO 42

IF(COP.EQ.0.5) GO TO 1042

COPX=(COPX+COP)*0.5

42 CONTINUE

X(1)=TC2-TE

X(2)=TA-TC12

X(3)=TC-TC2

X(4)=TH2-TG

Y(1)=ACONST

Y(2)=PCONST

Y(3)=PCONST

Y(4)=PCONST

DO 47 I=1,4

IF(X(I).LT.Y(I)) AND KTONX.GE.0.1) GO TO 45

IF(X(I).LT.Y(I)) GO TO 45

44 CONTINUE

47 CONTINUE

IF(X1.GT.CONST1.AND.X4.LT.CONST4.AND.X4.GT.X1) GO TO 46

46 CONTINUE

IF(KTON2-100) 49,43,43

49 CONTINUE

TONM=TONX

TONX=(TONX+TONMAX)*0.5

IIX=FIX(1X)

TONX=FLOAT(1X)+1.0

KTON=KTON2+1

GO TO 11

41 CONTINUE

43 CONTINUE

IF(KTON1-100) 44,5C,5

44 CONTINUE

TONMAX=TONX

TONX=(TONX+TONMIN)*0.5
ITONX=IFIX(TONX)

IF(TONX.LE.1.0.OR.TONX.GT.(TONMIN+1.0)) GO TO 50

ITONX=FLOAT(ITONX)-1.0

KTON1=KTON1+1

GO TO 11

C

46 CONTINUE

IF(KTONX.EQ.1.0.OR.TONX.LE.0) GO TO 60

IF(TON.GE.TONX) GO TO 50

C CHECK MAX. STRONG SOLUTION PUMP RATE ---------------

GSTC=GSD*TON

GPSUMP=GS*TONX

IF(GSPUMP.GT.GSTG) GO TO 43

C

TST=TONX

IF(TON.LE.1.0. OR TE2.LE.TE1) GO TO 48

IF(TON.LE.1.0) S=0.0

GO TO 11

C

60 CONTINUE

C

C

C-----ALL SPEC HEAT = 1. ------------------

UAG=XNTU(1)*GH

UAC=XNTU(2)*GC

UAE=XNTU(3)*GE

UAA=XNTU(4)*GC

C

A=ALOG(1.0)

B=1555.0/(TF+273.15)

C=11.2414/((TE+273.15)**2)

PE=EXP(A*7.8553-B*C)

B=1555.0/(TC+273.15)

C=11.2414/((TC+273.15)**2)

PC=EXP(A*7.8553-B*C)

C

RG CONTINUE

QG=QG*CALBTU

QC=QC*CALBTU

OE=OE*CALBTU

OA=OA*CALBTU

H1=H1*CALBTU

H6=H6*CALBTU

H7=H7*CALBTU

H8=H8*CALBTU

H10=H10*CALBTU

H12=H12*CALBTU

C

T3=T3*TFCC+TFCC

T5=T5*TFCC+TFCC

C

TH1=TH1*TFCC+TFCC

TH2=TH2*TFCC+TFCC

TC1=TC1*TFCC+TFCC

TC12=TC12*TFCC+TFCC

TA2=TA2

TC2=TC2*TFCC+TFCC

TF1=TF1*TFCC+TFCC

TF2=TF2*TFCC+TFCC

37
X )

404 FORMAT(1X*4HA---,7F10.3,5E10.3/)
405 FORMAT(1X*4HE---,7F10.3,5E10.3/)
406 FORMAT(5X*12DH G T1 T2 T NTU
407 FORMAT(5X*12DH TA T5 T3 TG H1
408 FORMAT(5X*12DH H5 H7 H8 H10 NTUX EFFNX COP
409 FORMAT(5X*12DH X
410 WRITE (6,96)
411 WRITE (6,97) DT12E,DTE2,TE2,DT12A,DTA2,TA2,DT12C,DTC2,TC2,DT12G,DTG
413 X2,TH2
414 95 FORMAT(5X*12DH KTA KCOP KTON2 KTON1 GHT-GPM
415 X GCT-GPM GET-GPM TH1-F TC1-F TE1-F TON-CAL TON-ST
416 XART NO. J)
417 96 FORMAT(5X*12DH DT12E DTE2 TE2 DT12A DTA2
418 X TA2 DT12C DTC2 TC2 DT12G DTG2 TH2
419 X )
420 97 FORMAT(1X*4HTEMP,12F10.3///)
421 98 FORMAT(5X*4(I8,2X),8F10.3,3X,I3///)
422 99 FORMAT(7F10.1)
423 C
424 59 CONTINUE
425 5C3 CONTINUE
426 OUT(1)=TH2
427 OUT(2)=XIN(1)
428 OUT(3)=TON*OE
429 OUT(4)=TH2
430 OUT(5)=G1
431 OUT(6)=TF2
432 OUT(7)=XIN(3)
433 OUT(8)=TC2
434 OUT(9)=XIN(3)
435 OUT(10)=TON*OE
436 C
437 INPUT=1+INPUT
438 RRETURN
439 C STOP
440 C END
REFERENCES


Figure 1. - Flow diagram of single-stage LiBr–H₂O absorption unit.
A computer model of a LiBr-H₂O single-stage absorption machine has been developed. The model, utilizing a given set of design data such as water-flow rates and inlet or outlet temperatures of these flow rates but without knowing the interior characteristics of the machine (heat transfer rates and surface areas), can be used to predict or simulate off-design performance. Results from 130 off-design cases for a given commercial machine agree with the published data within 2 percent.

Absorption machine; LiBr-H₂O; Cooling unit; Air conditioning; Refrigeration; Solar cooling