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Conceptual Design and Performance Modeling of a 3KW Single Effect Lithium Bromide/Water Absorption Chiller Operating with Plate Heat Exchangers

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Abstract- This presents the design and performance modeling of a lithium bromide/ water absorption chiller operating with plate heat exchangers for the main components: generator, condenser, evaporator, absorber and solution heat exchanger. The design process involves an optimization procedure. Detailed mathematical equations representing the design of each of the system components are coded into a computer program using MATLAB programming language. An optimization program written in MATLAB was used to study the effect of varying some design parameters: plate thickness, plate width, channel spacing, number of channels on the overall heat transfer coefficient. A mathematical model was developed and programmed in MATLAB to predict the performance of the design. The optimum values of the parameters and system sizes obtained from the design optimization were used in the performance modeling. The performance was predicted as a function of varying external driving conditions. Results indicate that a coefficient of performance (COP) of 0.67 is attainable with hot water inlet temperature to the generator of 90^{o}C and 13^{o}C evaporator chilled water inlet temperature.

Keywords- *Absorption cooling, Design, Modelling, MATLAB, plate heat exchange*

I. INTRODUCTION

The energy demand for air conditioning has increased continuously in the last few decades especially in developing countries [3]. This increase is caused among other reasons by a rise in thermal load of air conditioned spaces to ensure occupants comfort, optimum performance of human and equipment in the space. Conventional vapour compression air conditioning units contribute to environmental degradation because of the refrigerants which they use (chloro fluoro carbon (CFCs) and hydro flouoro carbon (HFCs)) [4]. Also, these units consume a considerable amount of power owing to the electrically driven compressor. Absorption chillers using LiBr and water mixture are increasingly becoming acceptable alternatives to overcome these challenges. They use harmless refrigerants and are run using low grade and inexpensive energy such as waste heat and solar energy. However absorption chillers are mostly available in large capacities of hundreds of kilowatts, unsuitable for residential applications. The main components of these absorption chillers are tube bundle heat exchangers [4]. For absorption chillers with low capacities, tube bundle heat exchangers are not the right technology [4]. Over the years, low capacity, compact absorption chillers operating with plate heat exchangers are being developed. There has been extensive work in the study of absorption chillers in which selected components operate with plate heat exchangers. Reference [5] carried out an experimental study to investigate how absorption takes place in a plate heat exchanger under typical condition of absorption chillers driven by low temperature heat sources. Theoretical and experimental methods have been used to study the evaporation and condensation process in plate heat exchangers [6, 2 and 16]. Experiments have been conducted for the analysis of NH_3-H_2O absorption process in both falling film and bubble mode in a plate heat exchanger type absorber. It was found that the bubble mode is superior to the falling film mode [1].

Reference [7] modeled a vapour absorption refrigeration system in which the generator was replaced by a plate heat exchanger. Reference [8] carried out a theoretical performance study of a LiBr – water absorption chiller in which the generator, condenser and solution heat exchanger are plate heat exchangers. There is however limited information on performance prediction of absorption chiller unit operating with plate heat exchangers for all major components. In this work, we will carry out a conceptual design and performance modeling of a single effect LiBr /water absorption chiller in which all major components (generator, condenser, absorber, evaporator, solution heat exchanger (SHX)) are plate heat exchangers. The design stage involves an optimization procedure carried out in MATLAB. Energy balance and heat transfer equations are employed in the performance modeling. The performance will be predicted as a function of external driving conditions: Hot water inlet temperature at the generator, cooling water inlet temperature at the absorber and condenser, chilled water inlet temperature at the evaporator.

II. SYSTEM DESCRIPTION

The main components of an absorption chiller are the generator: (G), condenser: (C), evaporator: (E), absorber: (A) and solution heat exchanger: (SHX), as seen in fig. 1.

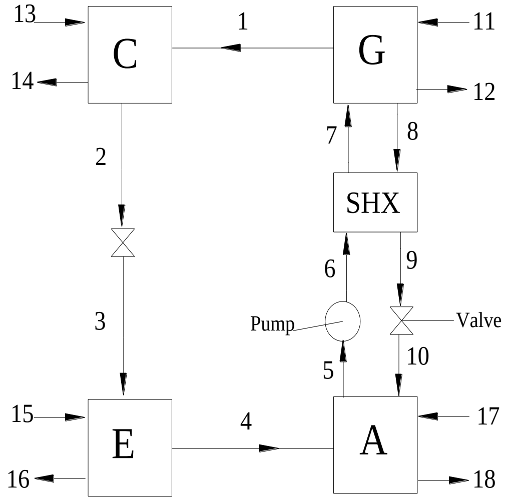

Figure 1. Schematic of a single effect absorption cooling cycle

The system description presented here is done using the state points as seen in fig. 1. Hot water flows from an external heat source (11/12) to the generator. In the generator exists a weak solution (solution with high concentration of refrigerant) which is made up of the absorbent (LiBr) and the refrigerant (water). Due to heat transfer from the hot water to the weak solution, the refrigerant boils off from the weak solution and the vapour flows to the condenser (1). The remaining strong solution (solution with high concentration of absorbent) flows to the absorber through the solution heat exchanger and valve (8, 9, 10). In the condenser, under condensing temperature and pressure, the refrigerant vapour condenses. The heat of condensation is transferred to the cooling water which flows through the condenser and absorber (13/14 and 17/18). The condensate (liquid refrigerant) flows through the valve to the evaporator (2, 3). In the evaporator, the refrigerant liquid takes away heat from the chilling water (15/16) and evaporates again under evaporator temperature and pressure. The refrigerant vapour streams to the absorber (4) where it is absorbed by the strong solution. The resulting weak solution is then pumped via the solution pump, through the solution heat exchanger, back to the generator (5, 6, 7) and the cycle repeats again.

A. System Design Concept

The channel is the space formed between two plates, separated by a gasket that runs round the circumference of the plates. The process fluids flow inside the channels and heat transfer is carried out through the plates. Hot and cold fluids flow through alternate channels and are distributed by flow pipes. The concept is to have generator and condenser plates integrated, evaporator and absorber plates integrated. In the generator, hot water flows in alternating channels, while heating up the weak aqueous lithium bromide solution flowing in pool mode in the channels between. In the condenser, cooling water flows in alternating channels, while steam from the generator condenses in the channels between. In the evaporator, chilled water from the building flows in alternating channels, losing heat to the liquid refrigerant which flows in falling film mode in the channels between, thereby vaporizing the refrigerant. In the absorber, cooling water flows in alternating channels, while strong solution of aqueous lithium bromide flows in falling film mode in the channels between, absorbing water vapour from the evaporator. In the solution heat exchanger, strong solution and weak solution of aqueous lithium bromide flow in alternate channels. A cut – away section showing three channels of a plate heat exchanger is shown in fig. 2.

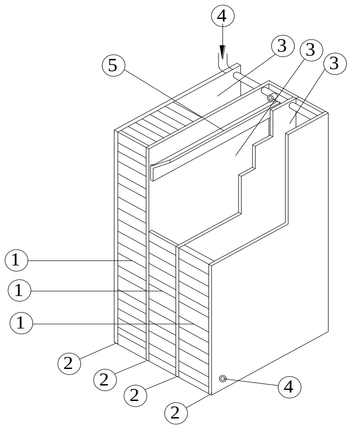

Figure 2. Plate heat exchanger

B. System Design Specifications

It is desired to have the chiller operate with low hot water temperatures obtainable from flat plate collectors, also, care is taken to avoid risk of chiller operating in the crystallization range. For these reasons, the operating temperatures at inlet and outlet to the generator are set at 90^oC and 80^oC respectively. The vapour leaving the generator into the condenser is considered as pure water vapour at generator pressure and temperature of 90^oC.

Based on an available cooling water temperature of 29^oC and the operating temperatures at the generator, a design condenser temperature of 40^oC and absorber temperature of 40^oC have been used. With outlet temperatures of 35^oC and 37^oC set at the condenser and absorber respectively.

The lowest operating refrigerant temperature attainable for lithium bromide water pair, in which water is used as refrigerant is about 6^oC [9]. This is due to the freezing point of water. For this reason, the design inlet and outlet refrigerant temperature to the evaporator have been set at 6^oC and 8^oC respectively.

The design chilled water inlet and outlet temperatures to the evaporator are set at 15°C and 10°C respectively.

Starting with a condenser and evaporator temperature of 40°C and 8°C respectively, we obtain from table of water vapour pressures, the condenser pressure (P_c) and evaporator pressure (P_e) corresponding to the respective temperatures:

$$P_c = 7.34KPa, \qquad P_e = 1.072KPa$$

The condenser pressure equals the generator pressure (P_g) and the evaporator pressure equals the absorber pressure (P_a):

$$P_c = P_g = 7.34KPa \text{ and } P_e = P_a = 1.072KPa$$

With these pressures and temperatures, using the Equilibrium chart for aqueous lithium bromide solution, the design strong solution and weak solution concentrations are obtained as $(X_{ss}) = 0.63$ and $(X_{ws}) = 0.55$ respectively. The remaining design temperatures are obtained as well. The design temperatures corresponding to each state point are shown in table 1.

TABLE I. TEMPERATURES OF STATE POINTS

Point	Design Temp. (°C)
1	90
2	40
3	6
4	8
5	40
6	40
7	74
8	81
9	55
10	55
11	90
12	80
13	29
14	35
15	15
16	10
17	29
18	37

III. SYSTEM DESIGN THEORY

Reference is made to the point numberings as in fig.1. For a cooling capacity of 3KW and desired coefficient of performance (COP) of 0.7

A. Calculation of system loads

The system loads are calculated according to [10] as follows:

Generator:

$$Q_g = \frac{Q_e}{COP} \tag{1}$$

Condenser:

$$Q_c = \frac{Q_e(h_1 - h_2)}{(h_4 - h_3)} \tag{2}$$

Absorber:

$$Q_a = Q_g + Q_e - Q_c \tag{3}$$

Solution heat exchanger

$$Q_{SHX} = \dot{m}_{ws}(h_7 - h_6) \tag{4}$$

B. Calculation of mass flow rates

The mass flow rates of the refrigerant, weak solution and strong solution are calculated according to [10] respectively as:

$$\dot{m}_{ref} = \frac{Q_e}{h_4 - h_3} \tag{5}$$

$$\dot{m}_{ws} = \dot{m}_{ref}\left(\frac{X_{ss}}{X_{ss} - X_{ws}}\right) \tag{6}$$

$$\dot{m}_{ss} = \dot{m}_{ws} - \dot{m}_{ref} \tag{7}$$

The mass flow rates of the hot water at the generator, cooling water at the condenser and absorber, chilled water at the evaporator are calculated using their respective loads, fluid temperature differentials and specific heat capacities as:

$$\dot{m} = \frac{Q}{C_p \Delta T} \tag{8}$$

Where C_p = specific heat capacity of the fluid, ΔT = temperature differential

C. Calculation of Logarithmic Mean Temperature Difference (LMTD)

The Logarithmic Mean Temperature Difference is calculated as:

$$LMTD = \frac{\theta_1 - \theta_2}{ln\left[\frac{\theta_1}{\theta_2}\right]} \tag{9}$$

Where:

Generator: $\theta_1 = (T_{11} - T_8)$ and $\theta_2 = (T_{12} - T_7)$

Condenser: Considering the condensing part only, according to [11]

$\theta_1 = (T_2 - T_{13})$ and $\theta_2 = (T_2 - T_{14})$

Evaporator: $\theta_1 = (T_{15} - T_4)$ and $\theta_2 = (T_{16} - T_3)$

Absorber: $\theta_1 = (T_{10} - T_{18})$ and $\theta_2 = (T_5 - T_{17})$

SHX: $\theta_1 = (T_8 - T_7)$ and $\theta_2 = (T_9 - T_6)$

D. Calculation of hydraulic diameter

The hydraulic diameter (D_h) is calculated as in [12]

$$D_h = 2 \times b \tag{10}$$

E. Calculation of flow cross sectional area

The space between two parallel plates in which either hot or cold fluid flows is referred to as a channel. Flow cross sectional area for a single channel (Ax) is calculated as in [12]

$$Ax = b \times w \tag{11}$$

F. Calculation of mass velocity and flow velocity

The mass velocity (G) and flow velocity (v) for the flows in the hot fluid channels and cold fluid channels for the generator, condenser, absorber, evaporator and SHX are calculated according to [12] respectively as:

$$G = \frac{\dot{m}_f}{nAx} \quad (12)$$

$$v = \frac{G}{\rho_f} \quad (13)$$

Where the densities of the aqueous lithium bromide solution in the generator, absorber and SHX are computed according to [14]. The densities of water flows in the channels are obtained from [15] using the mean temperatures at inlet and outlet to the channels.

G. Calculation of Reynolds number

The Reynolds numbers (Re) for flows in the hot and cold fluid side channels for the generator, condenser, absorber, evaporator and SHX are calculated using the expression [15].

$$Re = \frac{\rho_f v D_h}{\mu_f} \quad (14)$$

Where the dynamic viscosities of the aqueous lithium bromide solutions at the generator, absorber and SHX are calculated using the empirical relations of [14]. The dynamic viscosities of water flows in the channels are obtained from [15] using the mean temperatures at inlet and outlet to the channels.

H. Calculation of Prandtl number

The Prandtl numbers (Pr) of flows in the hot and cold fluid side channels for the generator, condenser, absorber, evaporator and SHX are calculated according to [15] as:

$$Pr = \frac{C_p \mu_f}{k_f} \quad (15)$$

The thermal conductivities of the aqueous lithium bromide solutions at the generator, absorber and SHX are obtained as in [13].

The specific heat capacities of the aqueous lithium bromide solutions at the generator, absorber and SHX are calculated using the relation of [19]. The specific heat capacities and thermal conductivities of water flows in the channels are obtained from [15] using the mean temperatures at inlet and outlet to the channels.

I. Calculation of heat transfer coefficients

Heat transfer coefficients are computed using empirical correlations obtained from literature.

The water sides of the generator, absorber, condenser and evaporator, as well as both sides of the solution heat exchanger, are all single phase flows. Therefore the correlation for heat transfer coefficient of single phase flow in rectangular cross section channel proposed by [16] is used. Where the empirical constants 0.2 and 0.7 are for plates with chevron angle at 60^o.

$$h_f = \frac{k_f}{D_h} 0.2 Re^{0.7} Pr^{\frac{1}{3}} \quad (16)$$

For boiling heat transfer coefficients at the generator and evaporator, correlation proposed by [2] is used.

$$h_b = \frac{k_f}{D_h} c_2 Re^{0.7} Pr^{\frac{1}{3}} (c_3 Bo^{0.5}) \quad (17)$$

Where:

$c_2 = 0.2 \; and c_3 = 88$ are empirical constants

$$Bo = \frac{q}{\lambda_{sol} G_{sol}} \quad (18)$$

At the generator,

$$q_g = \frac{k_p (T_{11} - T_7)}{\delta} \quad (19)$$

At the evaporator,

$$q_e = \frac{k_p (T_{15} - T_3)}{\delta} \quad (20)$$

Where the film thickness is given as [17]:

$$\delta = \left(\frac{3\mu\Gamma}{\rho^2 g}\right)^{\frac{1}{3}} \quad (21)$$

The falling film flow rate is given as [17]:

$$\Gamma = \frac{\dot{m}_{sol}}{2w} \quad (22)$$

For the heat transfer coefficient at the condenser refrigerant side, the correlation of Nusselt for film condensation on vertical plates as given in [11] is used

$$h_{c,ref} = 0.943 \left[\frac{\rho_l (\rho_l - \rho_v) g h'_{fg} k_p^3}{4\mu (T_{sat} - T_w)}\right]^{\frac{1}{4}} \quad (23)$$

Where:

$$h'_{fg} = h_{fg} \left[1 + 0.68 \frac{C_p (T_{sat} - T_w)}{h_{fg}}\right] \quad (24)$$

Where h_{fg}= enthalpy of vaporization

For heat transfer coefficient at the absorber solution side, correlation of [18] is used.

$$h_{a,sol} = \frac{k}{\delta} (0.029 Re^{0.53} Pr^{0.344}) \quad (25)$$

J. Calculation of overall heat transfer coefficient

The overall heat transfer coefficient for each of the heat exchangers is calculated using the following relation by [12]:

$$U = \frac{1}{\frac{1}{h_h} + \frac{t_p}{k_p} + \frac{1}{h_c}} \quad (26)$$

K. Calculation of total effective surface area for heat transfer

The total effective surface area for heat transfer for each of the heat exchangers is calculated as:

$$A_T = \frac{Q}{U \times LMTD} \quad (27)$$

L. Calculation of effective plate area

The effective plate area (area of an individual plate for heat transfer) for each of the heat exchangers is calculated as [12]

$$A_p = \frac{A_T}{2(N-2)} \quad (28)$$

M. Calculation of plate length

The required plate length for each of the heat exchangers is calculated as:

$$L_p = \frac{A_p}{w} \quad (29)$$

IV. DESIGN CALCULATIONS AND OPTIMIZATION PROCEDURE

To determine the design parameters and component sizes for the generator, condenser, absorber, evaporator and SHX required producing a cooling power of 3kW at the evaporator, equations 1 to29 were coded using MATLAB programming language according to the design sequence calculations. The design specifications served as inputs to these calculations. To optimize the absorption chiller design, an optimization program was written in MATLAB programming language. This was used in studying the effect of varying some of the design parameters (plate thickness, plate spacing, plate width, number of channels) on the objective function (the overall heat transfer coefficient). This was done for each of the heat exchangers. The optimization procedure is not shown in this publication. The final values of parameters and system component sizes obtained from the design optimization are shown in table 2.

V. PERFORMANCE MODELLING

A mathematical model was developed to evaluate the performance of the absorption chiller. An energy balance equation and heat transfer equation were developed for each of the heat exchangers. The values for the overall heat transfer coefficient (U values) and the heat transfer surface area (A values) of each of the heat exchangers obtained from the optimized design (as seen in table 2.) were used to evaluate the UA values. The values of mass flow rates as shown in table 2 are used in the model equations. The pair of equations for each of the heat exchangers were programmed and solved simultaneously in MATLAB. The model uses as inputs, known stream temperatures: hot water inlet temperature at generator, cooling water inlet temperature and chilled water inlet temperature at the evaporator, to calculate unknown stream temperatures: hot water outlet temperature, cooling water outlet temperature, chilled water outlet temperature, solutions temperatures, COP. This it does through an iterative process.

The assumptions made in the model development are:

i. Steady state conditions.

ii. Constant UA values for the heat exchangers.

iii. The pressure in the generator is equal to that in the condenser, while the pressure in the absorber is equal to that in the evaporator.

iv. Pressure changes occur only in the expansion valves and in the pump.

v. No heat losses to the environment.

vi. The refrigerant leaves the condenser and evaporator as saturated liquid and saturated vapour respectively.

vii. Steam leaving the generator is at superheated state.

viii. Weak solution leaving the absorber and strong solution leaving the generator are in saturated state.

The following relations were used in the equations development, using the point numberings in fig. 1.

$\dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4 = \dot{m}_{ref} =$ Mass flow rate of refrigerant.

$\dot{m}_5 = \dot{m}_6 = \dot{m}_7 = \dot{m}_{ws} =$ Mass flow rate of weak solution.

$\dot{m}_{11} = \dot{m}_{12} = \dot{m}_g$ Mass flow rate of hot water at the generator

$\dot{m}_8 = \dot{m}_9 = \dot{m}_{10} = \dot{m}_{ss} =$ Mass flow rate of strong solution.

$\dot{m}_{17} = \dot{m}_{18} = \dot{m}_a =$ Mass flow rate of cooling water at absorber

$\dot{m}_{15} = \dot{m}_{16} = \dot{m}_e =$ Mass flow rate of chilled water at evaporator

$\dot{m}_{13} = \dot{m}_{14} = \dot{m}_c =$ Mass flow rate of cooling water at condenser

The point numberings as in fig. 1 are used in the energy balance equations and heat transfer equations.

A. Generator

1) Energy Balance Equation:

$$\dot{m}_g h_{11} - \dot{m}_g h_{12} - \dot{m}_{ref} h_1 + \dot{m}_{ws} h_7 - \dot{m}_{ss} h_8 = 0 \quad (30)$$

2) Heat transfer equation:

$$\dot{m}_g C_p (T_{11} - T_{12}) = U_g . A_{Tg} \left[\frac{(T_{11} - T_8) - (T_{12} - T_7)}{ln\left[\frac{(T_{11} - T_8)}{(T_{12} - T_7)}\right]} \right] \quad (31)$$

B. Condenser

1) Energy Balance Equation:

$$\dot{m}_{ref} h_1 + \dot{m}_c h_{13} - \dot{m}_{ref} h_2 - \dot{m}_c h_{14} = 0 \quad (32)$$

2) Heat transfer equation:

By considering the condensing part only according to [11], the heat transfer equation is:

$$\dot{m}_c C_p (T_{14} - T_{13}) = U_c A_{Tc} \left[\frac{(T_2 - T_{13}) - (T_2 - T_{14})}{ln\left[\frac{(T_2 - T_{13})}{(T_2 - T_{14})}\right]} \right] \quad (33)$$

C. Evaporator

1) Energy Balance Equation:

$$\dot{m}_{ref} h_3 + \dot{m}_e h_{15} - \dot{m}_{ref} h_4 - \dot{m}_e h_{16} = 0 \quad (34)$$

2) Heat transfer equation:

$$\dot{m}_e C_p (T_{15} - T_{16}) = U_e A_{Te} \left[\frac{(T_{15} - T_4) - (T_{16} - T_3)}{ln\left[\frac{(T_{15} - T_4)}{(T_{16} - T_3)}\right]} \right] \quad (35)$$

D. Absorber

1) Energy Balance Equation:

$$-\dot{m}_a h_{18} + \dot{m}_a h_{17} + \dot{m}_{ss} h_{10} - \dot{m}_{ws} h_5 + \dot{m}_{ref} h_4 = 0 \quad (36)$$

2) Heat transfer equation:

$$\dot{m}_a C_p(T_{18} - T_{17}) = U_a . A_{Ta} \left[\frac{(T_{10}-T_{18})-(T_5-T_{17})}{ln\left[\frac{(T_{10}-T_{18})}{(T_5-T_{17})}\right]} \right] \quad (37)$$

E. SHX

1) Energy balance equation:

$$\dot{m}_{ss}h_8 - \dot{m}_{ws}h_7 + \dot{m}_{ws}h_6 - \dot{m}_{ss}h_9 = 0 \quad (38)$$

2) Heat transfer equation

$$\dot{m}_{ws} C_p(T_7 - T_6) = U_{shx} . A_{Tshx} \left[\frac{(T_8-T_7)-(T_9-T_6)}{ln\left[\frac{(T_8-T_7)}{(T_9-T_6)}\right]} \right] \quad (39)$$

$$COP = \frac{Q_e}{Q_g} \quad (40)$$

The iterative process starts at the generator, with an initial guess of the strong solution outlet temperature from the generator(T_8). The energy equation and heat transfer equation are solved simultaneously in matrix form to obtain the unknown stream temperatures. The pair of equations for the other components is solved in a similar manner. The process continues until convergence is attained. An efficiency of 0.8 is assumed for the SHX. The enthalpies of the strong solution and weak solution as well as their respective specific heat capacities are calculated according to the method of [19].

VI. RESULTS AND DISCUSSION

Table 2 below shows the final values of parameters and system component sizes obtained from the system design optimization results. The values obtained are used in the performance modeling and simulation

The performance model developed in section VI was programmed and simulated in MATLAB. The set of conditions for which the model equations have been solved are: hot water inlet temperature ranging from 40°C to 95°C, cooling water inlet temperature ranging from 24°C to 40°C, evaporator temperature ranging from 9°C to 13°C. Figures 3 to 8 are graphs that show the performance of the system.

From fig. 3 the generator heat load is seen to increase with increasing hot water inlet temperature to the generator. A generator heat load of 3.5KW is attainable with 90°C hot water inlet temperature.

TABLE II. SYSTEM OPTIMUM DESIGN PARAMETERS AND COMPONENT SIZES

Parameter	Description	Value
Q_e	Evaporator cooling power (KW)	3
Q_g	Generator load (KW)	4.28
Q_c	Condenser load (KW)	3.19
Q_a	Absorber load (KW)	4.09
Q_{shx}	SHX load (KW)	0.684
U_g	Overall heat transfer coefficient at generator (KW/m^{2}K)	799.05
U_c	Overall heat transfer coefficient at condenser (KW/m^{2}K)	508.97
U_a	Overall heat transfer coefficient at absorber (KW/m^{2}K)	374.96
U_e	Overall heat transfer coefficient at evaporator (KW/m^{2}K)	578.58
U_{shx}	Overall heat transfer coefficient at SHX (KW/m^{2}K)	189.32
A_{Tg}	Total heat transfer area at generator (m^2)	0.7249
A_{Tc}	Total heat transfer area at condenser (m^2)	1.0185
A_{Ta}	Total heat transfer area at absorber (m^2)	0.7681
A_{Te}	Total heat transfer area at evaporator (m^2)	0.9672
A_{Tshx}	Total heat transfer area at SHX (m^2)	0.3445
L_{pg}	Length of plate for generator (m)	0.1908
L_{pc}	Length of plate for condenser (m)	0.268
L_{pa}	Length of plate for absorber (m)	0.2021
L_{pe}	Length of plate for evaporator (m)	0.2545
L_{pshx}	Length of plate for SHX (m)	0.0907
$\dot{m}_{gh}$	Mass flow rate of hot water at generator (Kg/s)	0.1024
$\dot{m}_{cw,a}$	Mass flow rate of cooling water at absorber (Kg/s)	0.1222
$\dot{m}_{cw,c}$	Mass flow rate of cooling water at condenser (Kg/s)	0.0953
$\dot{m}_{chw}$	Mass flow rate of chilled water at evaporator (Kg/s)	0.1433
$\dot{m}_{ss}$	Mass flow rate of strong solution (Kg/s)	0.008
$\dot{m}_{ws}$	Mass flow rate of weak solution (Kg/s)	0.0101
$\dot{m}_{ref}$	Mass flow rate of refrigerant (Kg/s)	0.0013
t_p	Thickness of plate (m)	0.0015
w_p	Width of plate (m)	0.1
b	Channel spacing (m)	0.008
n	Number of channels	20

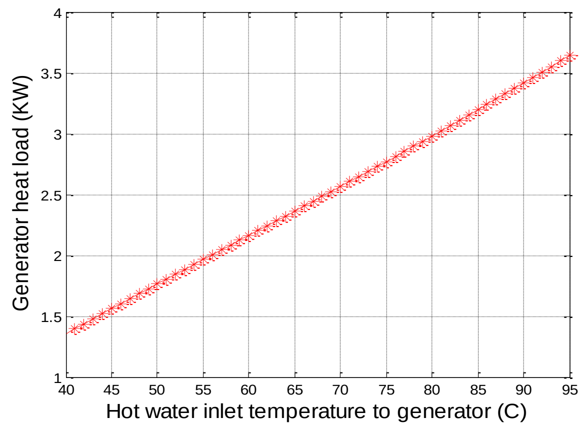

Figure 3. Variation of generator heat load with hot water inlet temperature to generator

From fig. 4, the lithium bromide concentration in solution at exit from the generator is seen to increase with increasing hot water inlet temperature to the generator. At hot water inlet temperatures in excess of 95°C, the concentration of lithium bromide in solution is seen to exceed 0.65, which shifts the chiller operation into the crystallization zone, this should be avoided. At hot water inlet temperature of 90°C, the concentration of lithium bromide in solution exceeds 0.65 for condenser temperatures of 30°C and 35°C. The concentration is however below 0.65 for condenser temperature of 40°C. This makes the condenser temperature of 40°C safe for chiller operation outside the crystallization zone.

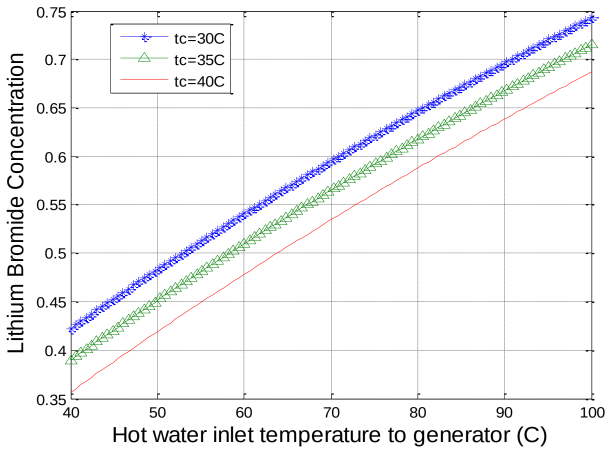

Figure 4. Variation of lithium bromide concentration at exit from generator with hot water inlet temperature to generator at condenser temperatures of 30°C, 35°C and 40°C

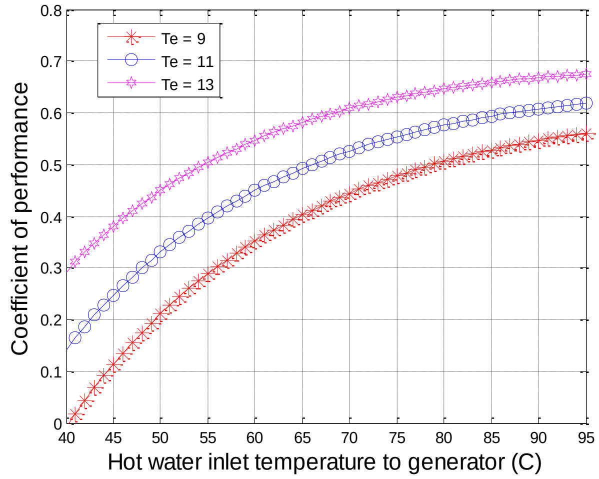

Figure 6. Variation of COP with hot water inlet temperature to generator at evaporator chilled water inlet temperatures of 9°C, 11°C and 13°C. (Condenser temperature Tc = 40°C)

From fig. 5 the evaporator cooling power increases with increasing hot water inlet temperature to the generator. Also, the evaporator cooling power increases with increasing chilled water inlet temperature to the evaporator. The cooling power can be seen to be highly dependent on the hot water inlet temperature at the generator. A cooling power of 2.5KW is attainable with 90°C hot water inlet temperature at 13°C chilled water inlet temperature.

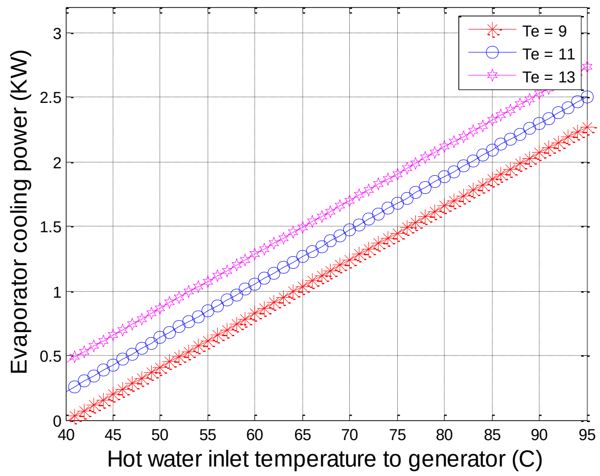

Figure 5. Variation of evaporator cooling power with hot water inlet temperature to generator at evaporator chilled water inlet temperatures of 9°C, 11°C and 13°C. (Condenser temperature Tc = 40°C)

From figure 6, the COP increases with increasing hot water inlet temperature, it peaks at about 90°C hot water inlet temperature. Also the COP increases with increasing chilled water inlet temperature to the evaporator. A COP of 0.67 is attainable with 90°C hot water inlet temperature at 13°C chilled water inlet temperature.

From fig. 7, the evaporator cooling power is seen to decrease as the cooling water inlet temperature increases. This implies that the chiller should be operated with cooling water temperature below the condenser temperature.

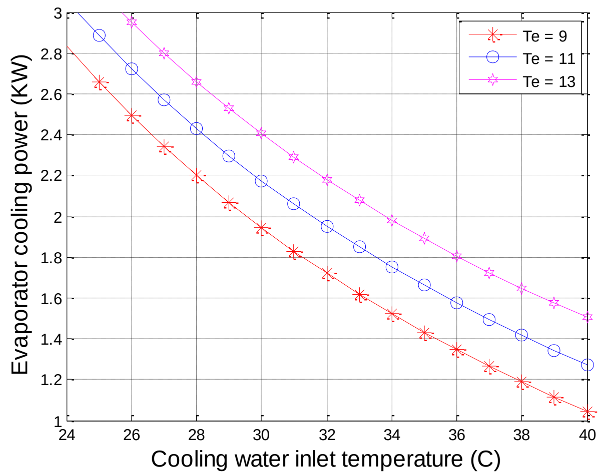

Figure 7. Variation of evaporator cooling power with cooling water inlet temperature, at evaporator chilled water inlet temperatures of 9°C, 11°C and 13°C. (Condenser temperature Tc = 40°C)

From fig. 8, the COP is seen to decrease with increasing cooling water inlet temperature. It can be seen that the performance of the chiller is highly dependent on the cooling water inlet temperature as well as the hot water inlet temperature.

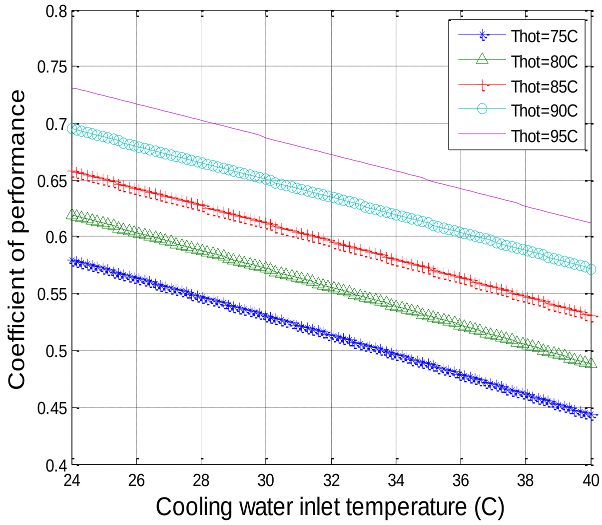

Figure 8. Variation of coefficient of performance with cooling water inlet temperature, at hot water inlet temperatures of 75°C, 80°C, 85°C, 90°C and 95°C. (Condenser temperature Tc = 40°C)

VII. CONCLUSION

A conceptual design of a 3KW single effect LiBr/H_2O absorption chiller operating with plate heat exchangers has been carried out. Outputs from the optimized design parameters have been used in a developed model to evaluate the performance of the chiller. Results indicate that the chiller is capable of operating with a coefficient of performance of 0.67 at 90°C hot water inlet temperature, 29°C cooling water temperature, 13°C evaporator temperature and producing a cooling power of 2.5KW.

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NOMENCLATURE

A_p	Effective area of single plate (m^2)
A_T	Total effective surface area for heat transfer (m^2)
Ax	Flow cross sectional area for a single channel (m^2)
Bo	Boiling number
b	Channel spacing (m)
C_p	Specific heat capacity (KJ/KgK)
D_h	Hydraulic diameter (m)
G	Mass velocity (Kg/m^2s)
g	Acceleration due to gravity (m/sec^2)
h	Enthalpy, Heat transfer coefficient (KJ/Kg, W/m^2K)
k	Thermal conductivity (W/m°C)
L	Plate length (m)
$\dot{m}$	Mass flow rate (Kg/s)
N	Sum of hot and cold fluid channels
n	Number of channels for hot or cold fluid
P	Pressure (N/m^2)
Q	Load (KW)
q	Heat flux (W/m^2)
T	Temperature (°C)
t	Thickness (m)
v	Flow velocity (m/s)
w	Plate width (m)
X	Concentration

Subscripts

a	Absorber
c	Condenser, cold
chw	Chilled water
cw	Cooling water
e	Evaporator
f	Fluid
g	Generator
h	Hot
hw	Hot water
l	Liquid
p	Plate
ref	Refrigerant
sat	Saturation
shx	Solution heat exchanger
sol	Solution
w	Wall
ws	Weak solution
ss	Strong solution

1, 2, 3……18 Point numberings

Greek

μ	Dynamic viscosity	(kg/ms)
ρ	Density	(kg/m^3)
δ	Liquid film thickness	(m)
Γ	Falling film flow rate	(Kg/ms)
U	Overall heat transfer coefficient	(KW/m^2k)
λ	Latent heat	(KJ/Kg)