Design and Construction of a Lithium Bromide Water Absorption Refrigerator

SOTERIS KALOGIROU, *Department of Mechanical Engineering, Higher Technical Institute, P. O. Box 20423, Nicosia 2152, Cyprus, Tel: +357-2-306266, Fax: +357-2-494953, Email: skalogir@spidernet.com.cy*
GEORGE FLORIDES, *Department of Mechanical Engineering, Higher Technical Institute, Cyprus*
SAVVAS TASSOU, *Mechanical Engineering Department, Brunel University, Uxbridge, Middlesex, UB8 3PH, UK.*
LOUIS WROBEL, *Mechanical Engineering Department, Brunel University, Uxbridge, Middlesex, UB8 3PH, UK.*

ABSTRACT

The objective of this work is to design and construct a lithium bromide–water (LiBr-H2O) absorption refrigerator with a nominal capacity of 1 kW. Absorption refrigerators are machines, which produce cooling by using heat energy, and have no moving parts. The various stages of design are presented including the design of the evaporator, absorber, solution heat exchanger, generator and condenser. The major problem faced during the design stage was the calculation of the heat transfer coefficient (U-value) of the various components. Single-pass vertical-tube heat exchangers have been used for the absorber and for the evaporator. The solution heat exchanger was designed as a single-pass annulus heat exchanger. The condenser and the generator were designed using horizontal tube heat exchangers. The condenser handles pure water vapour and adequate equations exist for the determination of the U-value. A pool-boiling generator has been employed and its U-value was estimated experimentally as published work on this area is limited.

1. INTRODUCTION

Absorption machines are thermally activated and for this reason, large amount of input (shaft) power is not required. In this way, where power is expensive or unavailable, or where there is waste, gas, geothermal or solar heat available, absorption machines provide reliable and quiet cooling (ASHRAE, 1997).

A number of refrigerant-absorbent pairs are used, for which the most common ones are water-lithium bromide and ammonia-water. These two pairs offer good thermodynamic performance and they are environmentally benign.

Lithium bromide-water chillers are available in two types, the single and the double effect. The single effect absorption chiller is mainly used for building cooling loads, where chilled water is required at 6-7^{o}C. The coefficient of performance (COP) varies to a small extent (0.65-0.75) with the heat source and the cooling water temperatures. Single effect chillers can operate with hot water temperature ranging from about 80^{o}C to 120^{o}C when water is pressurised.

The double effect absorption chiller has two stages of generation to separate the refrigerant from the absorbent. Thus the temperature of the heat source needed to drive the high-stage generator is essentially higher than that needed for the single-effect machine and is in the range of 150 to 200 °C. Double effect chillers have a higher COP of about 0.9-1.2 (Dorgan *et al.,* 1995). Although double effect chillers are more efficient than the single-effect machines they are obviously more expensive to purchase. However, every individual application must be considered on its merits since the resulting savings in capital cost of the single-effect units can largely offset the extra capital cost of the double effect chiller.

The objective of this paper is to design a small LiBr-water absorption machine and evaluate its performance characteristics.

1.1 Single Effect Lithium Bromide (LiBr) - Water Cooling

The single effect absorption technology provides a peak cooling coefficient of performance (COP) of approximately 0.7 and operates with heat input temperatures in the range of 80°C to 120°C. It should be noted that the refrigerant in the water-lithium bromide system is water and the LiBr acts as the absorbent, which absorbs the water vapour thus making pumping from the absorber to the generator easier and economic. A single-effect, two shell, LiBr - water chiller is illustrated in Figure 1.

With reference to the numbering system shown in Figure 1, at point (1) the solution is rich in refrigerant and a pump forces the liquid through a heat exchanger to the generator (3). The temperature of the solution in the heat exchanger is increased.

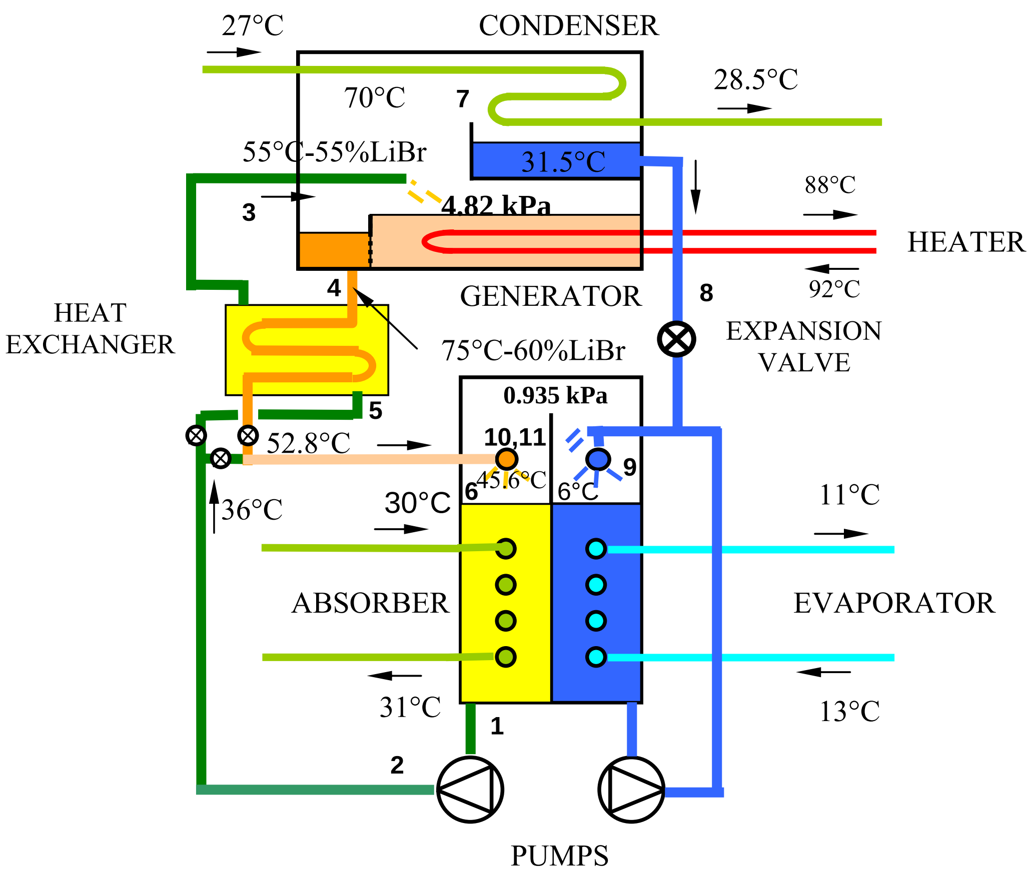

Figure 1. Schematic representation of the system.

In the generator thermal energy is added and refrigerant boils off the solution. The refrigerant vapour (7) flows to the condenser, where heat is rejected as the refrigerant condenses. The condensed liquid (8) flows through a flow restrictor to the evaporator (9). In the evaporator, the heat from the load evaporates the refrigerant, which flows back to the absorber (10). A small portion of the refrigerant leaves the evaporator as liquid spillover (11). At the generator exit (4), the steam consists of absorbent-refrigerant solution, which is cooled in the heat exchanger. From points (6) to (1), the solution absorbs refrigerant vapour from the evaporator and rejects heat through a heat exchanger.

2. DESIGN OF A SINGLE-EFFECT LITHIUM BROMIDE – WATER SYSTEM

To perform estimations of equipment sizing and performance evaluation of single-effect water-lithium bromide absorption cooler basic assumptions and input values must be considered. With reference to Figure 1, the basic assumptions are: (1) the steady state refrigerant is pure water, (2) there are no pressure changes except through the flow restrictors and the pump, (3) at points 1, 4, 8 and 11 there is only saturated liquid, (4) at point 10 there is only saturated vapour, (5) flow restrictors are adiabatic, (6) the pump is isentropic, and (7) there are no jacket heat losses.

The objective is to design and construct a small pilot system of 1 kW of cooling power. The method of design is demonstrated in ASHRAE (1997) and need not be repeated here. The load and characteristics required by each component of the unit is calculated using a computer program written for this purpose. Mathematical curve fitting equations have been used for the estimation of the various properties of the LiBr-H_2O solution and water.

The performance characteristics of the 1 kW system, operated under a number of generator temperatures were considered but since in Cyprus solar heat is in abundance and is collected with flat plate collectors working at low temperatures, a low generator temperature is preferred. The system on which the design is based is that presented in Table 1. This system operates with a generator temperature of 75 °C, which can easily be reached and has a good COP (0.74). Lower generator temperatures cannot be used since by lowering this temperature the condenser exit temperature is lowered too. In the case of this study the condenser exit temperature is 31.5°C (point 8). By lowering the generator temperature to 67°C the condenser exit temperature becomes 25°C, which cannot be reached. This is due to the fact that the temperature of the external sink which cools the condenser (in this case the water of a well) is about 27°C. A number of other designs have been tried but the one presented here is the most suitable for the environmental conditions of Cyprus.

3. SYSTEM HEAT EXCHANGERS SIZING

In single-pass heat exchangers, the temperature difference ΔT between the hot and the cold fluids is not constant but it varies with distance along the heat exchanger. In the heat transfer analysis, it is convenient to establish a mean temperature difference (ΔT_m) between the hot and cold fluids such that the total heat transfer rate $\dot{Q}$ between the fluids can be determined from the following expression:

$$\dot{Q} = AU\Delta T_m \tag{1}$$

where A (m^2) is the total heat transfer area and U (W/m^2-°C) is the average overall heat transfer coefficient, based on that area. For Equation 1,

$$\Delta T_m = F\ \Delta T_{\ln} = F\left(\frac{\Delta T_0 - \Delta T_L}{\ln(\Delta T_0 / \Delta T_L)}\right) \tag{2}$$

The overall heat transfer coefficient (U) based on the outside surface of the tube is defined as (Ozisik 1985):

$$U = \frac{1}{(D_o / D_i)(1/h_i) + (D_o / D_i)F_i + [1/(2k)]D_o \ln(D_o / D_i) + F_0 + 1/h_o} \tag{3}$$

Table 1. Water - LiBr absorption refrigeration system calculations based on a generator temperature of 75°C and a solution heat exchanger exit temperature of 55°C

Point	H (kJ/kg)	$\dot{m}$ (Kg/s)	P (kPa)	T (°C)	%LiBr (X)	Remarks
1	85.3	0.00517	0.93	36	55	
2	85.3	0.00517	4.82	36	55	
3	124.7	0.00517	4.82	**55**	55	Sub-cooled liquid
4	183.2	0.00474	4.82	**75**	60	
5	140.3	0.00474	4.82	52.8	60	
6	140.3	0.00474	0.93	45.6	60	
7	2624.8	0.000431	4.82	**70**	0	Superheated Steam
8	131.0	0.000431	4.82	31.5	0	Saturated liquid
9	131.0	0.000431	0.93	6	0	
10	2511.8	0.000421	0.93	6	0	Saturated vapour
11	23.45	0.000011	0.93	6	0	Saturated liquid

Description	Symbol	kW
Capacity (evaporator output power)	$\dot{Q}_e$	**1.0**
Absorber heat, rejected to the environment	$\dot{Q}_a$	1.28
Heat input to the generator	$\dot{Q}_g$	1.36
Condenser heat, rejected to the environment	$\dot{Q}_c$	1.08
Coefficient of performance	COP	0.74

3.1 Condenser Heat Exchanger Design

For the condenser heat exchanger design the data shown in Table 2 are considered. These data are extracted from Table 1 and Figure 1.

The overall heat transfer coefficient is given by Equation 3. For this equation, the value of the fouling factors (F_I, F_o) at the inside and outside surfaces of the tube can be taken as 0.09 m^2-°C/kW (Howell *et al*., 1998) and k for copper = 383.2 (W/m-°C). The heat transfer coefficients, h_i, h_o, for the inside and outside flow need to be calculated.

The Petukhov-Popov equation (in Kreith and Bohn, 1997) for turbulent flow inside a smooth tube gives:

$$\overline{Nu_D} = \frac{(f/8)\mathrm{Re}_D \mathrm{Pr}}{K_1 + K_2 (f/8)^{1/2}(\mathrm{Pr}^{2/3} - 1)} \tag{4}$$

Table 2. Condenser Heat Exchanger Characteristics

Parameter	Type / Value
Heat Exchanger Type	Single pass horizontal tubes (outside diameter D_o = 9.5mm and inside diameter D_i=8.1mm)
Cooling water inlet temperature	27°C
Cooling water outlet temperature	28.5°C
Mass flow rate ($\dot{m}$)	0.172 kg/s
Condenser load ($\dot{Q}_c$)	1080 W
Condensing water vapour temperature	From 70°C to 31.5°C
Condensing water vapour pressure	4.82 kPa
Condensing water vapour mass flow rate	0.000431 kg/s

Equation 4 applies for Reynolds numbers $10^4 < \mathrm{Re}_D < 5x10^6$ and Prandtl numbers, 0.5<*Pr<2000.* The Petukhov-Popov equation agrees with the experimental results for the specified range within ± 5%.

The water properties at the mean temperature of (27+28.5)/2=27.75°C are; ρ = 997.5 kg/m^3, ν = 0.8365x10^{-6} m^2/s, k = 0.610 W/m-°C, Pr = 5.72 and c_p = 4178 J/kg-°C. $\dot{Q}_c$ equals to 1080kW (Table 2), therefore:

$\dot{m} = \dot{Q}_c$ / (c_p* Δt)=1080 / (4178*(28.5-27)) =0.172 kg/s and Re=32279

Substituting the above values into Equation 4 and replacing $Nu = h_i\ D_i\ /\ K,$ gives h_I = 16325 W/m^2-°C.

Nusselt's analysis of heat transfer for condensation on the outside surface of a horizontal tube, gives the average heat transfer coefficient as (Nusselt in Ozisik, 1985):

$$\mathrm{h_m} = 0.725\left[\frac{\mathrm{g}\rho_1(\rho_1-\rho_\mathrm{v})h_{fg}k_l{}^3}{\mu_l(T_v-T_w)D_o}\right]^{0.25} \tag{5}$$

The physical properties in Equation (5) should be evaluated at the mean wall surface and vapour saturation temperature.

The above equation is recommended in the case of condensation on a single horizontal tube, although a comparison of the average heat transfer coefficient for vertical surfaces with that found by experiments has shown that the measured heat transfer coefficient is about 20% higher than the values suggest by theory (Ozisik 1985).

In the case of this study, the average temperature of the condensate film is (31.5+27.75)/2=29.6°C and its physical properties are; ρ_l = 996.97 kg/m^3, ρ_v = 0.03285 kg/m^3, h_{fg} = 2431.2x10^3 J/kg, k_l = 0.613 W/m-°C, μ_l = 801.4x10^{-6} kg/m-s, T_v = 31.5°C, T_w = 27.75°C and D_o = 0.0095 m. Therefore h_m = 15200 W/m^2-°C.

By substituting the above values in Equation 3 a resulting overall heat transfer coefficient of U=2980 W/m^2-°C is determined.

From the temperature values shown in Table 2 the required temperature differences can be calculated as: ΔT_L = *3*°C and ΔT_o =4.5°C, Therefore the logarithmic mean temperature difference is 3.7°C and Equation 1 gives A=1080/(3.7*2980)m^2 =0.0984 m^2.

The heat exchanger tubes that are used have a length of 1.0 m and an outside diameter of 0.0095 m, resulting in an area of 0.0298 m^2. The calculated length of pipe is therefore 3.3 m. Assuming the average heat transfer coefficient (h_m) for condensation on the outside surface of a horizontal tube to be 20% higher, 3.19 m of pipe are calculated. If h_m is considered to be 20% lower, then 3.46m of pipe are necessary. Therefore 4 horizontal pipes may be used.

3.2 Generator Heat Exchanger Design

For the Generator Heat Exchanger design the data shown in Table 3 are used. These data are extracted from Table 1 and Figure 1.

The generator provides sensible heat and latent heat of vaporisation. The sensible heat raises the inlet stream temperature up to the saturation temperature. This amount of heat, typical in practice, is 13% of the total heat required (Herold *et al.*, 1996). The heat of vaporisation consists of the heat of vaporisation of pure water and the latent heat of mixing of the liquid solution. Typically, the heat of mixing is about 11% of the heat of vaporisation for water/ lithium bromide.

Table 3. Generator Heat Exchanger Characteristics

Parameter	Type/ Value
Heat Exchanger Type	Single pass horizontal tubes (outside diameter D_o= 9.5mm, inside diameter D_i = 8.1mm)
Generator load ($\dot{Q}_g$)	1360 W
Generator pressure	4.82 kPa
Generator solution	Entering: 55% LiBr at 55°C Leaving: 60% LiBr at 75°C
Generator water vapour mass flow rate ($\dot{m}$)	0.000431 kg/s

The above analysis indicates that the heat to be provided by the generator can be based on the heat of vaporisation of pure water, increased by about 23% in a typical design.

Although considerable research work has been done in the past on the pool boiling of liquids, the data on water/ lithium bromide solutions are found to be meagre (Varma *et al.*, 1994). Experimental results indicate that the boiling phenomenon is not significantly affected by tube diameter but is greatly affected by the solution concentration. As the solution concentration increases the heat transfer coefficient decreases. Also the heat transfer coefficient increases as the heat flux increases. Average heat transfer coefficients were found to vary between 1600 W/m^2-°C and 7500 W/m^2-°C (Varma *et al.*, 1994). For the above work, stainless steel tubes were used. The tube material also affects the heat transfer coefficient as shown by an empirical relation developed by Rohsenow (in Kreith and Bohn, 1997) for nucleate boiling. Since no formula is available for calculating the exact heat transfer coefficient, the generator design is based on actual experimental results (see section 4.4).

3.3 Solution Heat Exchanger Design

For the Solution Heat Exchanger design the data shown in Table 4 are considered. These data are extracted from Table 1 and Figure 1. Equation 3 gives the overall heat transfer coefficient. As before, the value of the Fouling factors (F_I , F_o) at the inside and outside surfaces of the tube can be taken as 0.09 m^2-°C/kW (Howell *et al.*, 1998) and k for copper at a mean temperature of 54.7°C equal to 381.5 W/m-°C. The Heat transfer coefficients, h_i , h_o, for the inside and outside flow need to be calculated.

The 60% LiBr solution properties at 75+52.8/2=63.9°C are; μ = 3.48x10^{-3} N-s/m^2, k = 0.466 W/m-°C, $Pr = c_{p*}\, \mu / k$ = 14.4 and c_p = 1926 J/kg-°C.

Since $Re = 4\dot{m} / \pi D \mu$ = 214.14 the flow is laminar, and $Nu = h_i D_i / K$=3.66. Therefore h_i = 3.66*0.466/0.0081 = 210.5 W/m^2-°C.

The 55% LiBr solution properties at 55+36/2=45.5°C are; μ = 3.17x10^{-3} kg/m-s, k = 0.465 W/m-°C, $Pr = c_{p*}\, \mu / k$ = 14.0 and c_p = 2057 J/kg-°C.

The hydraulic diameter D_H for the annulus (Kreith and Bohn, 1997) is the difference between the inside diameter of the external tube (D_2) and the outside diameter of the internal tube (D_1). In the case of the present study this is $D_H = D_2 - D_1$ = 0.013-0.0095 = 0.0035m.

Table 4. Solution Heat Exchanger Characteristics

Parameter	Type / Value
Heat Exchanger Type	Single pass annulus, 15.0mm outside pipe diameter with 1mm wall thickness and 9.5mm inside pipe diameter with 0.7mm wall thickness
Cooled solution (60% LiBr) inlet temperature	75°C
Cooled solution (60% LiBr) outlet temperature	52.8°C
Cooled solution Mass flow rate ($\dot{m}$)	0.00474 kg/s
Heated solution (55% LiBr) inlet temperature	36°C
Heated solution (55% LiBr) outlet temperature	55°C
Heated solution Mass flow rate ($\dot{m}$)	0.00517 kg/s

The Reynolds number based on the hydraulic diameter and bulk temperature properties is 92.4 indicating that the flow is laminar since its value is below 2100 and $Nu = h_o D / K$=3.66. Therefore h_o = 3.66*0.465/0.0035 = 486.5 W/m^2-°C.

By substituting the above values in Equation 3 the resulting overall heat transfer coefficient (U) based on the outside surface of the tube is 127.8 W/(m^2-°C).

From the temperature values shown in Table 4 the required temperature differences can be calculated as: ΔT_o=75-55=20°C and ΔT_L=52.8-36=16.8°C. Therefore the logarithmic mean temperature difference is 18.34°C.

Finally, from Equation 1, the area of the heat exchanger needed is A= 0.0867 m^2. The heat exchanger tube will have an outside diameter of 0.0095 m, resulting in an area of 0.0298 m^2. Therefore a length of 0.0867/0.0298=2.9 m tube is needed.

3.4 Absorber Heat Exchanger Design

For this design, the solution film can flow downward either on horizontal or on vertical tubes. The design of the horizontal tubes for the absorber, although theoretically well studied, presents a great problem with the shell tightness because of the large length of welds. For this reason an alternative design with vertical tubes, housed in a cylindrical shell is employed by considering the data shown in Table 5. These data are extracted from Table 1 and Figure 1.

In the case of this study the water vapour produced in the evaporator is absorbed in the flow of the LiBr-water solution and is not condensing on the heat exchanger tubes. The design of the heat exchanger therefore requires values for heat and mass transfer coefficients.

A number of researchers studied this type of absorber. Morioka *et al.* (1993) conducted experiments on steam absorption for films flowing down a vertical pipe. The obtained results show that for Film Reynolds numbers in the range of 40-400 the heat transfer coefficients of the film are between 1500 W/m^2-°C and 3000 W/m^2-°C.

Table 5. Absorber Heat Exchanger Characteristics

Parameters	Type / Value
Heat Exchanger Type	Single pass vertical tubes, with outside diameter D_o = 9.5mm and inside diameter D_i = 8.1mm
Cooling water inlet temperature	30°C
Cooling water outlet temperature	31°C
Cooling water mass flow rate $(\dot{m})$	0.307 kg/s
Absorber load $(\dot{Q}_a)$	1280 W
Solution cooling	From 45.6°C to 36°C
Absorber pressure	0.9346 kPa
Water vapour mass flow rate	0.000421 kg/s
Inlet solution mass flow rate	0.00474 kg/s

Grossman (1983), described a theoretical analysis of the combined heat and mass transfer process in the absorption of gas or vapour into a laminar liquid film.

Conlisk (1992), developed a design procedure for predicting the absorption capacity of a given tube based on the governing geometrical and physical parameters.

Patnaik *et al*. (1993) presented a model, based on the solution of differential equations to calculate axial solution concentration and temperature distributions along a vertical tube absorber.

A practical model for absorption of vapours into a laminar film of water and LiBr falling down along a constant temperature vertical plate was described in Andberg and Vliet (1983). The model developed considers non-isothermal absorption and the equations presented showed good agreement to experimental results. For this reason this method is chosen for the design of the absorber.

The independent variables, which affect the problem, are solution mass flow rate, solution inlet concentration, absorber pressure and wall temperature. The data are correlated with the introduction of the "absorption percentage (A_p)", defined as:

$$A_p = \frac{C_{IN} - C_{OUT}}{C_{IN} - C_{EQ}} * 100 \qquad (6)$$

Determination of the equilibrium concentration, C_{EQ}, requires the solution of the following set of expressions:

A = -2.00755 + .16976 * X - 3.133362 * .001 * X^2 + 1.97668 * .00001 * X^3

B = 321.128 - 19.322 * X + .374382 * X^2 - 2.0637 * .001 * X^3

C = 6.21147

D = -2886.373

E = -337269.46

T′ = (-2 E / [D + (D^2 – 4 E (C – LOG(P / 6894.8))]$^{.5}$) - 459.72

T_W = (5 / 9) * (A * T' + B - 32)

The above set of expressions requires an iterative type of solution to find C_{EQ}, given T_W and P. In the case of this study T_W=30.5 °C and P=935 Pa therefore C_{EQ} = 0.52 and from equation (4) A_P=62.5. A_P is correlated to the length of plate (L) by the expression:

$$L = a\overset{*}{m}^{b} \tag{7}$$

Assuming the length of every pipe of the absorber to be 1m, an iterative solution gives $\overset{*}{m}$=0.0292 kg/m-s corresponding to the area of 5.4 pipes. Therefore 6 pipes are used.

The next step is to check the area of pipes needed to cool the solution to the required level. Patnaik *et al.* (1993), suggest that Wilke's correlation, valid for constant heat flux wall with progressively decreasing difference from isothermal wall outside the entrance region, can be used for the falling film. It is assumed that the flow is fully developed in a wavy, laminar regime and that the bulk solution temperature profile is linear with respect to the transverse coordinate (Patnaik *et al.,* 1993). Wilke's correlation is:

$$h_s = \frac{k_s}{\delta}\left(0.029 * \left(\mathrm{Re}_s\right)^{0.53} \mathrm{Pr}_s^{\ 0.344}\right) \tag{8}$$

The film thickness δ, (m) is given by:

$$\delta = \left(\frac{3\mu\Gamma}{\rho^2 g}\right)^{1/3} \tag{9}$$

and the solution Reynolds number (Re_s)for the vertical tube *is:*

$$Re_s = 4\Gamma/\mu \tag{10}$$

In the case of this study the mean properties of the solution at 45.6°C and 57.5%LiBr are; ρ= 1663 kg/m^3, μ = 4.20x10^{-3} N-s/m^2, k = 0.453 W/m-°C and c_p = 1991 J/kg-°C, with a resulting Pr=18.46.

Assuming 6 pipes and substituting the above values into Equations 8 - 10, a solution convective heat transfer coefficient h_s *of* 865 W/m^{2-}°C results.

The cooling water properties at the mean temperature of (30+31)/2=30.5°C are; ρ = 996.7 kg/m^3, ν = 0.7876x10^{-6} m^2/s, k = 0.615 W/m-°C, Pr = 5.34 and c_p = 4177.5 J/kg-°C.

Therefore, substituting the above values into Equation 4 and replacing $Nu = h_i\, D_i / K$, $h_I =$ 6175 W/m^2-°C.

By substituting the above values in Equation 3 the resulting overall heat transfer coefficient (U) based on the outside surface of the tube is 650 W/m^{2-}°C.

In this case $\Delta T_{ln} = 9.3°C$. Therefore from Equation 1, the resulting length of each pipe is 1.19m instead of 1m assumed. This means that the area of 6 pipes is not enough to cool the solution to the required level. Checking for 7 pipes by repeating the above procedure, a length of 1.0 m results, with h_s= 840 W/m^2-°C, $h_I =$ 5450 W/m^{2-}°C and U= 625 W/m^2-°C, which indicates that 7 pipes are adequate to cool the solution.

3.5 Evaporator Heat Exchanger Design

For the Evaporator Heat Exchanger design the data shown in Table 6 are considered. These data are extracted from Table 1 and Figure 1.

To facilitate construction, it was decided to construct the evaporator heat exchanger in a similar way as the absorber heat exchanger. A search in the literature has shown that this method is not studied and that the preferred method is to allow the liquid to enter inside a tube. The fluid inside the tube is heated by the run of fluid at the outer surface of the tube, so that progressive vaporisation occurs. The heat transfer coefficient increases with distance from the entrance since heat is added continuously to the fluid. It is also not yet possible to predict all of the characteristics of this process quantitatively because of the great number of variables upon which the process depends and the complexity of the various two-phase flow patterns that occur as the quality of the vapour-liquid mixture increases during vaporisation (Kreith and Bohn, 1997). Therefore, in the case of this study, the mean heat transfer coefficient is determined experimentally (see section 4.5).

Table 6. Evaporator Heat Exchanger Characteristics

Parameter	Type / Value
Heat Exchanger Type	Single pass horizontal tubes (outside diameter D_o = 9.5mm and inside diameter D_i = 8.1mm)
Water inlet temperature	27°C
Water outlet temperature	17°C
Mass flow rate $(\dot{m})$	0.0239 kg/s
Evaporator load $(\dot{Q}_e)$	1000 W
Evaporator pressure	0.9346 kPa
Evaporator water vapour mass flow rate	0.000431 kg/s

4. CONSTRUCTION OF THE UNIT AND EXPERIMENTAL RESULTS

All heat exchangers described above, are constructed in a way that permits the use of varying number of tubes. The objective is to modify the number of heat exchanger tubes and thus the heat exchange area, until the required operating conditions, depicted in Tables 1-6, are obtained. This will ensure a design with a good COP as shown in Table 1. For this purpose thermometer pockets and flow-meters are installed at various points of the unit for measurements and adjustments.

4.1 Condenser Heat Exchanger

The shell of the condenser is constructed from a copper tube 1m in length, 67mm external diameter and 1.2mm thick. The shell is insulated with 13mm armaflex. Inside the shell, 8 copper tubes with 9.5mm outside diameter and 8.1mm inside diameter are housed as shown in Figure 2. These tubes are connected externally with hoses. In this way the required length of pipe can be utilized. The condenser tube is inclined, with its right side at a slightly lower level than the left side. In this way the condensate is collected in a measuring tube. The vapour reaches the condenser tubes through a vertical copper tube 30cm length and 35mm external diameter. The experimental results show that 3 pipes are adequate for the condensation of the produced vapour indicating that the overall heat transfer coefficient is slightly higher that the calculated one. The experimental overall heat transfer coefficient is 3265 W/m^2-°C, i.e. 10% greater than the theoretical. In the case that the vapour produced in the generator is not liquefied in the condenser the pressure of the system gradually increases and the system becomes unbalanced.

4.2 Generator Heat Exchanger

Since no formula is available for calculating the exact heat transfer coefficient, the generator is constructed in such a way so that variable lengths of tube can be used in order to heat the solution. The shell of the generator is constructed from a copper tube 1m in length, 67mm external diameter and 1.2mm thick. The shell is insulated with 13mm armaflex. Ten copper tubes with 9.5mm outside diameter and 8.1mm inside diameter were housed inside the shell as shown in Figure 2. Every tube can be isolated with a valve; therefore the required number of tubes can heat the water/lithium bromide solution. A sight glass is also fitted on the generator to observe the solution level. Hot water is circulated in the pipes with a pump. The water is heated to the required temperature with two electrical heaters, 3kW each. The electrical heaters are fitted on a water cylinder consisting of a galvanised pipe 150mm in diameter and 1m in length (Figure 3). The temperature is controlled by two thermostats, which permit accurate setting. It should be noted that although electricity is used to power the unit, the system is designed to be used with solar energy collectors (see section 2).

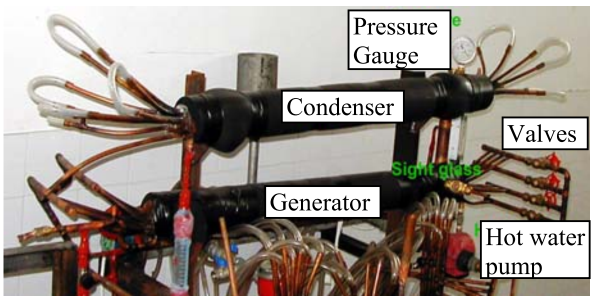

Figure 2. Generator and Condenser.

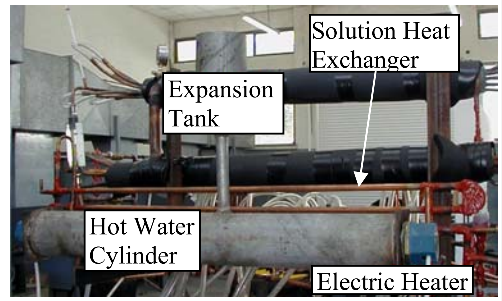

Figure 3. Hot Water Cylinder.

The energy flow, $\dot{Q}$ supplied to the generator, is calculated from $\dot{Q} = \dot{m}\ c_p\ \Delta t$. The experimental results indicate that for 92°C heating water inlet temperature and 88°C outlet temperature, 1.4 kW were delivered to the solution of the generator per meter tube. Since the solution temperature of the generator is about 70°C, the logarithmic mean temperature difference is 19.9°C and the tube area used is 0.03 m^2, the average heat transfer coefficient is 1400/(19.9*0.03)=2345 W/m^2-°C. For 88°C heating water inlet temperature and 85°C outlet temperature, 1.14 kW is delivered to the generator solution per meter tube. The logarithmic mean temperature in this case is 16.5°C and the average heat transfer coefficient is 1140/(16.5*0.03)=2300 W/m^2-°C.

Lowering the heating water inlet temperature to 80°C the outlet temperature becomes 78°C and the input heat to the generator is 0.61 kW per meter tube. The average heat transfer coefficient in this case becomes 2270 W/m^2-°C.

Therefore for delivering the required amount of heat to the generator solution (1.36 kW), one pipe can be used with 92°C heating water inlet temperature, two pipes with 88°C and just over two pipes with 80°C. In every case the heating operation is intermitted.

4.3 Solution Heat Exchanger

The solution heat exchanger is constructed with the copper pipes specified in Table 4 and is positioned slightly below the generator as indicated in Figure 3. During operation the solution of the generator is flowing to the absorber by gravity and pressure difference and the flow is adjusted with a valve. The solution being heated is pumped from the lower pressure of the absorber to the generator. The operation of the heat exchanger showed that the predicted length is approximately correct. Some variation in temperature, of a few degrees, is observed during operation that is due to a difference in the concentration of the solution and flow rate.

4.4 Absorber Heat Exchanger

The absorber is constructed from a number of vertical tubes with 9.5mm outside diameter. On the top part of every tube, another tube 12mm inside diameter and 100mm in length is fitted in such a way that the absorber solution can pass inside the 2mm gap between the two tubes as shown in Figure 4. The top part of the 100mm tube is soldered onto the vertical tube so that the thin film of fluid can escape only downwards. The film of the absorber solution can thus be cooled by water flowing inside the vertical tube.

In the case of this study, two absorber shells are constructed from a copper tube each being 1-meter length, 67mm external diameter and 1.2mm thick. The two absorber shells are joint to the evaporator shell with copper tubes 20cm in length and 35mm external diameter insulated with 13mm armaflex (Figure 5). Eight vertical tubes, constructed as explained above are housed inside each shell, as shown in Figure 6. The required number of tubes is connected externally with hoses. The solution tubes are also grouped for easy feeding of the solution.

During the operation of the unit it is observed that the overall heat transfer coefficient (*U*) based on the outside surface of the tube is 400 W/m^2-°C. Therefore for removing the heat load of the absorber approximately 12 tubes (six on every side) are needed instead of the 7 calculated. This is due to a lower value of the overall heat transfer coefficient attributed to the fact that in practice the wetting of the tubes is not uniform.

Also during operation no increase of the absorber pressure is observed indicating that the water vapour produced in the evaporator is absorbed completely.

4.5 Evaporator Heat Exchanger

The evaporator heat exchanger (Figure 5) is constructed in the same way as the absorber heat exchanger. The shell is made from a copper tube 1m in length, 67mm external diameter and 1.2mm thick.

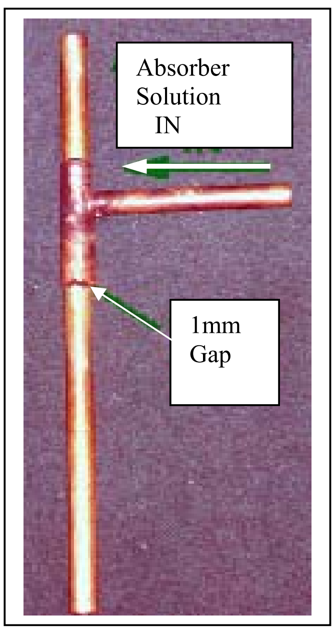

Figure 4. Absorber tubes detail

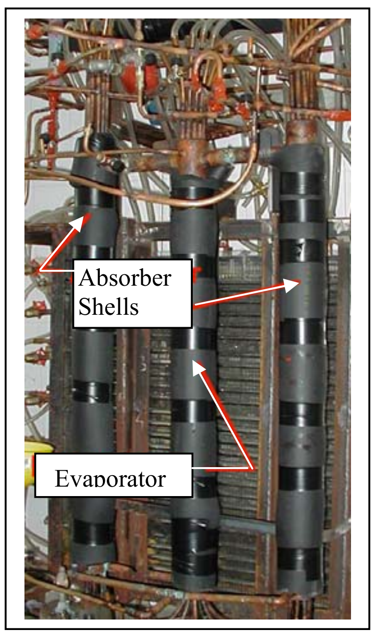

Figure 5. Absorber and Evaporator shells

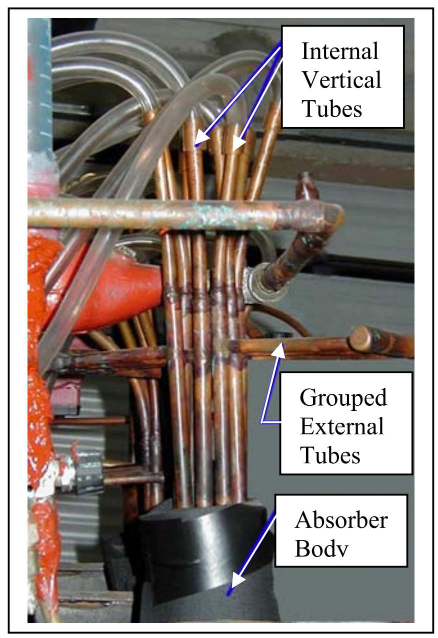

Figure 6. Absorber detail

The shell is insulated with 13mm armaflex. Inside the shell eleven vertical tubes, similar to the absorber tubes, are fitted. Supplied water runs at the external surface of every tube where it evaporates. In the case of falling films evaporating on the outer side of a tube it is difficult to predict the heat transfer coefficient. The advantage of high coefficient in falling-film exchangers is partially offset by the difficulties involved in distribution of the film and maintaining complete wettability of the tube (Perry and Green, 1984). For this reason the heat transfer coefficient of the specific construction is found experimentally.

In the case of the present study in order to avoid the secondary heat exchanger, water from the water sink is directly used in a single pass. Also to improve wettability, a mass flow rate of 0.3 g/s per tube is applied. The resulting overall heat transfer coefficient determined experimentally is U= 195 W/m^2-°C at a ΔT_{ln} = 15.5°C.

The heat transfer coefficients, h_i *and* h_b, for the inside and outside flow can therefore be determined as follows:

The evaporator heat exchanger uses water from the well. During the experiment the entering temperature is 27°C and the leaving temperature is 17°C. The water properties at (27+17)/2 = 22°C are; ρ = 999.2 kg/m^3, v = 0.9581x10^{-6} m^2/s, k = 0.600 W/m-°C, Pr = 6.67 and c_p = 4180.7 J/kg-°C.

Since 11 vertical tubes are used, the mass flow rate per tube is $\dot{m} = \dot{Q}e\ /\ 11*(c_p*\ \Delta t)$=1000 / (11*4180.7*(27-17) = 0.00217 kg/s and Re = 356.5. Therefore the flow is laminar, resulting in $h_i = 3.66*K\ /\ D_i$ = 271.1 W/m^2-°C

Substituting the known values into Equation 3, h_o = 695 W/m^2-°C which actually is a low value. Assuming that this value does not change significantly with ΔT_{ln} which in an actual case is about 6°C (cooling water entering at 13°C and leaving at 11°C), 29 tubes are needed.

5. GENERAL OBSERVATIONS

The absorption refrigeration unit constructed as explained above operates under the design conditions, specified in Tables 1 - 6. For the operation of the unit two pumps are used, one for the evaporator and one for the absorber. The flow of the two pumps needs to be adjusted in such a way so as to provide the required amount of fluids to the two heat exchangers. The evaporator pump simply re-circulates water from a reservoir (Figure 7) to the tubes of the evaporator. Care is needed to maintain the level of the water in the reservoir steady, thus allowing only the extra water fed from the condenser to evaporate.

The absorber pump sends the required amount of solution back to the generator. The solution level in the generator must be maintained constant.

In the experimental unit the generator solution is passing to the solution heat exchanger and to the absorber by the pressure difference and by gravity. Also the condenser water runs down to the evaporator due to the same reasons. A manual throttle valve is used in both cases to adjust the flow. During operation a fine adjustment of the operation of the pumps and the setting of the valves is needed to maintain the operation of the unit at the design level. This operation is not easy and continuous observation is necessary. In an industrial unit an electronic control device should be employed. This device can use widely available level-sensing transducers, electronic valves and temperature sensors.

During operation conventional controls can detect the chilled water temperature and when it rises, more heat is supplied to the generator. This in turn, tends to concentrate the salt solution and can cause crystallization. Crystallization can be avoided if it is ensured that the solution mass fraction does not exceed a certain value (e.g. 0.65 mass fraction LiBr) and can be guaranteed by the operation of the machine in the designed temperature range.

Another point of interest is the introduction of the LiBr solution into the unit. For this task a vessel can be connected with a valve to the appropriate point of the unit (i.e., generator) as shown in Figure 8. When the air in the unit is removed, the atmospheric pressure will drive the solution into the unit. In the experimental unit 5 kg of LiBr salt were used.

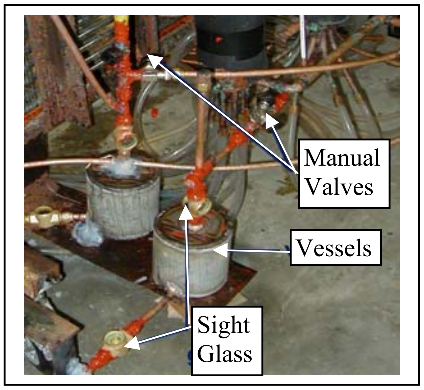

Figure 7. Absorber and evaporator liquid vessels.

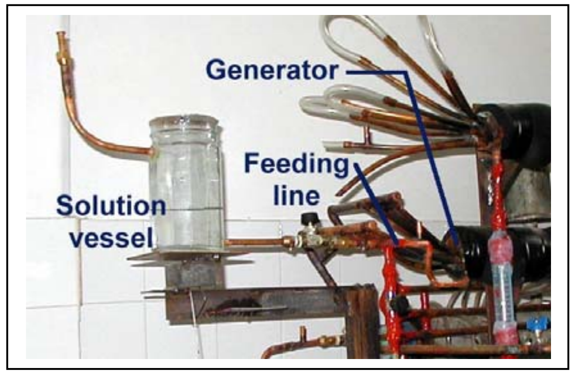

Figure 8. Introduction of LiBr solution into the generator

6. CONCLUSION

The unit designed is constructed and each heat exchanger is adjusted to the required output. In this way the designed COP is ensured. Based on the construction experience of the 1kW unit, the cost for a 12kW unit, that can cover the needs of a typical insulated house, is estimated as C£ 4800. The present cost of the absorption unit together with its running cost is economically viable. Considering also the destruction of the ozone layer caused by the use of electric chillers, absorption units will offer a better environment, especially if some form of renewable or waste energy is used for their operation.

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Nomenclature

a=-132* ln ((100- A_P)/86.0) (see Eq. 7)
b=1.33 (see Equation 7)
C_{EQ}= solution equilibrium concentration (mass fraction)= C_{EQ} =X_{EQ}/100
C_{IN}= solution inlet concentration (mass fraction)
C_{OUT}= solution outlet concentration (mass fraction)
D_i = Inside diameters of the tube (m)
D_o = outside tube diameter (m)
F= Correction factor depending on the type of the heat exchanger
$f = (1.82 \log_{10} \mathrm{Re}_D - 1.64)^{-2}$
F_I , F_o= Fouling factors at the inside and outside surfaces of the tube (m^2-°C/W), respectively
g = gravitational constant, 9.81 m/s^2
h_s= solution convective heat transfer coefficient (W/m^2-°C)
h_i , h_o = Heat transfer coefficients for inside and outside flow (W/m^2-°C), respectively
h_{fg} = latent heat of condensation (kJ/kg)
h_m= average heat transfer coefficient (W/m^2-°C)
k = Thermal conductivity of tube material (W/m-°C)
k_s= solution thermal conductivity (W/m-°C)
k_l = liquid thermal conductivity (W/m-°C)
$K_1 = 1 + 3.4 f$ (see Equation 4)
$K2 = 11.7 + \frac{1.8}{\mathrm{Pr}^{1/3}}$ (see Equation 4)
$\overset{*}{m}$ = mass flow rate per unit width of plate (kg/m-s)
m = mass flow rate (kg/s)
$\overline{Nu_D}$ = Nusselt number = $h_i\, D_i / K$
P= pressure (Pa)
Pr = Prandtl number
Pr_s= solution Prandtl number
Re_D = Reynolds number =$V_m\, D_i\ / \nu = 4\, m\ / \pi D_i\, \mu$
Re_s= solution Reynolds number for vertical tube
T_v = vapour saturation temperature (°C)
T_w = wall surface temperature (°C)
V_m = mean velocity (m/s)
X=concentration of LiBr in solution (%)

Greek:

Γ= mass flow rate per wetted perimeter (kg/m-s)
δ = film thickness (m)
$\Delta T_{\ln}$ = Logarithmic mean temperature difference (LMTD), (°C)
ΔT_0 = Temperature difference between the hot and cold fluid at the inlet, (°C)
ΔT_L = Temperature difference between the hot and cold fluid at the outlet, (°C)
μ = absolute viscosity (N-s/m^2) = $\nu\, \rho$
ν = kinematic viscosity (m^2/s)
ρ = density (kg/m^3)
ρ_l = liquid density (kg/m^3)
ρ_v = vapour density (kg/m^3)
μ_l = absolute viscosity of liquid (N-s/m^2)