Theortical Model of Absorber for Miniature LiBr-H_2o Vapor Absorption Refrigeration System

[1]Manu.S
Dept.of Mech. Engg.,
Sri Siddhartha Institute of Technology, Tumkur

[2]T.K.Chandrashekar
Dept.of Mech.Engg.,
Sri Siddhartha Institute of Technology,Tumkur

[3]T.B.Prasad
Dept. of Mech. Engg.,
Sri Siddhartha Institute of Technology,Tumkur

[4]Nagendra
Dept.of Mech.Engg.,
Sri Siddhartha Institute of Technology,Tumkur

ABSTRACT

Convective air cooling techniques are facing difficulties in removing high heat flux under limited space for thermal management. In order to overcome this problem the researchers are developing new technologies like microchannel heat sink, heat pipes and miniature refrigeration system. Among all the methods, refrigeration is the promising technology to overcome the problem. The absorption based heat pump system pressurizes the fluid by chemical compressor and consumes less power compared with the mechanical compressor. The coefficient of performance of this system is more compared with the other systems. In absorption based refrigeration system the absorber is widely acknowledged as the bottle neck which defines the performance and cost of the system. On account of these drawbacks, the present study aims to design the absorber for miniature LiBr-H_2O Vapor Absorption Refrigeration for cooling electronic devices.

***Keywords* - : Absorber; LiBr-$H_2$0; Miniature**

I. INTRODUCTION

Recent advances in semi conductor technologies are accompanied by rapid increase in the power density levels from high performance chips such as microprocessors, IC chips etc.. Convective air cooling techniques are facing difficulties in removing high heat flux under limited space for thermal management. The researchers are developing new technologies like microchannel heat sink, heat pipes, miniature refrigeration system and etc… Among all the methods of cooling, refrigeration is the promising technology to overcome the problem. In this aspect, small miniature vapor compression system for electronics cooling

which remove 50W of heat from central processing unit having Coefficient of Performance of 2.25 [Mongia et.al, (2006)]. [Drost and Friedrich, (1997)] developed the mesosale water lithium bromide absorption based heat pump system for portable and distributed space conditioning applications with 350W of cooling capacity. The merit of system is, it uses the chemical compressor which is driven by heat [A.Beutler et.al (1997), Yoon Jo Kim et.al (2008)].

In the chemical compressor the power consumption for liquid compression is less compared to mechanical compression. Libr (Lithium-Bromide)/water pair absorption system posses several advantages over the other types of absorption systems such as highest Coefficient of Performance compared to other absorption units operating under same conditions. The system works efficiently without the need of rectification columns and a basic generator is sufficient to vaporizes the water, as the absorbent is non-volatile and having higher evaporating temperature. This will be beneficial for the cooling of electronic device operating under room temperature.

In absorption refrigeration system the absorber is widely acknowledged as the bottle neck which defines the viability of the entire absorption cycle and also it is the most critical component in absorption heat pump system in terms of cycle performance and system cost [A.Beutler et.al (1997), J.D Killion and Garimella (2001)]. The different models of absorber were analyzed based on working fluid pair like ammonia/water and water/Libr. The Libr is non volatile and the water vapor will be in pure state as such there is no resistance to mass transfer in the vapor phase during the absorption process [Killion and Garimella (2001)].

The appropriate relations for describing the properties of the working fluid for different operating conditions were mentioned in design and construction of the vapor absorption system [G.A Florides et.al, (2003)].The series of experimental studies on falling film absorber for smooth horizontal tubes were carried out. The results reveal that, effect of cooling water inlet temperature on absorber performance is significant as the cooling water inlet temperature decreases; there is an enhancement in the heat transfer coefficient. In addition to that, inlet solution concentration of LiBr has a critical impact on heat transfer coefficient. As the inlet concentration increases there is an increase in the heat transfer coefficient [S.M Deng and W.B.Ma (1999)]. From the numerical model of the vertical absorber provides relation between inlet temperature of coolant and size of the absorber [A. Yigit, (1999)]. By providing the flow guidance medium between the falling film and the coolant tube results in the reduction of the absorber size without affecting the vapor and coolant side pressure drop [Nitin Geol and D.Yogi Goswami (2005)].

The series of experiments by [William Miller and Majid Keyhani(2001) on simultaneous heat and mass transfer on a vertical falling film absorber showed that local heat and mass transfer rates are linear along the length of the absorber. The correlation of Nusselt and Sherwood Number provides the simple calculation for load and mass absorber under given inlet conditions of the absorber. The experiments on the adiabatic vapor absorption into aqueous Libr solution for three different arrangements like mono-disperse droplets, unstable jets breaking into Poli-disperse droplets and film falling ramps by [D.Arzoz et.al (2005)]. The results showed that falling film can handle higher flows per unit of disperse area and simultaneously, offer more contact area with water vapor pre-unit length. [Vinod Naryan and Jeromy Jenks, (2008)]have found that as the channel depth increases there was is decrease in the overall heat transfer coefficient for weak solution and increases for vapor refrigerant in the Micro scale-Film Ammonia-water bubble. In the light of the above, the present work is focussed on systematic minature absorber for vapor absorption refrigeration system working under minimum load.

II. Cycle Description

A single effect LiBr water absorption refrigeration system is shown in the Fig.1. The water was used as a refrigerant and LiBr used as absorbent in the present system. The heat was added to the weak solution in generator which boils the refrigerant and flows into the condenser. The refrigerant rejects the heat as it condenses in the condenser and flow restrictor reduces the pressure and temperature of the refrigerant. The low pressure and temperature liquid refrigerant enters into the evaporator. In the evaporator liquid refrigerant evaporates by absorbing the heat and flows to the absorber. The vapor refrigerant (water) is absorbed by the strong LiBr solution forming the weak solution in the absorber and this process is exothermic. The weak solution is pumped to the generator through the liquid heat exchanger. The remaining strong solution in the generator flows to the absorber through the liquid heat exchanger and flow restrictor. In liquid heat exchanger, heat flows from the strong solution to weak solution and temperature, pressure were reduced in the flow restrictor.

Improving the performance of the absorber directly increases the efficiency of the system. The absorber is the largest part of the system. The literature reveals the different design of the absorber by changing the configuration such as tube spacing, tube diameter, tube length, tube material and different operating conditions. Since the absorption is associated with heat and mass transfer. The vapor flow mechanism divides the absorber into two broad categories namely, falling film absorption and bubble absorption mode. The falling film mode was widely accepted because of its simplicity and effectiveness with respect to heat and mass transfer. The absorber is also been categorized as counter flow and co current flow based on the direction of the flow. Since The counter flow is an attractive approach in which rate of heat and mass transfer is more compared with the other.

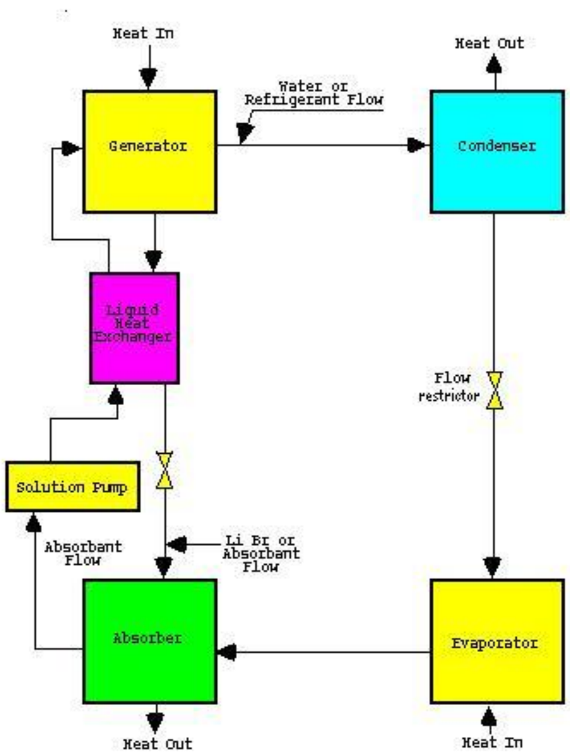

Figure 1: Description of LiBr Absorption Refrigeration Cycle

One of the disadvantage of vapor absorption refrigeration was its large size of components especially the absorber which is the most critical component and its characteristics have significant effect on the overall system performance. So in order to miniature the absorption system, it is necessary to miniaturize the absorber. In this context the paper aims in the efficient design of the miniature absorber having high heat and mass transfer coefficients by providing minichannel.

III. Conceptual Design

Since the absorption process is associated with heat and mass transfer, the various system parameters are included in the designing a miniature absorber. The main parameters that can be grouped are as follows:- 1) Operating parameters, 2) Geometric parameters & 3)Thermal/Fluid parameters based on Weilin and Issam Mudwar for the heat sink.

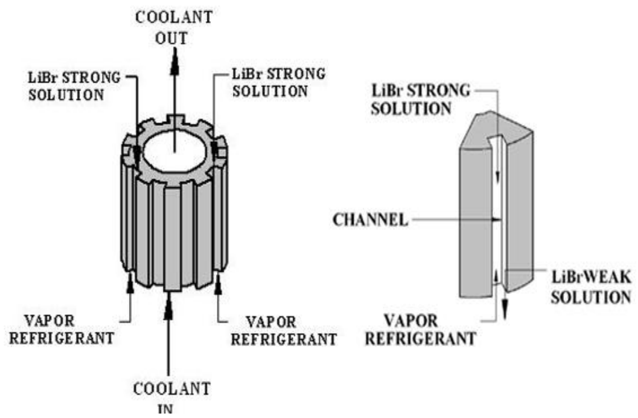

Figure 2:Miniature of Absorber

The Fig. 2 illustrates the construction of a minichannel absorber like minichannel heat sink with a coolant system. The absorber was made by high thermal conductivity material such as aluminum. In order to obtain the compactness, series of vertical parallel mini-slots of different ranges hydraulic diameters were cut on the circumference and centre hole was drilled at the centre for the coolant.

The Fig 2 shows the conceptual diagrams of the liquid cooled counter flow falling film absorber. It is shown that the refrigerant vapor flows through outer channels and the aqueous Libr solution counter-currently flow along the outer channel walls which are cut on the circumference. All the heat generated during the absorption process is rejected to the coolant through the cylinder wall. A liquid cooling scheme provides higher heat transfer coefficient and lower coolant temperature than air cooling leads to improve the absorber performance.

IV. SOLUTION SIDE

4.1 OPERATING PARAMETERS

The operating parameters represent the condition under which the absorber is expected to operate. They are inlet strong solution concentration (x_{ss}) and outlet weak solution concentration(x_{ws}), inlet mass flow rate of strong solution (m_{ss}), inlet mass flow rate of refrigerant(m_r) and outlet mass flow rate of weak solution (m_{ws}), inlet strong solution temperature(T_{ss}) and inlet refrigerant temperature(T_r) and outlet weak solution temperature(T_{WS}). One of the important considerations in designing absorber is the absorber temperatures (T_{ws}), Generator temperatures/ Bubble pump temperature (T_g), Condenser temperatures (T_c) and Evaporator temperatures (T_e) and their pressures. [F.L Lansing] provided the systematic model for determination of the operating parameters and it as follows.

Temperature of the strong solution enters in to the absorber is given by:

$$T_{ss} = T_g - z(T_g - T_{ws}) \quad 1$$

Based on generator and condenser temperature the concentration of strong solution is given by:

$$x_{ss} = \frac{49.04 + 1.25T_g - T_c}{134.65 + 0.47T_g} \quad 2$$

In consideration of absorber and evaporator temperature the concentration of weak solution is given by:

$$x_{ws} = \frac{49.04 + 1.25T_a - T_e}{134.65 + 0.47T_a} \quad 3$$

Based on the heat load the mass flow rate of refrigerant is given by

$$\dot{m}_r = \frac{Q_e}{h_{eout} - h_{cout}} \quad 4$$

Where, h_{eout} outlet enthalpy of refrigerant which is coming out of evaporator

$$h_{eout} = (572.8 + 0.417T_e)4.187 \quad 5$$

Where, h_{cout} outlet enthalpy of refrigerant which is coming out of condenser

$$h_{cout} = (T_c - 25)4.187 \quad 6$$

Mass flow rate of strong solution is given by:

$$m_{ss} = m_r\left(\frac{x_{ws}}{x_{ss} - x_{ws}}\right) \quad 7$$

Mass flow rate of weak solution weak solution is given by:

$$m_{ws} = m_r\left(\frac{x_{ss}}{x_{ss} - x_{ws}}\right) \quad 8$$

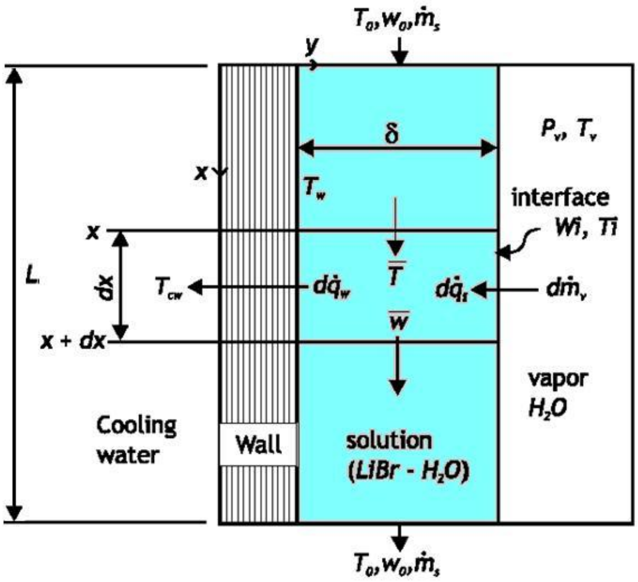

Fig 3: Schematic of the Model (Khalid, 2001)

The properties of solution were determined by average temperature of inlet and outlet of absorber but refrigerant properties are found at the evaporator temperature.

4.2 Thermal/Fluid Parameters and geometric parameters

The parameters are dependent transport parameters that determine the performance of absorber under given operating and geometric parameters. Geometric parameters includes length, diameter, width of channel, depth of channel, thickness of the fin and the thermal parameters include heat and mass transfer coefficient, pressure drop etc… Due to the negligible frictional losses the pressure drop is neglected. The absorber model started with following assumptions and the procedure of Wassenaar, Reinderhette (2008).

- The liquid is Newtonian has constant physical properties. The values of the properties based on the liquid entry conditions.
- The film flow may be considered laminar and one dimensional.
- Momentum effects and shear stress at the interface are negligible.
- The absorbed mass flow is small relative to the film mass flow.
- At the interface, thermodynamic equilibrium exists between the vapor and liquid. The relation between surface temperature and mass fraction is linear with constant coefficient at constant pressure.
- All the heat of absorption is released at the interface.
- The liquid is a binary mixture and only one of the components is present in the vapor phase
- There is no heat transfer from the liquid to vapor and no heat transfer because of radiation, viscous dissipation, pressure gradient or gravitational effects.
- There is no diffusion because of pressure gradient, temperature gradient or chemical reactions.
- Diffusion of heat and mass in the flow direction is negligible relative to the diffusion perpendicular to it.

Under the above assumption, the equations of momentum, energy and diffusion of mass and their specific boundary are represented in four dimensionless combined ordinary differential equations [19-11] as below. These equations describe the average mass fraction of water in the solution, the average solution temperature, the heat transfer to the cooling medium across the plate wall per unit width (dq_w), and the mass fraction of the water vapor to the film per unit width(dm_v), in one infinitesimal part of the film with length(dx) as shown fig 3.

Change of average mass fraction of water is given by [Wassenaar, Reinderhette (2008)]:

$$d\gamma = \frac{-A}{m_s(T_{eq}-T_o)}\left[1-(w_{eq}-w_o)\bar{\gamma}-w_o\right]d\,m_s \qquad 9$$

Where, $m_s = \rho_s \Gamma_v$

The change in the Mass transfer of the water vapor to film is given by [Wassenaar, Reinderhette (2008)]:

$$d\,m_v = \frac{m_s\, dFo(w_{eq}-w_o)(1-\bar{\gamma}-\bar{\theta})}{\frac{1}{Le}\left[\frac{A(1-w_0)}{Nu_i}-\frac{1}{Sh}\right]} \qquad 10$$

The average solution temperature change is given by [Wassenaar, Reinderhette (2008)]:

$$d\theta = \frac{1}{m_s C_p(T_{eq}-T_o)}\left\{\left[h_m+\phi_w(w_{eq}-w_o)\bar{\gamma}\right]d\,m_v - dq_w\right\} \qquad 11$$

Heat transfer to cooling medium across the plate is given by [Wassenaar, Reinderhette (2008)]:

$$dq_w = \frac{C_p\, m_s\, dF_h(T_{eq}-T_o)(\bar{\theta}-\theta_c)}{\left[\frac{1}{Nu_w}+\frac{1}{Nu_c}\right]} \qquad 12$$

Dimensionless temperature is given by [Wassenaar, Reinderhette (2008)]:

$$\theta = \frac{(T-T_o)}{(T_{eq}-T_o)} \qquad 13$$

Dimensionless mass fraction is given by [Wassenaar, Reinderhette (2008)]:

$$\gamma = \frac{(w-w_o)}{(w_{eq}-w_o)} \qquad 14$$

Sherwood number (Sh) is given by [Wassenaar, Reinderhette (2008)]:

$$Sh = 0.69(Fo\ Le)^{-0.5} \qquad 15$$

A good estimation for Nusselt, found analytically [Khalid A .Joudi, and Ali H.Lafta, (2001), Wassenaar, Reinderhette (2008)]

These numbers are

$$Nu_i = 2.67$$

$$Nu_w = 1.6$$

Constant in interface thermodynamic equilibrium [Wassenaar, Reinderhette (2008)]:

$$A = \frac{-\rho_s D_s h_m}{[(1-w_o)k_s C_1]} \quad 16$$

Lewis number is given by [Wassenaar, Reinderhette (2008)]:

$$Le = \frac{D_s}{\alpha_s} \quad 17$$

Heat of absorption is given by [Wassenaar, Reinderhette (2008)]:

$$h_m = h_v - [h_{ss} + (1-w_0)\phi_w] \quad 18$$

Dimensionless *x* (heat transfer Fourier number) is given by Wassenaar, Reinderhette (2008)]:

$$Fo = \frac{x\alpha_s}{\delta_s \Gamma_v} \quad 19$$

Volumetric flow rate per wetted length is given by [Wassenaar, Reinderhette (2008)]:

$$\Gamma_v = \frac{m_{ss}}{2L\rho_s} \quad 20$$

Wassenaar, Reinderhette (2008) provides Solution film thickness based on volumetric flow rate per wetted length is given by

$$\delta_s = \left(\frac{3\nu_s \Gamma_v}{g}\right)^{\frac{1}{3}} \quad 21$$

The Equilibrium temperature and Equilibrium mass fraction of water are defined under the above assumption, by a linearization of the thermodynamic equilibrium equation of LiBr-H_2O solution at a fixed pressure. The equilibrium equation is expressed in the solution temperature T_s as a function of the LiBr concentration (X) in the solution and the vapor pressure (P) or refrigerant temperature T_e=f(X, P) or T_s=f(X, T_r). [Khalid, (2001)]

$$T_s = C_1 w_{eq} + C_2 \quad 22$$

Where

$$C_1 = -21.8789 - 0.58527 T_r$$

$$C_2 = 0.0436688 + 1.407 T_r$$

Equilibrium temperature is given by Wassenaar, Reinderhette (2008)

$$T_{eq} = C_1 w_0 + C_2 \quad 23$$

Equilibrium mass fraction of water is given by [Wassenaar, Reinderhette (2008)

$$w_{eq} = \frac{1}{C_1}(T_0 - C_2) \quad 24$$

By integrating the differential equations and by assuming the length and width of the absorber until the equation are balanced. The obtained width of the absorber is converted into circular dimension. The circumference of the cylinder is same as the perimeters of a rectangular wall by this the diameter of the circle is determined. And the complete channel geometry can be determined by assuming the aspect ratio, hydraulic diameter and fin spacing ratio of the channel.

Aspect ratio is given by:

$$\beta_s = \frac{a_s}{b_s} \quad 25$$

Fin spacing ratio is given by

$$FA_s = \frac{t_s}{b_s} \quad 26$$

Hydraulic diameter is given by

$$D_{\mathbf{hs}} = \frac{2a_s b_s}{a_s + b_s} \quad 27$$

No of channels is given by:

$$N_s = \frac{\pi d_s}{a_s + t_s} \quad 28$$

By assumed the cooling plate is fully wet by the liquid film so that there is no direct contact or heat exchange between the refrigerant vapor and cooling plate is allowed and thermal equilibrium exist at the interface between the aqueous LiBr solution and refrigerant vapor and also mass transfer driven by thermal and pressure difference is regarded to be negligible [Yoon Jo Kim et.al (2008),W.Wilke (1962)] developed the correlation for heat transfer coefficient between the cooling plate and liquid film solution is determined by:

$$h_s = 1.88 \frac{k_s}{\delta} \quad 29$$

Where, δ is the film thickness

$$\delta = \left[\frac{3\mu\Gamma}{\rho_s^2 g} \right]^{1/3} \quad 30$$

Mass transfer coefficient for the range 40<Re_{film}<280 and Sc<2.13×10^5/$Re_{film}^{1.6}$, is given by [Yoon Jo Kim et.al (2008)]

$$k_m = 3.7522 \, \mathrm{Re}_{film}^{\frac{1}{3}} . \frac{D_{AB}}{S_r} \quad 31$$

Where, Re_{film} Reynolds number of the film

$$\mathrm{Re}_{film} = \frac{4\Gamma}{\mu}$$

To account for heat and mass transfer at the interface, the modified heat transfer coefficient is given by [Yoon Jo Kim et.al (2008)]

$$h_s^* = \frac{N.c_{ps}}{1 - e^{-\left(\frac{N.c_{ps}}{h_s}\right)}} \quad 32$$

The wall temperature at the solution side can be determined by [Yoon Jo Kim et.al (2008)]

$$\frac{d(\dot{m}_s .h_s)}{dz} = h_s.p_s(T_w - T_s) + \overset{*}{h}_s .p_s(T_i - T_s) + \frac{d\,\dot{m}_{ss}}{dz} h_{si} \quad 33$$

Liquid vapor interface temperature by [Yoon Jo Kim et.al (2008)]

$$T_i = \frac{q_{sen.l}}{h_s^* p_s L} + T_s \quad 34$$

Perimeter of the solution

$$P_s = \left[2[\delta] + a_s \right] \quad 35$$

Perimeter of refrigerant

$$P_r = \left[2[b_s - \delta] + a_s \right] \quad 36$$

The mean of inlet and outlet wall temperature at solution side of the absorber determines the final wall temperature of solution side.

V. COOLANT SIDE

5.1 Operating Parameters

The operating parameters represent condition under which the Absorber coolant side is expected to operate. They include material, type of coolant, Inlet temperature (T_{ci}), Outlet temperature (T_{co}), Mass flow rate of coolant(m_c), total heat (Q_a) represents the heat absorbed by the coolant. By assuming coolant only absorbs the heat and the complete surface is insulated and temperature rise is limited to 10°C .

$$m_c = \frac{C_{p.c}(T_{c.o} - T_{c.i})}{Q_a} \quad 37$$

5.2 Thermal/Fluid Parameters

The parameters are dependent transport parameters that determine the performance of Absorbers under given operating and geometric parameters. Thermal parameters includes heat transfer coefficient, pressure drop etc…

5.3 Heat transfer coefficient for coolant side

[Satish.G.Kandlikar et.al (2006)] provides the Nusselt number for fully developed laminar flow in ducts,

$$Nu=3.66 \quad 38$$

Heat transfer coefficient (h_c) in coolant side is given by:

$$h_c = \frac{Nu_c k_c}{D_c} \quad 39$$

By neglecting the conductive resistance, the Overall heat transfer coefficient is given by:

$$\frac{1}{U} = \frac{1}{h_s} + \frac{1}{h_c} \quad 40$$

Logarithmic mean temperature difference is given by:

$$LMTD = \frac{(T_a - T_{c.i}) - (T_a - T_{c.o})}{\ln \frac{(T_a - T_{c.i})}{(T_a - T_{c.o})}} \quad 41$$

Number of channels in coolant side is given by:

$$Q = UA_c LMTD \quad 42$$

5.4 Geometric Parameters

The geometric parameters include miniature Absorber are illustrated in Fig.3. The overall Absorber dimensions are length (L), Diameter (d_c).

The thickness of wall between solution side and coolant side is proportional to the thermal conduction resistance between the solution side and the coolant side due to this, the thickness always should less so that all the heat will be absorbed by the coolant. Surface area of the coolant side (A_c) is given by

$$A_c = \pi d_c L_c \qquad 43$$

VI. THEORETICAL RESULTS

The thermodynamic properties of solution and refrigerant are determined by relations given by G.A Florides et.al, (2003).The above model theoretically designed and the results are tabulated as below:-

Table 1:Thermal/ Fluid Parameters

Parameter	Values
Heat transfer coefficient between the cooling plate and liquid film in W/m^2K	25453
Average wall temperature between inlet and outlet in ^oC	38
Coolant side heat transfer coefficient W/m^2K	1050
Overall heat transfer coefficient W/m^2K	1008

Table 2:Operating parameters for solution

Parameter	Values
Fluid	Steam and LiBr
Generator Temperature in ^oC	70.88
Condenser Temperature in ^oC	49
Evaporator Temperature in ^oC	24
Absorber Outlet Temperature in ^oC	32.35
Evaporator Load in W	100
Absorber Load in W	108
Refrigerant flow rate in kg/s	$4.2X10^{-5}$
Strong Solution Concentration in kg Libr	0.475
Weak Solution Concentration in kg Libr	0.41
Mass flow rate of strong solution in kg/s	$2.69X10^{-4}$
Mass flow rate of weak solution in kg/s	$3.120X10^{-4}$
Operating Pressure in Pa	3000

Table 3:Geometric Parameter for Solution and Refrigerant

Parameter	Values
Length in m	0.08
Width in m	0.075
Diameter in m	0.05
Hydraulic Diameter in m	0.003
Fin spacing ratio (α)	0.16
Material	Aluminum
Channel Geometry	
Depth in m	0.002
Width in m	0.01
Thickness in m	0.00095
No. of channel (N)	13

Table 4:Geometric parameters for Coolant

Parameter	Values
Length in m	0.08
Hydraulic Diameter in m	0.003
Material	Aluminum

Table 5:Operating parameters for coolant

Parameter	Values
Temperature rise in ^oC	10
Inlet temperature in ^oC	26
Mass flow rate in kg/s	$2.58\ X10^{-3}$

VII. Conclusion

The paper concludes with following results:-

- The design procedure developed for miniature based is simple and systematic.
- The design procedure based on different parameters like operating, geometric and thermal/fluid.
- The theoretical results showed the enhancement in the heat transfer coefficient 25453 W/m^2K and mass transfer coefficient 6.78 $X10^{-5}$ m^2/s which leads to improve the performance of the system.

References

1. Beutler,; F.Ziegler;G.Alfed.(1996):Falling film absorption with solutions of a hydroxide mixture, International Absorption Heat Pump Confrence,Monteral,Cannada, pp.303-309.
2. R.Mongia,:K.Masahiro,;E.Distefano,;J.Barry,;W.Chen,; M.Izenson,;F.Possasami,;A.Zimmermann,;M.Mochizuki.(2006):Small scale refrigeration system for electronic cooling within note book computer, Proc.Tenth Intersociety confrence on Thermal and Thermomechanical Phenemenon in Electronics system,San Diego, USA, pp.751-758.
3. Drost,;Friedrich.(1997):Miniature heat pump for portable and distributed space conditioning application: Proc. Thirty-Second Intersociety Energy Conversion Engineering Confrence,Honolou, Hawaii,USA, pp.1227-1274.
4. Yoon Jo Kim,;Yogendra K.Joshi,;Andrei G.Fedrov.(2008): Performance Analysis of air-cooled microchannel absorber in absorption-based minature electronic cooling system, Journal of Mechanical Science and Technology, 22, pp.338-349.
5. J.D Killon,; S.Garimella.(2001): A critical review of models coupled heat and mass transfer in falling film absorption, 24, pp.755-797.
6. S.M Deng,;W.B.Ma. (1999): Experimental studies on the characterstics of an absorber using LiBr/H2O solution as working fluid, International Journal of Refrigeration, 22, pp.1999.
7. Yigit.(1999):A Numerical Study of Heat and Mass Transfer in Falling Film Absorber, International Communication Heat Mass Transfer, 26(2), pp.269-278.

8. Nitin Geol,;D.Yogi Goswami. (2005):A Compact Falling Film Absorber,Journal of Heat Transfer,127, pp.957-1013.
9. G.A. Florides,; S.A. Kalogiroua,; S.A. Tassoub,; L.C. Wrobelb. (2003):Design and construction of a LiBr–water absorption machine, 44(15),pp.2483-2508.
10. William Miller,;Majid Keyhani.(2001):The Correlation of Simulataneous Heat and Mass Transfer Experimental Data for Aqueous Lithium Bromide Falling Film Absorption, 123, pp30-42.
11. D. Arzoz,; P. Rodriguez,; M. Izquierdo. (2005): Experimental study on the adiabatic absorption of water vapor into LiBr–H2O solutions, 25, pp.797-811.
12. Vinod Naryan,; Jeromy Jenks. (2008); Effect of Channel Geometry variations on the performance of a constrained Microscale-Film Ammonia-Water Bubble Absorber, 130, pp. 1-9.
13. Wassenaar,;Reinderhette (2008): Simultaneous heat and mass transfer in a horizontal tube absorber: Numerical tools for present and future absorber designs (Ph.D. Thesis), Delft University of Technology,1994.
14. W.Wilke, 1962:Warmeubergang and riselfime,VDI-Forschungsheft 490,B28.
15. Satish G. Kandlikar,; Srinivas Garimella,; Dongqing Li, Stéphane Colin,; Michael R. King,2006: Heat Transfer and Fluid Flow in Minichannels and Microchannels, Elsevier publication, 1st Edn.
16. F.L Lansing, http://ipnpr.jpl.nasa.gov/progress_report2/42-32/32AA.PDF, pp.1-11.
17. Khalid A.Joudi,;Ali H.Lafta.(2001):Simulation of a simple absorption refrigeration system, Energy Conversion and Management 42, pp.1575-1605.

Nomenclature

A_c	Area in m^2
C1	Constant
C2	Constant
Cp	Specific heat in kJ/kg
d	Diameter
D_{hs}	Hydraulic diameter in m`
D_s	Mass diffusivity in m^2/s
FA_s	Fin spacing ratio
Fo	Heat transfer Fourier no
g	Acceleration due to gravity in m^2/s
h	Enthalpy in kJ/kg
h	Heat transfer coefficient in w/m^2k
k	Thermal conductivity in w/m.k
L	Length in m
Le	Lewis no
m	Mass flow rate in kg/s
N	No of Channel, Mass flux at the phase interface
Nu	Nusselt number
P	Perimeter in m
Q	Heat load in W
Sh	Sherwood number
T	Temperature in ^{O}C
t	thickness
w	Mass fraction of the volatile component
X	concentration
h_{si}	Interface enthalpy
x	Distance along co-ordinate axis in m
Z	Effectiveness

Subscript

*	Modified
a	Absorber
c	condenser, coolant
e	Evaporator
eq	Equilibrium
g	Generator
i	inlet
l	Liquid
m	mixing
o	Entrance or outlet
r	Refrigerant
s	solution
sen	Sensible
ss	strong solution
ws	Weak Solution

Greek words

γ	Activity Coefficient / dimensional less mass fraction
A	Constant in interface thermodynamic equilibrium
α	Thermal diffusivity in m^2/s
β	Aspect ratio
δ	Film Thickness
Γ	Mass flow per wetted perimeter.
μ	Dynamic viscosity in pa-s
ν	Kinematic viscosity in m^2/s
θ	Non dimensional less temperature
ρ	Density in kg/m^3
Γ_v	Volume flow per wetted length