Proceedings of the
International Conference on Mechanical Engineering 2009
(ICME2009) 26- 28 December 2009, Dhaka, Bangladesh

ICME09-RT-11

OPTIMIZATION STUDY OF A SOLAR-OPERATED LITHIUM BROMIDE-WATER COOLING SYSTEM WITH FLAT PLATE COLLECTORS

K. K. Datta Gupta[1], **D. N. Basu**[1] **and S. Chakravarti**[2]

[1]Dept. of Mechanical Engg., Bengal Engg. & Science University, Howrah, West Bengal, India
[2]B.P. Poddar Institute of Management & Technology, Kolkata, West Bengal, India

ABSTRACT
A general theoretical study on the design and optimization of the solar-operated LiBr-H_2O cooling system has been undertaken. The results of the study show that, in general for fixed initial conditions and given system cooling capacity, there exists an optimum generator temperature at which the required collector area is minimum. From the study it has also been revealed that for a single-stage LiBr-H_2O cooling system flat plate collectors with two covers are preferable.

Keywords: Absorption Cooling Cycle, Flat Plate Collector, COP.

1. INTRODUCTION
Solar energy is gaining popularity because it increases energy independence and sustainability leaving no negative impact on the environment. Absorption cooling is one of the first and oldest forms of air-conditioning and refrigeration system which uses thermal energy to produce cold, and thus solar energy, waste heat and other forms of low grade heat can be employed. Since the demand for cooling, whethere for comfort or food preservation, is greatest at the time of maximum solar radiation, the seasonal and diurnal variations of solar insolation matches very well with the demand for cooling .Thus the utilization of solar energy for absorption cooling systems has been a very popular field of research for the scientific communities. The basic findings of these these activities is the demonstrative ability of flat plate collectors to achieve required temperatures necessary for operation of LiBr-H_2O cooling system.

Alizadeh et al.[1] have carried out performance comparison of solar operated Libr-H_2O and NH_3-H_2O cooling systems, and concluded that LiBr-H_2O system is simpler and operates with higher cooling ratio and heat exchanger parameter. Ward [2] has studied the technical and economic feasibilities of solar operated absorption cooling system viz-a-viz the conventional vapour compression cooling systems. Results indicate that the thermodynamic efficiency of solar absorption cooling is very nearly equivalent to that of an electrically-driven vapour compression system and economically solar absorption cooling is marginal, but improves significantly with tax incentives. Mansoori and Patel [3] have presented a comparative study for different combinations of refrigerant-absorbent: NH_3-H_2O, NH_3-NaSCN and LiBr-H_2O and concluded that LiBr-H_2O combination is preferred for localities with high environmental temperatures. Li and Sumathy [4] have carried out simulation of a solar air conditioning system with a single storage tank, and concluded that a system with a partitioned hot water storage tank results a quicker early-hour cooling effect and higher overall system COP as compared to a normal stratified storage tank. Using TRANSYS software and weather parameters of Syprus, Florides et al.[5] have carried out modeling and simulation to compare the performance of a solar assisted LiBr-H_2O cooler powered by three different types of solar thermal collectors, and concluded that for a cooling load of 1.5 ton, the optimized system consists of a 15m^2 compound parabolic collector with a 600l capacity hot water storage tank. Liu and Wang [6] have presented performance prediction of a solar/gas driven double effect LiBr-H_2O cooling system where solar energy together with high pressure water vapour from high pressure generator supply heat to the low pressure generator. Simulation results demonstrate that this kind of double effect absorption system is feasible both technically and economically. Fathi et al.[7] have presented performance analysis of a solar absorption refrigerator by considering four temperature reservoirs and both internal and external irreversibilities to predict optimal design and operating conditions. Assilzadeh et al.[8] have carried out modeling and simulation for a LiBr-H_2O cooler using TRANSYS program and solar radiation data for Malaysia. The results presented show that the system performance is in phase with the diurnal variation of solar radiation. For optimum generator temperature and higher system reliability, a hot water storage tank has been recommended. Mittal et al.[9] have reported the the effects of hot water inlet temperatures on COP, contributed solar fractions and surface areas of cooling system components. Ardehali et al.[10] developed a computer code to simulate the performance of a solar assisted LiBr-H_2O cooling system, and examined the effect of clearness index on the auxiliary heating source. Based on experimental results of a solar absorption cooling system in Spain, Rodriguez et al.[11] have reported saving in conventional energy and economy, and reduction in CO_2 emission. Mazloumi et al.[12] have carried out simulation of a single effect LiBr-H_2O cooling system for a city in Iran powered by parabolic trough collectors with a hot water storage tank. For a cooling load of 5 ton, they have recommended a minimum collector area of 57.6 m^2.

The objective of this paper is the conjugate analyses of the solar collector and LiBr-H_2O cooling systems to predict the overall system performance with the help of a computer program. Finally, schemes have been evolved to determine the relative amount of collector material required for optimum system performance, and to observe the influence of number of glass cover(s) on the performance of the whole system in the operating regime of the generator.

2. MATHEMATICAL MODEL

2.1 Absorption Refrigeration System

Fig.1. illustrates the schematic diagram of a solar operated LiBr-H_2O absorption cooling system which operates on two pressure levels: low pressure in the evaporator and absorber, and high pressure in the generator and condenser. High pressure liquid refrigerant (water) leaving the condenser undergoes a throttling process in the expansion valve and enters the evaporator as a low-pressure, low-quality wet vapour.The refrigerant leaves the evaporator as saturated vapour after extracting heat (Q_E) from the cooling load, and enters the absorber where it is absorbed by the strong solution returning from the generator via the liquid-liquid heat exchanger and the expansion valve. The weak solution thus formed in the absorber is pumped back to the generator via the heat exchanger. In the generator external heat addition (Q_G) from the solar collector causes boiling of the solution resulting in the formation of water vapour which enters the condenser and leaves as saturated liquid, thus completing the cycle.

In order to relate the characteristics of the cooling cycle to the properties of the working fluids, mass and energy balances are to be performed on each component by taking a control volume around each of them. Several assumptions are made to simplify the analysis, two important ones are that the system is operating under the steady state condition and the pump work is negligible. The following equations are obtained:

Evaporator: $$Q_E = m_w\left(h_{10} - h_9\right) \tag{1}$$

Absorber: $$Q_A = m_{ss}h_6 + m_w h_{10} - m_{ws}h_1 \tag{2}$$

Generator: $$Q_G = m_w h_7 + m_{ss}h_4 - m_{ws}h_3 \tag{3}$$

Condenser: $$Q_C = m_w\left(h_7 - h_8\right) \tag{4}$$

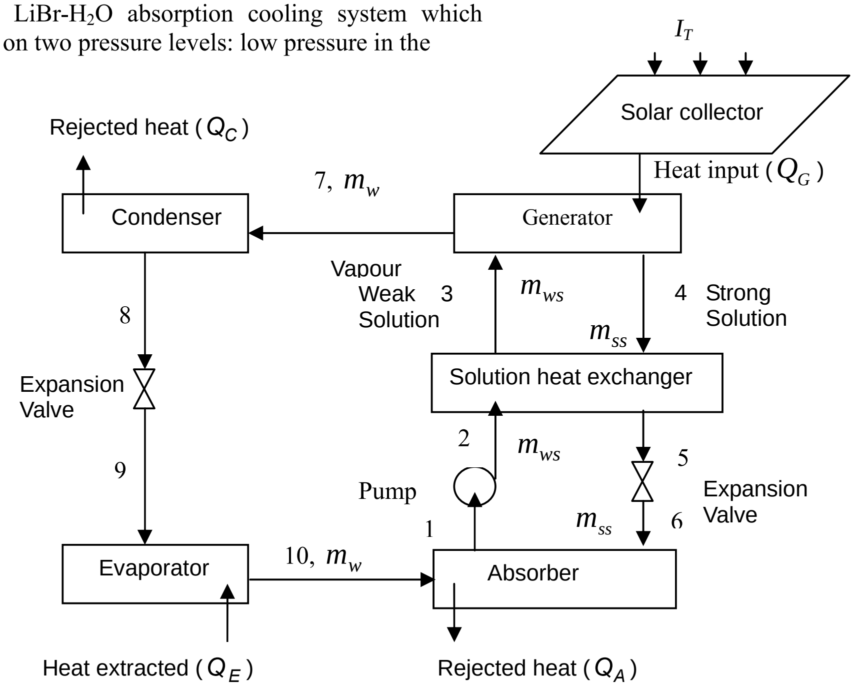

Fig 1. Schematic of solar-operated LiBr-water absorption system

From the mass balance the relation between the mass of refrigerant (water), m_w, and the mass of weak solution leaving the absorber , m_1, is obtained as

$$m_w = m_1 \left(X_4 - X_3\right)/X_4 \quad (5)$$

where X_4 and X_3 are the concentrations of LiBr in the solution at the generator outlet and inlet respectively. The ratio of mass of weak solution leaving the absorber, m_{ws}, to the ratio of mass of refrigerant, m_w, is called circulation factor.

Finally,

$$(COP)_{cycle} = \frac{Q_E}{Q_G} \quad (6)$$

$$(COP)_{system} = \frac{Q_E}{Q_S} \quad (7)$$

where Q_S is the solar radiation incident on the collector plane. It can be easily shown that

$$(COP)_{system} = (COP)_{cycle} \times \eta_c \quad (8)$$

where η_c is the collector efficiency.

2.2 Solar Collector

A flat plate collector is the most simple and widely used means to convert solar radiation into useful heat. The useful heat gain (Q_U) by the working fluid is given by [13]

$$Q_U = A_c F_R \left[S - U_c \left(T_{fi} - T_a\right)\right] \quad (9)$$

where A_c is collector area, F_R is collector heat removal factor, S is absorbed solar radiation, U_c is collector loss coefficient, T_{fi} and T_a are collector fluid inlet and ambient temperatures respectively.
Absorbed radiation, S is given by

$$S = I_T (\tau\alpha)_e \quad (10)$$

where $(\tau\alpha)_e$ is the effective transmissivity-absorptivity product and I_T is the solar radiation flux on the collector plane.
Collector loss coefficient, U_c is given by [13]

$$U_c = U_t + U_b + U_s \quad (11)$$

where U_t, U_b, and U_s are top loss, bottom loss and side loss coefficients respectively.

Top loss coefficient U_t is given by [14]

$$U_t = \left[\frac{N}{\left(\frac{C}{T_{PM}}\right)\left(\frac{T_{PM} - T_a}{N + f}\right)^{0.252}} + \frac{1}{h_w}\right]^{-1} + \left[\frac{\sigma\left[(T_{PM})^2 + (T_a)^2\right]\left[T_{PM} + +T_a\right]}{\frac{1}{\varepsilon_c + 0.0425\mathrm{N}(1 - \varepsilon_c)} + \frac{2N + f - 1}{\varepsilon_g} - N}\right] \quad (12)$$

In the above equation N is the number of cover(s), T_{PM} is mean plate temperature, σ is Stefan Boltzman constant, ε_c and ε_g are emissivity of collector plate and glass cover respectively, and

$$f = \left(\frac{9}{h_w} - \frac{30}{h_w{}^2}\right)\left(\frac{T_a}{316.9}\right)(1 + 0.091N) \quad ,$$

$$C = \frac{204.429\,(Cos\beta)^2}{L^{0.24}} \quad \text{, and}$$

$$h_w = 0.86(\mathrm{Re})^{-1/2} \rho_a C_{Pa} V_\alpha \,\mathrm{Pr}^{-2/3}$$

Collector heat removal factor, F_R is given by [14]

$$F_R = \frac{\dot{m}_w C_{pw}}{U_c A_c}\left[1 - \exp\left(-\frac{F' U_c A_c}{\dot{m}_w C_{pw}}\right)\right] \quad (13)$$

where A_c is the collector area and F' is the collector efficiency factor. Ignoring collector bond resistance, collector efficiency factor, F' is given by [14]

$$F' = \frac{1}{WU_c\left[\frac{1}{U_c\left(2L_f F + D_o\right)} + \frac{1}{\pi D_i h_i}\right]} \quad (14)$$

where W is collector tube spacing, L_f is fin length, h_i is tube to fluid heat transfer coefficient, D_i and D_o are collector tube inner and outer diameters. F is fin efficiency and is given by [13]

$$F = \frac{\tanh\left\{\mathrm{m}\left(\mathrm{W} - \mathrm{D_o}\right)/2\right\}}{\mathrm{m}\left(\mathrm{W} - \mathrm{D_o}\right)/2} \quad (15)$$

$$\mathrm{m} = \sqrt{\mathrm{U_c}/\mathrm{K_c}\, t_c} \quad (16)$$

where K_c is the thermal conductivity of collector plate material, t_c is thickness of collector plate.
Thermal efficiency of the collector is given by [13]

$$\eta = F_R(\tau\alpha)_e - U_c F_R \frac{T_{fi} - T_a}{I_T} \qquad (17)$$

3. COMPUTATIONAL SIMULATION

LiBr-H_2O cooling cycle has been analysed by evaluating the properties of the working fluids expressed in polynomial equations given by [15, 16]. The simulation has been carried out for specific temperatures and pressures in the evaporator and condenser, while the generator temperature has been varied from 65^{0}C to 95^{0}C taking a step of 1^{0}C. For solar collectors, the transmissivity-absorptivity product $(\tau\alpha)_e$ of the cover-absorber system has been taken from [14]. Top loss coefficient, U_t for N number of covers, bottom loss, U_b and side loss, U_s coefficients have been evaluated from the equations given by [14]. The collector heat removal factor F_R has been calculated using the procedure followed by [13]

4. RESULTS AND DISCUSSION

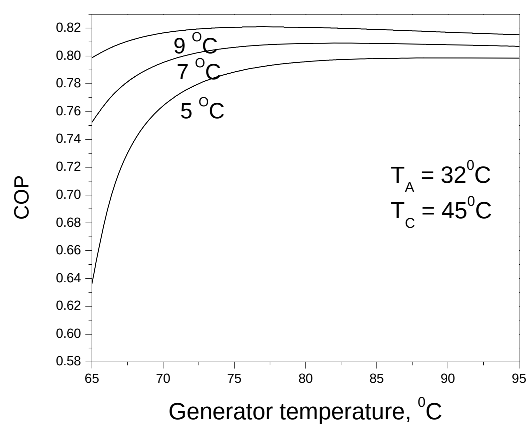

Fig 2. Effect of generator temperature on COP for different evaporator temperatures

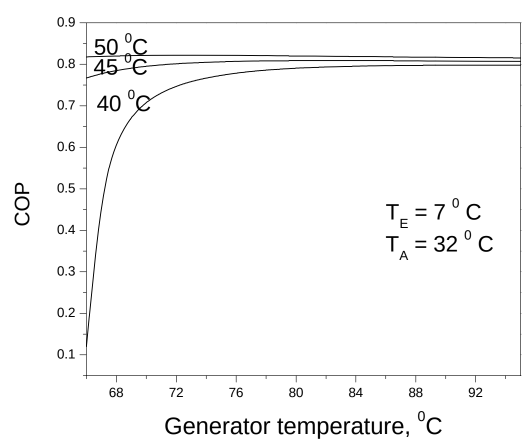

Fig 3. Effect of generator temperature on COP for different condenser temperatures

Figure 2 illustrates the variation of COP with generator temperatures for three evaporator temperatures of 5^{o}C, 7^{o}C and 9^{o}C. The improvement of COP with higher values of evaporator temperatures may be attributed to the fact that as the evaporator temperature is raised, the refrigeration capacity per unit mass of refrigerant increases. Figure 3 depicts the variation of COP with

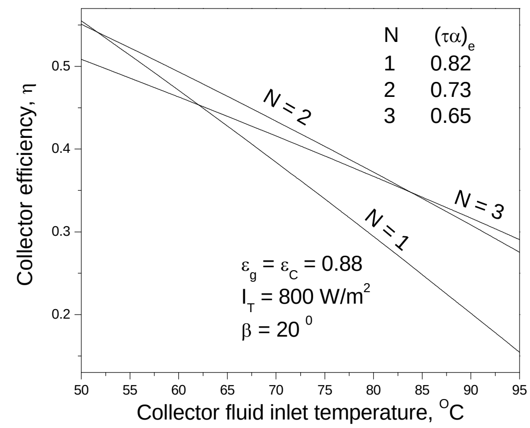

Fig 4. Collector efficiency as a function of collector fluid inlet temperature

generator temperature for condenser temperatures of 40^{o}C, 45^{o}C and 50^{o}C. The variation of collector efficiency with fluid inlet temperatures is shown in Fig.4 for one cover, two cover and three cover systems. This figure demonstrates the effect of number of cove(s) on the collector performance. Normally increase in the number of cover glasses increases the optical losses, but to a greater extent decreases the thermal losses, thus increasing the stagnation temperature of the collector. So for higher operating temperatures, multi-cover flat plate collectors will operate at higher efficiency and thus are preferred.

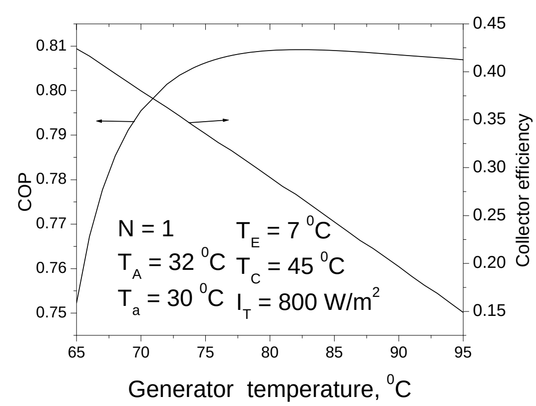

Fig 5. Effect of generator temperature on system performance

The variation of cycle COP and collector efficiency with generator temperature is depicted in Fig.5. The break even point indicates the optimum generator temperature for the solar operated absorption cooling system.

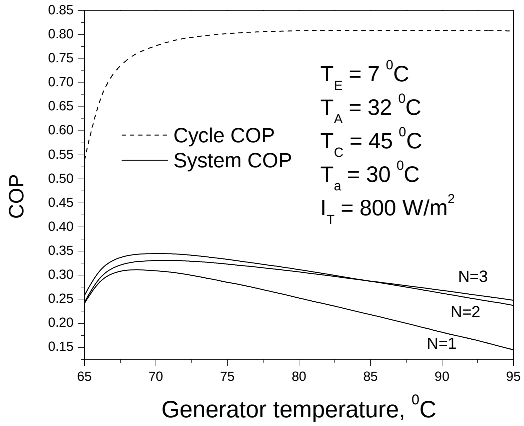

Fig.6. Effect of generator temperature on system and cycle COP

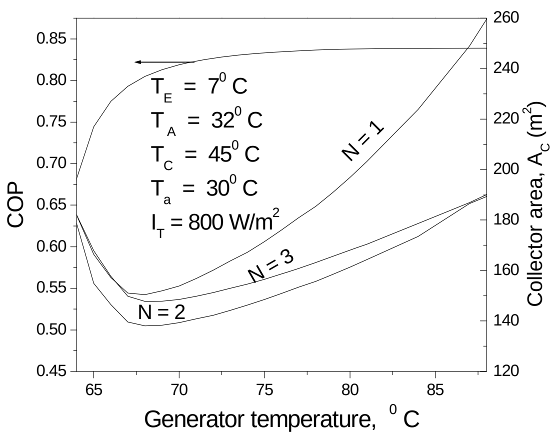

Fig.7. Effect of generator temperature on COP and collector area

Figure 6 illustrates the variation of cycle COP and system COP with generator temperatures for the cooling system driven by solar flat plate collectors having one, two and three glass covers. It is seen that the highest system COP is obtained when flat plate collectors with two glass covers are used. Finally in Fig.7 is shown the variation of required collector area with generator temperatures for one, two and three glass cover systems. If the design generator temperature is assumed to be 5^{o}C below the fluid inlet temperature of collector, then the system could be designed for this generator temperature, and the collector area required for a cooling load of 10 ton can be determined from Fig. 7. It is seen that the required collector area is least for flat plate collectors built with two glass covers.

5. CONCLUSION

The coupling of solar thermal collectors and thermally driven chillers needs a sophisticated control since both components exhibit a reverse dependence of their figure of merits from the operating temperatures. In the design and optimization of the performance of such system the most important variable which has to be taken into account is the temperature of the generator because the other parameters of the system depend on the existing initial conditions, and consequently are fixed. The operating temperature range of hot water supplied to the generator of a solar-operated LiBr-H_2O absorption cooling system is restricted between 72 and 95^{o}C. The lower limit is imposed for two reasons. Firstly, the hot water temperature must be sufficiently high to ensure effective boiling to produce sufficient water vapour from the weak solution in the generator. Secondly, the temperature of the concentrated lithium bromide solution on its passage back to the absorber must be kept high enough to prevent crystallization of lithium bromide. The upper limit is imposed to avoid boiling of water in an unpressurized flat plate collector and formation of crystals in the strong solution after the heat exchanger. Determination of the generator temperature is influenced by several factors. For the optimum design temperature of the generator, the choice of right type collector is essential so that the overall system COP exists near the maximum. Before carrying out analysis of the coupled systems, it is important to know the values of $F_R(\tau\alpha)_e$ and $F_R U_c$ products for different types of collectors which are determined by testing of collectors and are provided by collector manufactures.

6. REFERENCES

1. Alizadeh, S., Bahar, F. and Geoola, F.,1979, “Design and Optimization of an Absorption Refrigeration System Operated by Solar Energy”, Solar Energy, 22:149-154.
2. Ward, D.S., 1979, “Solar Absorption Cooling Feasibility”, Solar energy, 22:259-268.
3. Mansoori, G. A. and Patel, V., 1979, “Thermodynamic Basis for the Choice of Working Fluids for Solar Absorption Cooling Systems”, Solar Energy, 22:483-491.
4. Li, Z. F. and Sumathy, K., 2001, “Simulation of a solar absorption air conditioning system”, Energy Conversion and management, 42:313-327.
5. Florides, G. A., Kalogirou, S. A., Tassou, S. A. and Wrobel, L. C., 2002, “Modeling and Simulation of an Absorption Solar Colling System for Cyprus”, Solar Energy, 72:43-51.
6. Liu, Y. L. and Wang, R.Z., 2004, “Performance prediction of a solar/gas driving double effect LiBr-H2O absorption system”, Renewable Energy, 29:1677-1695.
7. Fathi, R., Gumimi, C.and Ouaskit, S., 2004, “An irriversiblle thermodynamic model for solar absorption refrigerator”, Renewable Energy, 29:1349-1365.
8. Assilzadeh, F. Kalogirou, Ali, Y. and Sopian, K., 2005, “Simulation and optimization of a LiBr solar absorption cooling system with evacuated tube collectors”, Renewable Energy, 30:1143-1159.
9. Mittal,V.,Kasana, K. S. and Thakur, N. S., 2006, “Modeling andSimulation of a Solar aaAbsorption Cooling System for India”, Journal of Energy in South Africa, 17:65-70.

10. Ardehali, M. M., Shahrestani, M and Charles, C. A., 2007, "Energy simulation of solar assisted absorption system and examination of clearness index effects on auxiliary heating", Energy Conversion and Management, 48:864-870.
11. Rodriguez Hidalgo, M. C., Rodriguez Aumente, P., Izquierdo Millan, M., Lecuona Neumann, A. and Salgado Mangual, R., 2008, "Energy and carbon emission savings in Spanish housing air-conditioning using solar driven absorption system", Applied Thermal Engineering, 28:1734-1744.
12. Mazloumi, M., Naghashzadegan, M. and Javaherdeh, K., 2008,"simulation of a solar lithium bromide-water absorption cooling system with parabolic trough collectors", Energy Conversion and Management, 49: 2820-2832.
13. Duffie, J. A. and Beckmen, W. A., 1991, Solar Engineering of Thermal Processes, Wiley, New York.
14. Sukhatme, S.P., 1996, Solar Energy, Tata McGraw-Hill, New Delhi.
15. Jeter, S. M., Lenard, J. L. Y.and Teja, A. S., 1992, "Properties of lithium bromide-water solutions at high temperatures and concentrations-part IV:vapour pressure", ASHRAE Transactions, 98:167-172.
16. Talbi, M. M. and Agnew, 1992, "Exergy analysis of an absorption refrigerator using lithium bromide and water as the working fluids", Applied Thermal Engineering, 20:619-630.

7. NOMENCLATURE

Symbol	Meaning	Unit
A_c	Collector Area	(m^2)
Cp_w	Specific heat of water	(kJ/kg. K)
Di	Inner diameter of collector tube	(m)
Do	Outer diameter of collector tube	(m)
F	Fin efficiency	(Dimension less)
F'	Collector efficiency factor	(Dimension less)
F_R	Collector heat removal factor	(Dimension less)
h	Specific enthalpy	(kJ/kg)
h_i	Tube to fluid heat transfer coefficient	(W/m^{2}K)
h_w	Wind heat transfer coefficient	(W/m^{2}K)
I_T	Solar radiation flux on collector plane	(W/m^2)
K_c	Thermal conductivity of collector plate	(W/m K)
L	Spacing between collector plate and glass cover	(m)
L_f	Fin length	(m)
m_w	Mass of refrigerant	(kg)
m_{ss}	Mass of strong solution	(kg)
m_{ws}	Mass of weak solution	(kg)
m_w	Collector fluid flow rate	(kg/s)
N	Number of covers	(Dimension less)
P_r	Prandtl number	(Dimension less)
Q_A	Heat rejection in the absorber	(kW)
Q_C	Heat rejection in the condenser	(kW)
Q_E	Heat extraction in the evaporator	(kW)
Q_G	Heat addition in the generator	(kW)
Qs	Incident solar radiation	(kW)
Q_U	Useful heat gain by collector	(kW)
R_e	Reynolds number	(Dimension less)
S	Absorbed solar radiation	(kW)
T_a	Ambient temperature	(^{0}C)
T_{fi}	Collector fluid inlet temperature	(^{0}C)
T_{PM}	Mean plate temperature	(K)
t_c	Thickness of collector plate	(m)
U_c	Collector loss coefficient	(W/m^2.K)
V_a	Wind velocity	(m/s)
W	Collector tube spacing	(m)
X	Mass concentration of LiBr in the solution	(kg/kg of ol.)
$(\tau\alpha)_e$	Effective transmissivity – absorptivity product	(Dimension less)
σ	Stefan Boltzman constant	(W/m^2.K^{-4})
ε_c	Emissivity of collector plate	(Dimension less)
ε_g	Emissivity of glass plate	(Dimension less)
ρ_a	Density of air	(kg/m^3)
η	Efficiency	(Dimension less)

8. MAILING ADDRESS

K. K. Datta Gupta
Dept. of Mechanical Engg.
Bengal Engg. & Science University, Howrah-711103, West Bengal, India.
Email: k.dattagupta@rediffmail.com