International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2010

Heat and Mass Transfer Studies on Plate Heat Exchangers in R134a/DMF Based Compact Vapour absorption refrigeration system

Suresh Mariappan
Indian Institute of Technology Madras

Mani Annamalai
Indian Institute of Technology Madras

Heat and Mass Transfer Studies on Plate Heat Exchangers in R134a/DMF Based Vapour Absorption Refrigeration System

Suresh MARIAPPAN, Mani ANNAMALAI*

Refrigeration and Airconditioning Laboratory, Department of Mechanical Engineering,
Indian Institute of technology Madras, Chennai, Tamilnadu, India
Phone: +91 44 22574666; Fax: +91 44 22570509; E-mail: mania@iitm.ac.in

* Corresponding Author

ABSTRACT

Renewed interest in vapour absorption refrigeration systems (VARS) has increased due to their potential for utilization of waste heat, solar energy, etc. To improve the coefficient of performance (COP) of these systems, it is necessary to study heat and mass transfer processes in absorption system components. This study describes the experiments carried out to analyse the heat exchangers performance of a vapour absorption refrigeration system of 1 TR capacity. A new combination of 1,1,1,2 Tetrafluoroethane (R134a)-Dimethyl formamide (DMF) is used as working fluid to overcome the limitations of well known working pairs, ammonia-water and lithium bromide-water. Plate heat exchangers are used as system components such as absorber, generator, evaporator, condenser and solution heat exchanger. Bubble absorption principle is employed in the absorber. Hot water is used as the heat source to supply heat to the generator. Experimentation has been conducted with a cooling capacity of 2 to 5 kW by varying the operating parameters viz., liquid refrigerant flow rate from 30 lph to 90 lph, solution flow rate from 160 lph to 1600 lph, hot water temperature from 67 to 95°C, cooling water temperature from 15 to 30°C. Water flow rates are maintained from 1250 to 2450 lph for hot water, from 1950 to 2050 lph for cooling water and 650 lph for chilled water. The following temperature range of operating parameters has been used, with generator temperature: 63 – 90 °C, condenser temperature: 17 – 32 °C, absorber temperature: 15 – 30 °C and evaporator temperature: -2.5 – 10 °C. Volumetric mass transfer coefficient and mass transfer effectiveness have been evaluated for the absorber. Heat transfer rate, heat transfer effectiveness and overall heat transfer coefficient have been evaluated for the absorber, generator and solution heat exchanger. Effect of circulation ratio on heat transfer rate, heat transfer effectiveness, mass transfer effectiveness, overall hear transfer coefficient and volumetric mass transfer coefficient are investigated.

1. INTRODUCTION

Accomplishing refrigeration requirements through low-grade waste heat is becoming popular. Intensive research has been focused in the absorption refrigeration technology, since it uses waste heat, solar energy, etc. as energy source. Many investigators have suggested environment friendly working fluid combinations to replace well known working pairs ammonia-water and lithium bromide-water for VARS, in order to overcome some of their limitations. Of these, R22-organic solvent based absorption refrigeration systems have been extensively studied by Fatouh and Srinivasa Murthy (1995), Karthikeyan *et al.* (1994, 1995) and Sujatha *et al.* (1997a, 1997b, 1999). They observed that, in addition to having certain advantages over ammonia-water and lithium bromide-water systems, R22 based VARS can operate using low temperature heat sources in the range of about 75 – 95°C. However, along with Chloro Fluoro Carbons (CFCs), Hydro Chloro Fluoro Carbons (HCFCs) are also covered by Montreal and Kyoto International Protocols and are being phased out. So environment friendly R134a based VARS are being investigated. Borde *et al.* (1991, 1995) investigated the possibility of using R134a as a refrigerant in combination with different organic absorbents such as di-methyl ether tetra-ethylene glycol (DMETEG), N-methyl ε-caprolactam (MCL) or di-methyl-ethyleneurea (DMEU). They concluded that overall performance of R134a-DMETEG was better than that of R134a-MCL or R134a-DMEU. Though, there was reduction in COP and increase in circulation ratio with R134a systems compared to R22 systems, the R134a-DMETEG combination can be used as a replacement for R22-based working fluids in absorption machines since R22 is being phased out. Songara *et al.* (1998a) carried out a comparative thermodynamic study of VARS working with R134a and R22. They observed that R22-based system yields

significantly better COP than the R134a system. However, since R134a system operates at much lower pressures than R22 system, the possibility exists to improve its COP by resorting to two-stage operation. Songara *et al.* (1998b) also analysed thermodynamic studies on double effect VARS working with R134a as refrigerant and Dimethyl acetamide (DMA) as absorbent. COP yielded from these systems is comparable to single stage R22-DMA systems, but at slightly higher heat source temperatures. They suggested cascaded systems to achieve sub-zero temperatures. Nezu *et al.* (2002) examined the possibility of testing R134a as a refrigerant in VARS with various organic solvents and showed that the R134a-DMA and the R134a-DMF systems are considered attractive as the working-fluid pairs for the absorption refrigeration system than other R134a-absorbent systems. Yokozeki (2005) studied theoretical performance of various refrigerant-absorbent pairs in a VARS cycle by the use of equations of state. Of these, R134a-DMF and R134a-DMA systems exhibit better performance, compared to other R134a-absorbent systems. Review of the literature revealed that experimental studies on a single stage VARS using R134a-DMF are scanty. Also the heat exchanging components used by the investigators in their experimental studies on VARS were mainly shell and tube heat exchangers and tube-in-tube heat exchangers. The present investigation is focused on heat and mass transfer studies on plate heat exchangers in a 1 TR capacity VARS using R134a-DMF.

2. EXPERIMENTAL SETUP

The schematic diagram of experimental setup has been shown in Fig. 1. The setup consists of VARS, hot water simulator, cooling load simulator, cooling water simulator, instruments, valves and control devices. The VARS consists of refrigerant circuit and solution circuit. In refrigerant circuit, R134a vapour coming from the generator storage tank is condensed in the condenser and accumulated in the receiver. Heat of condensation is removed by the cooling water. Liquid R134a from the receiver is expanded through throttle valve/capillary tube and evaporated in the evaporator by cooling the heat load. Chilled water acts as the heat load. In solution circuit, R134a vapour from the evaporator is absorbed in the absorber by weak DMF solution. The heat of mixing is removed by cooling water. Strong solution collected in the absorber storage tank is pumped by hermetically sealed multistage centrifugal pump through solution heat exchanger to the generator. R134a vapour boiled off in the generator is separated in the generator storage tank. Hot water is supplied as heat source for the generator. Weak solution remaining in the tank is sent through solution heat exchanger and pressure reducing valve to the absorber for absorption. The solution heat exchanger is provided to cool the weak solution before entering the absorber and preheat the strong solution before entering the generator. Hot water simulator consists of a hot water tank insulated with glass wool, electric heaters, pump, flow meter, PT100 sensor, PID temperature controller, contactor, piping and valves. It supplies hot water to the generator. Cooling water simulator consists of two nos. of R22 based vapour compression refrigeration (VCR) circuits of 3.4 TR capacity each, a cooling water tank insulated with expanded polyethylene (EPE) sheets, electric heaters, pump, flow meter, PT100 sensor, PID temperature controller, contactor, piping and valves.
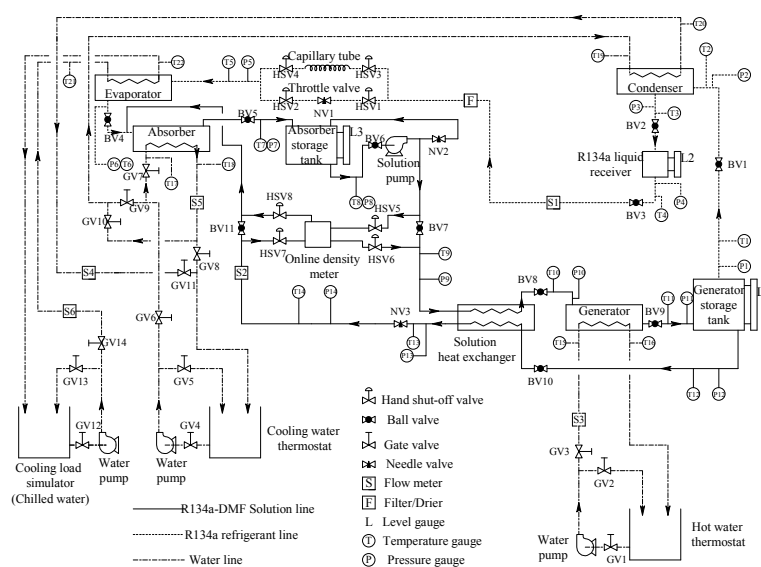

Fig. 1 Schematic diagram of R134a-DMF based vapour absorption refrigeration system

VCR circuit consists of hermetically sealed reciprocating compressor, air cooled condenser, thermostatic expansion valve and cooling coil. Cooling water thermostat supplies cooling water to the absorber and the condenser, which are connected in series as well as in parallel by cooling water pipelines. Cooling load simulator consists of a chilled water tank insulated with expanded polyethylene (EPE) sheets, electric heaters, pump, flow meter, PT100 sensor, PID temperature controller, contactor, piping and valves. It supplies water as heat load to the evaporator and maintains a constant desired value of chilled water temperature at evaporator inlet. The location of various temperature sensors, pressure sensors, flow meters and valves are indicated in Fig. 1. All these measuring instruments are pre-calibrated. Copper-constantan thermocouples of 22 numbers are used as temperature sensors with a measurement uncertainty up to $\pm$ 0.5 °C. Piezo-electric type pressure transducers of 14 numbers are used as pressure sensors with a measurement uncertainty up to $\pm$ 1.2 %. Metal tube rotameters are used to measure flow of liquid refrigerant, weak solution and hot water with a measurement uncertainty up to $\pm$ 4.6 %. Glass rotameters are used to measure flow of cooling water and chilled water with a measurement uncertainty up to $\pm$ 2.5 %. An online density meter is used to measure density of strong and weak solutions with a measurement uncertainty of $\pm$ 0.1 %. Concentrations of strong and weak solutions are evaluated from measured density values using HBT (Hankinson-Brobst-Thomson) equation used by Reid and others in their book (1989). Readings from all these instruments and sensors are monitored continuously by connecting them to a data acquisition system and a computer.

3. EXPERIMENTAL PROCEDURE

Initially the refrigerant and solution circuit are separated by closing the valve between the i) generator storage tank and condenser and the ii) evaporator and absorber. Absorbent charge is calculated based on system volume in solution circuit and charged at the absorber storage tank. Refrigerant charge is calculated based on system volume in both refrigerant and solution circuits. It is charged and mixed with absorbent at the absorber storage tank by circulating the solution through solution circuit using solution pump. Heat of absorption is allowed to be transferred from the solution circuit components to ambient air. Solution pump is switched off after charging. Hot water simulator and cooling water simulator are started. Hot water is circulated through generator at a temperature higher than that to be maintained in the generator. Cooling water is circulated through absorber and condenser in parallel or series circuit at a temperature lower than that to be maintained in both the components. Cooling load simulator is operated by circulating water through the evaporator. Water temperature in the chilled water tank is maintained constant by switching on heaters equivalent to cooling capacity of system.

Solution pump is then started to circulate strong solution through the generator. Level of weak solution collected in the generator storage tank, level of strong solution in the absorber storage tank and pressure in each component of solution circuit are monitored continuously. When pressure in the generator storage tank becomes greater than that in the condenser, the valve between the generator storage tank and the condenser is opened to allow refrigerant vapour to enter the condenser. Level of liquid refrigerant collected in the liquid refrigerant receiver is monitored continuously. When sufficient amount of refrigerant is stored, it is admitted though expansion devices to enter the evaporator. The valve between the evaporator and absorber is opened to allow refrigerant vapour to enter the absorber. Flow rates of weak solution and liquid refrigerant are regulated to maintain steady flow in the system. System is run continuously by monitoring pressure transducer, thermocouple, flow meter and level gauge readings at various locations. When all these readings remain constant over a period of time, it indicates that system has attained steady state operating conditions and all these readings are recorded in the computer. Water flow rates in the hot water simulator, cooling water simulator and cooling load simulator are maintained constant at the desired value. While shutting down the system after recording the readings, solution circuit and refrigerant circuit are isolated by closing the valve between the evaporator and the absorber and then closing the valve between the generator storage tank and the condenser. Then solution pump is switched off.

4. RESULTS AND DISCUSSION

Experimentation was conducted with a cooling capacity of 2 to 5 kW by varying the operating parameters viz., liquid refrigerant flow rate from 30 lph to 90 lph, solution flow rate from 160 lph to 1600 lph, hot water temperature from 67 to 95°C, cooling water temperature from 15 to 30°C. Water flow rates were maintained constant at 2450 lph for hot water, 2000 lph for cooling water in series flow path, 1250 lph for parallel flow path and 650 lph for chilled water. The parametric studies were carried out from the following temperature range of operating parameters:

Generator temperature: 63 – 90 °C, condenser temperature: 17 – 32 °C, absorber temperature: 15 – 30 °C, evaporator temperature: -2.5 – 10 °C. Cooling water can be supplied to absorber and condenser in parallel as well as series flow arrangements as in Fig. 1. Though parallel flow of cooling water supply gives better performance, it requires large quantity of water which is not met by cooling water pump in the system. Hence system is run at lower capacities with parallel flow and at higher capacities with series flow. In series flow arrangement, cooling water first goes to absorber and then to condenser.

Figure 2 shows the effect of circulation ratio (CR) on absorber heat load. Absorber heat load increases as CR increases at constant absorber temperature. As CR increases, solution flow rate through absorber increases thereby increasing the heat rejection.
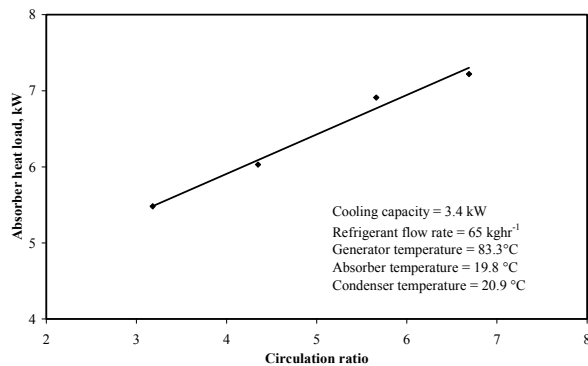
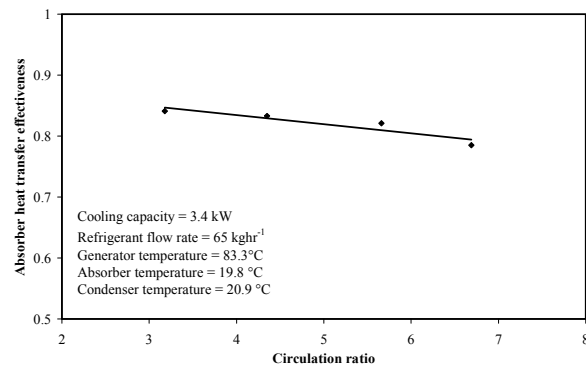Fig. 2 Effect of circulation ratio on absorber heat load

Fig. 3 Effect of circulation ratio on absorber heat transfer effectiveness
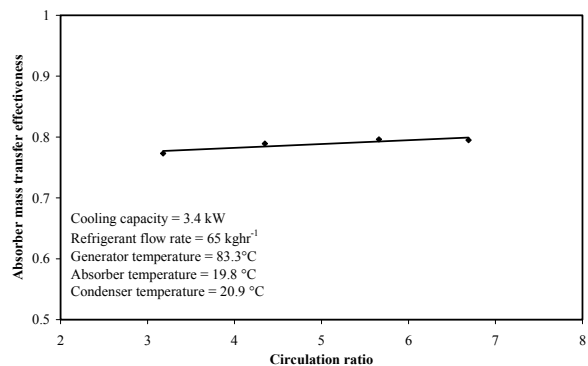

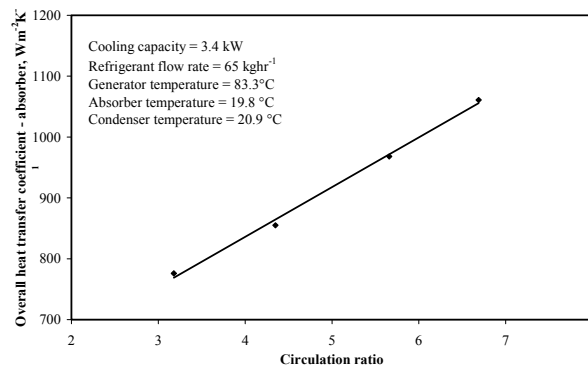Fig. 4 Effect of circulation ratio on absorber mass transfer effectiveness

Fig. 5 Effect of circulation ratio on overall heat transfer coefficient in the absorber

Figures 3 and 4 show the effect of CR on absorber heat transfer and mass transfer effectiveness, respectively. Both the heat transfer effectiveness and mass transfer effectiveness do not vary much with respect to CR. Figure 5 presents the effect of CR on overall heat transfer coefficient in the absorber. Overall heat transfer coefficient increases as CR increases at constant absorber temperature, due to increase in absorber heat load at high CR. Figure 6 presents the effect of CR on volumetric mass transfer coefficient in the absorber. Volumetric mass transfer coefficient increases as CR increases. The reason is due to decrease in log mean concentration difference (LMCD) at high CR. LMCD decreases at high CR due to decrease in strong and weak solution concentration difference.
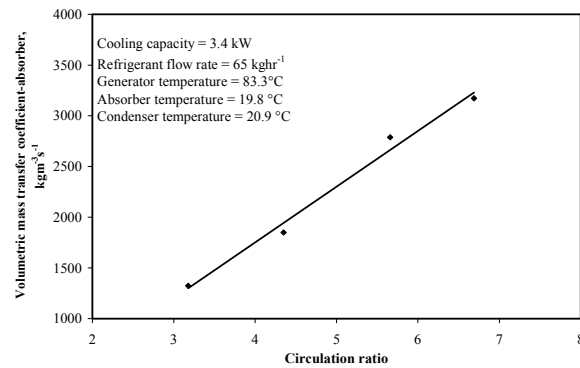
Fig. 6 Effect of circulation ratio on volumetric mass transfer coefficient in the absorber

Figure 7 shows the effect of CR on generator heat load. Generator heat load increases as CR increases at constant generator temperature. As CR increases, solution flow rate through generator increases thereby increasing the heat load.
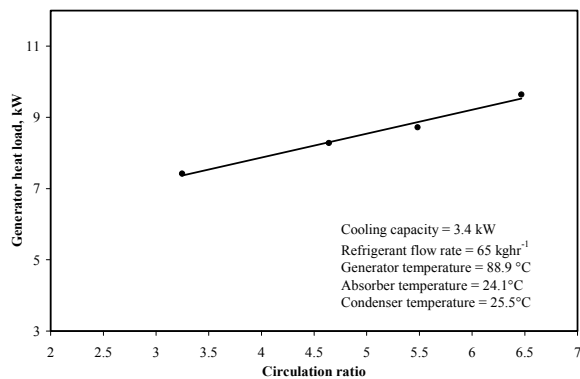

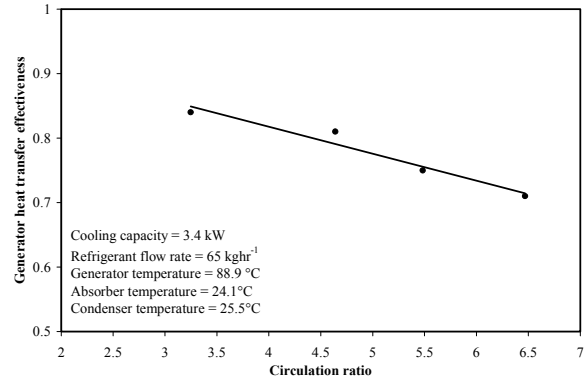Fig. 7 Effect of circulation ratio on generator heat load

Fig. 8 Effect of circulation ratio on generator heat transfer effectiveness
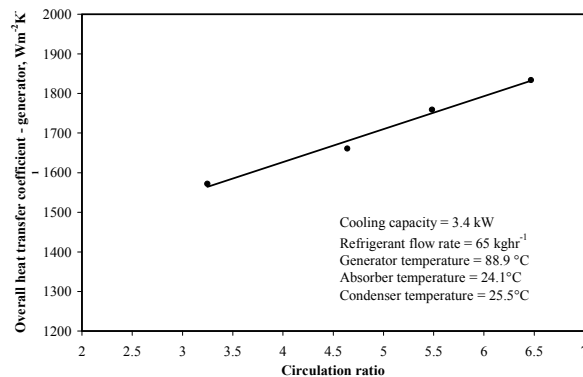

Fig. 9 Effect of circulation ratio on overall heat transfer coefficient in the generator

Figure 8 presents the effect of circulation ratio on generator heat transfer effectiveness. The heat transfer effectiveness is better at lower CR. Figure 9 presents the effect of CR on overall heat transfer coefficient. Overall heat transfer coefficient increases as CR increases. The reason is due to increase in generator load at high CR. At high CR, due to high solution flow rates, flow velocity and heat transfer coefficient are more, resulting in higher generator heat load.

Figure 10 shows the effect of CR on solution heat exchanger heat load. The heat load increases as CR increases. The increase in heat load at high CR is due to higher solution flow rates.
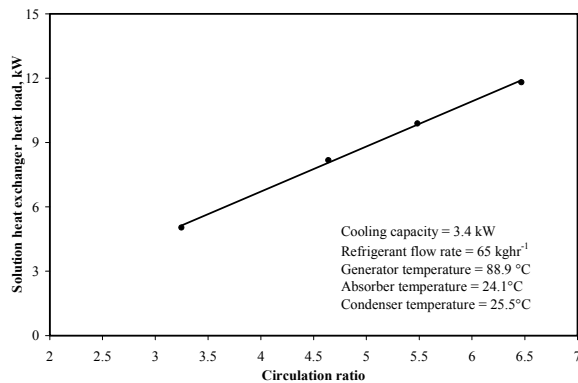

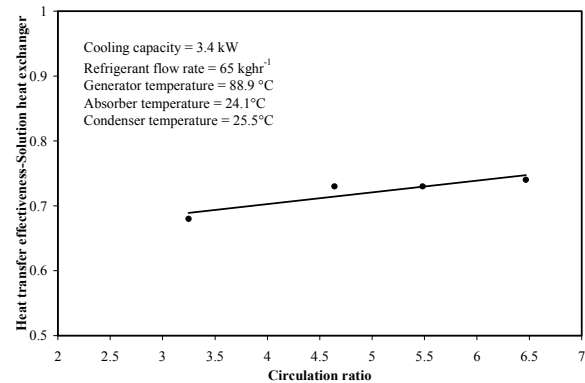Fig. 10 Effect of circulation ratio on solution HX heat load

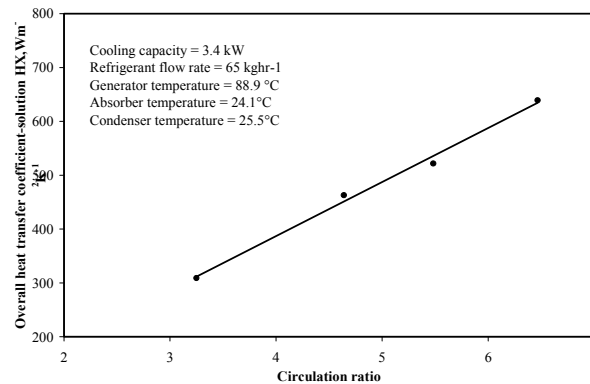Fig. 11 Effect of circulation ratio on solution HX heat transfer effectiveness

Fig. 12 Effect of circulation ratio on overall heat transfer coefficient in the solution heat exchanger

Figure 11 shows the effect of circulation ratio on heat transfer effectiveness in the solution heat exchanger. The heat transfer effectiveness does not vary much with respect to CR. Figure 12 shows the effect of CR on overall heat transfer coefficient in the solution heat exchanger. Overall heat transfer coefficient increases with respect to CR due to increase in heat load at high CR. At high CR, due to high solution flow rates, flow velocity and heat transfer coefficient are more, resulting in higher heat load in the solution heat exchanger. Figures 13 and 14 depict the variation of LMTD and overall heat transfer coefficient with respect to CR, respectively, for various components of VARS system.
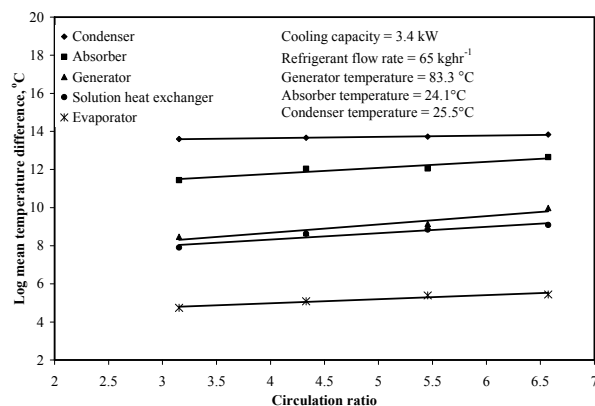
Fig. 13 Effect of circulation ratio on LMTD for various components of VARS system
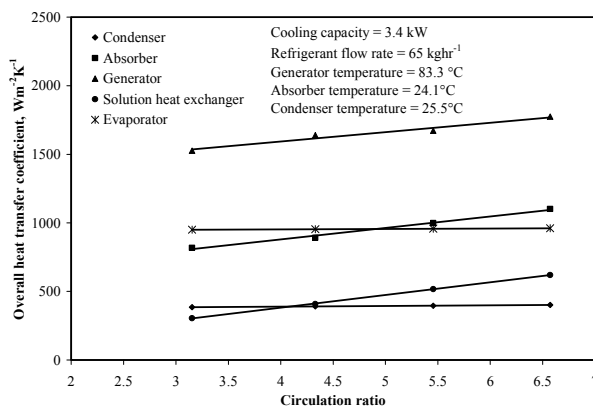

Fig. 14 Effect of circulation ratio on overall heat transfer coefficient for various components of VARS system

5. CONCLUSIONS

Experimental investigations have been carried out on a 1 TR capacity vapour absorption refrigeration system to study heat and mass transfer characteristics with plate heat exchangers as system components viz., absorber, generator, evaporator, condenser and solution heat exchanger. R134a-DMF is used as working fluid. The following parameters have been determined from the experiments: Volumetric mass transfer coefficient and mass transfer effectiveness for absorber; heat transfer rate, heat transfer effectiveness and overall heat transfer coefficient for absorber, generator and solution heat exchanger. The effect of circulation ratio on these parameters has been evaluated. The following conclusions are drawn from the present study.

- *Absorber*: Heat transfer rate, overall heat transfer coefficient and volumetric mass transfer coefficient increase as circulation ratio increases. Heat transfer effectiveness and mass transfer effectiveness do not vary much as circulation ratio increases.
- *Generator*: Heat transfer rate and overall heat transfer coefficient increase as circulation ratio increases. Heat transfer effectiveness is better at lower circulation ratios.
- *Solution heat exchanger*: Heat transfer rate and overall heat transfer coefficient increase as circulation ratio increases. Heat transfer effectiveness does not vary much as circulation ratio increases.

NOMENCLATURE

C_p	specific heat at constant pressure	$(kJkg^{-1}K^{-1})$	**Subscripts**	
d	channel diameter of plate heat exchanger	(m)	eq	equilibrium
L	channel length of plate heat exchanger	(m)	h	heat transfer
LMCD	log mean concentration difference	(-)	in	heat exchanger inlet
LMTD	log mean temperature difference	(-)	m	mass transfer
m	mass flow rate	(kgs^{-1})	out	heat exchanger outlet
MTC	mass transfer coefficient	$(kgm^{-3}s^{-1})$	r	refrigerant
Q	heat transfer rate	(kW)	s	solution
T	Temperature	(°C, K)	ss	strong solution
U	overall heat transfer coefficient	$(Wm^{-2}K^{-1})$	V	volumetric
X	solution mass fraction	$(kgkg^{-1})$	w	water
ε	effectiveness	(-)	ws	weak solution

REFERENCES

Borde, I., Jelinek, M. and Daltrophe, N. C., 1995, Absorption system based on the refrigerant R134a, *International Journal of Refrigeration*, vol. 18, no. 6: p. 387 - 394.

Fatouh, M. and Srinivasa Murthy, S., 1995, Performance of a HCFC22 based vapour absorption refrigeration system, *International Journal of Refrigeration*, vol. 18, no. 7: p. 465 - 476.

Karthikeyan, G., Mani, A., Srinivasa Murthy, S., 1994, Analysis of an absorption refrigeration system with transfer tank, *Renewable energy*, vol. 4, no. 1: p. 129 - 132.

Karthikeyan, G., Mani, A., Srinivasa Murthy, S., 1995, Performance of different working fluids in transfer-tank operated vapour absorption refrigeration systems, *Renewable energy*, vol. 6, no. 7: p. 835 - 842.

Songara, A. K., Fatouh, M., and Srinivasa Murthy, S, 1998, Comparative performance of HFC134a- and HCFC22- based vapour absorption refrigeration systems, *International Journal of Energy Research*, vol. 22, no. 4: p. 363 – 372.

Songara, A. K., Fatouh, M., and Srinivasa Murthy, S, 1998, Thermodynamic studies on HFC134a-DMA double effect and cascaded absorption refrigeration systems, *International Journal of Energy Research*, vol. 22, no. 7: p. 603 – 614.

Sujatha, K.S., Mani, A., and Srinivasa Murthy, S., 1997, Finite element analysis of a bubble absorber, *International Journal for Numerical Methods for Heat & Fluid Flow*, vol. 7, no. 7: p. 737–750.

Sujatha, K.S., Mani, A., and Srinivasa Murthy, S., 1997, Analysis of a bubble absorber working with R22 and five organic absorbents, *Heat and Mass Transfer*, vol. 32, no. 4: p. 255–259.

Sujatha, K.S., Mani, A., and Srinivasa Murthy, S., 1999, Experiments on a bubble absorber, *International Communications in Heat and Mass Transfer*, vol. 26, no. 7: p. 975–984.

Yokozeki, A., 2005, Theoretical performances of various refrigerant-absorbent pairs in a vapour absorption refrigeration cycle by the use of equation of state, *Applied Energy*, vol. 80, no. 4: p. 383 – 399.

Borde, I., Jelinek, M. and Daltrophe, N. C., 1991, Refrigerant-absorbent mixtures based on the refrigerant R134a, *Proc. XVIII th Int. Conf. of Ref.*, Montreal, Canada, p. 653 - 658.

Nezu, Y., Hisada, N., Ishiyama, T. and Watanabe, K., 2002, Thermodynamic properties of working-fluid pairs with R-134a for absorption refrigeration system. *In: Natural Working-Fluids 2002, IIR Gustav Lorentzen Conf. 5th.*, China, Sept. 17–20, 446 - 453.

Robert C. Reid, John M. Prausnitz, Bruce E. Poling, 1989, *The Properties of Gases and Liquids*, Fourth edition, McGraw-Hill Book Company.

Appendix A

Circulation ratio = strong solution mass flow rate/refrigerant vapour mass flow rate   (1)

$$LMCD = \frac{(X_{ws} - X_{eq,ws}) - (X_{eq,ss} - X_{ss})}{\ln\left(\dfrac{X_{ws} - X_{eq,ws}}{X_{eq,ss} - X_{ss}}\right)} \quad (2) \qquad MTC_V = \frac{m_r}{\left(\dfrac{\pi}{4} d^2 L\right) LMCD} \quad (3)$$

$$Q = m_w Cp_w (T_{w,out} - T_{w,in}) \quad (4) \qquad LMTD = \frac{(T_{s,in} - T_{w,out}) - (T_{s,out} - T_{w,in})}{\ln\left(\dfrac{(T_{s,in} - T_{w,out})}{(T_{s,out} - T_{w,in})}\right)} \quad (5)$$

$$U = \frac{Q}{(\pi dL) LMTD} \quad (6) \qquad \varepsilon_m = \frac{X_{ss} - X_{ws}}{X_{eq,ss} - X_{ws}} \quad (7) \qquad \varepsilon_h = \frac{Q}{Q_{max}} \quad (8)$$