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Analysis of R134a–DMAC vapour absorption refrigeration system with add-on components

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Abstract

R134a–DMAC vapour absorption refrigeration system (VARS) needs rectifier. Because of incomplete rectification, a small amount of DMAC is carried to evaporator. It results in a temperature gradient and also formation of residual liquid in the evaporator. This liquid causes cooling loss which can be recovered significantly by using liquid vapour heat exchanger (LVHX). The same cooling temperature can be attained at a range of evaporator pressures due to the temperature gradient. For fixed cooling temperature, the system COP enhances with evaporator pressure. The enhancement rate is more when efficiencies of rectifier and LVHX are high. Efficient solution heat exchanger (SHX) is vital owing to its large heat duty. Rectifier loses its importance if high efficiency LVHX is used. Roles of these three components increase at low cooling and high sink temperatures.

Keywords: R134a–DMAC VARS; Rectifier; LVHX; SHX; Evaporator pressure

1. Introduction

Importance of vapour absorption refrigeration system (VARS) cannot be overestimated nowadays when the world is facing energy related problem. Input to VARS is predominantly heat energy which may be available in abundance in many situations in the form of waste heat from industry or solar heat. Due to scarcity and increased cost of electricity VARS is gradually gaining importance. The most commonly used refrigerant–absorbent pairs are ammonia–water and water–lithium bromide. Attention was drawn to new refrigerants in order to overcome the limitations of these conventional working fluid pairs. The most important criteria for refrigerant and absorbent pair are thermal and chemical stability and large solubility of the refrigerant in the absorbent (Iedema, 1982). Environmental issues rule out the possibility of making new refrigerants using chlorine and bromine (Kopko, 1990). R134a (1, 1, 1, 2 tetrafluoro ethane) is a refrigerant which can be used for cooling at sub-zero temperature and where ammonia is forbidden. It has zero ozone depletion potential. DMAC (*N*, *N*dimethyl acetamide) is a good absorbent for R134a (Songara et al., 1998). Experiments were conducted on R134a–DMAC based VARS to show the feasibility of using the pair when temperature of heat

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Nomenclature

c	specific heat (kJ kg^{-1} K^{-1})
COP	coefficient of performance
h	enthalpy (kJ kg^{-1})
h_{fg}	latent heat of evaporation (kJ kg^{-1})
LVHX	liquid vapour heat exchanger
m	mass flow rate (kg s^{-1})
p	pressure (bar)
Q	rate of heat transfer (kW)
SHX	solution heat exchanger
VARS	vapour absorption refrigeration system
t, T	temperature (°C, K)
W	work input (kW)

Subscripts

a	absorber
c	condenser
e	evaporator
eq	equivalent
eV	vapour at evaporator outlet
eV′	vapour at LVHX outlet
g	generator
h	hot
l	liquid
lvhx	liquid vapour heat exchanger
max	maximum
min	minimum
r	rectifier
rl	residual liquid at evaporator outlet
rl′	residual liquid at evaporator outlet
s	strong solution
shx	solution heat exchanger
v	vapour
v′	vapour at outlet of the rectifier
v″	vapour at bottom of generator
w	weak solution
1…9	state points as shown in Fig. 1

Greek letters

ξ	mass fraction of R134a
ρ	density (kg m^{-3})
η	efficiency
λ	circulation ratio

source is low (Muthu et al., 2008). For cold storage application with low temperature heat source, two-stage half effect R134a–DMAC VARS was theoretically analysed (Arivazhagan et al., 2005). Thirteen percentage increase in system COP by inclusion of liquid vapour heat exchanger (LVHX) was claimed. During experimental study of similar system, −7 °C temperature was achieved with the source temperature in the range of 55–75 °C with a maximum COP of 0.36 (Arivazhagan et al., 2006). The high and low pressures of the circuit were decided by condenser and evaporator temperatures, respectively. The intermediate pressure was decided by simulation of the system.

Hitherto all the vapour absorption systems with new refrigerants have been analysed without rectifier as it was assumed that absorbent does not boil out in the generator. Hence use of rectifier has been confined only in ammonia–water VARS. Consequently in case of new refrigerants residual liquid has not been considered in the liquid vapour heat exchanger (LVHX). As the difference in boiling point temperatures of R134a and DMAC is only about 191 °C, in generator the latter also boils along with the former, particularly at high generator temperature which is required for low cooling and/or high sink temperatures. Depending on the rectifier efficiency some amount of DMAC escapes to condenser and results in residual liquid in the evaporator. Hence the role of LVHX becomes significant. Besides, a temperature gradient exists in the evaporator owing to the presence of DMAC. Hence, the same cooling temperature can be attained at a range of evaporator pressures. In addition, role of solution heat exchanger (SHX) becomes very vital in this system owing to its large heat duty. Hence in the present study, performance of R134a–DMAC VARS with two add-on components namely rectifier and LVHX, has been evaluated by thermodynamic analysis to find the role of rectifier, LVHX and SHX at different operating conditions and influence of evaporator pressure for the same cooling temperature.

2. System description

Fig. 1 represents the schematic of R134a–DMAC VARS analysed here. The working principle is mostly similar to the conventional system (Roy, 2011). Hence, for conciseness the difference between the two with focus on add-on components namely rectifier and LVHX are discussed here.

Due to less difference between normal boiling point temperatures of R134a and DMAC (191 °C), a little amount of the latter also boils along with the former particularly at high generator temperature. In the rectifying section, the vapour rejects some amount of heat in the reflux condenser and partially condenses. The condensed liquid (reflux) flows down due to gravity. Heat and mass transfer take place between the ascending vapour and the descending liquid. Then from the liquid phase, R134a vapourises due to its lower boiling point temperature and from the vapour phase DMAC condenses due to its higher boiling point temperature. Thus, R134a is transferred from liquid to vapour phase and DMAC is transferred from vapour to

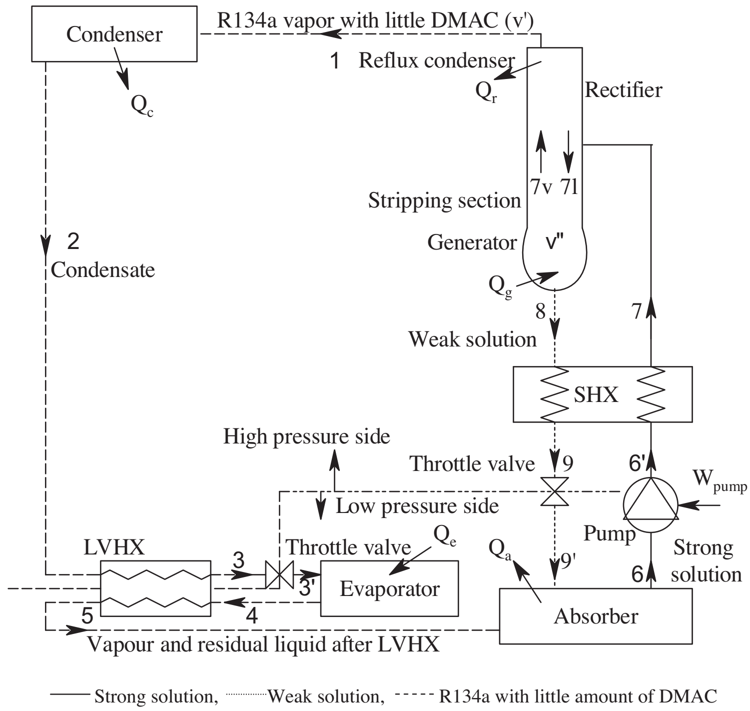

Fig. 1. Schematic of R134a–DMAC VARS with rectifier and LVHX.

liquid phase. In this way, mass fraction of R134a in vapour phase increases progressively from bottom to top. In the same way, DMAC is removed to some extent in stripping section and mixes with incoming strong solution. Depending on the rectifier efficiency, the vapour contains some amount of DMAC after rectification. Owing to higher boiling point temperature, all DMAC condense along with R134a in the condenser.

Owing to the presence of small amount of DMAC, the system differs in evaporation process from the conventional systems. A temperature gradient is formed along the length of the evaporator, as mass composition of this type of mixture varies continuously when it boils. The temperature at the outlet of evaporator is the highest and is considered as the desired cooling temperature. Hence, unlike the conventional system here any cooling temperature can be attained at a range of evaporator pressures considering flexible temperature gradient in the evaporator. DMAC does not vapourise in the evaporator due to its high boiling point temperature resulting in formation of residual liquid. This liquid contains significant amount of un-boiled R134a leading to apparent cooling loss which is defined as the amount of additional cooling that would be obtained if all the R134a in the residual liquid at evaporator outlet is completely boiled.

The apparent cooling loss can be recovered to a large extent by using LVHX between vapour and residual liquid from evaporator and incoming condensate. In LVHX, most of R134a boils from the residual liquid. This boiling and R134a vapour sub-cools the incoming condensate which yields an equal amount of extra cooling in the evaporator.

3. Mathematical model

The additional parameters to be considered in the present mathematical model are efficiencies of rectifier and LVHX and variation of evaporator pressure for the same cooling temperature. Hence, known parameters for this model are operating temperatures (generator, condenser, evaporator and absorber), evaporator (low side) pressure and efficiencies of SHX, LVHX and rectifier. However, for the sake of completeness and clarity of the model all the general equations are also presented along with the newly derived equations. All connecting pipes are assumed to be adiabatic with negligible pressure drop. Mass transfer efficiencies of generator and absorber are assumed to be unity which implies that vapour and liquid are in equilibrium at their outlets. The required thermo-physical properties of R134a–DMAC liquid mixture are obtained from the literature, viz. viscosity, enthalpy and density from Borde and Stephan (1993), thermal conductivity, specific heat and diffusivity from Reid et al. (1987) and the constants for estimation of specific heat from Yokozeki (2005). The properties of the gas phase (pure R134a) have been obtained by "REFPROP" software (Lemmon, 2002).

A small amount of DMAC present in R134a will not have any significant effect on condensation. Therefore, the high side pressure is decided only by condenser temperature. Hence,

$$p_1 = p_2 = p_3 = p_{6'} = p_7 = p_8 = p_9 = p_c = p_g = f(t_c) \tag{1}$$

The weak solution mass fraction ξ_w is determined as

$$\xi_w = f(p_g, t_g) \tag{2}$$

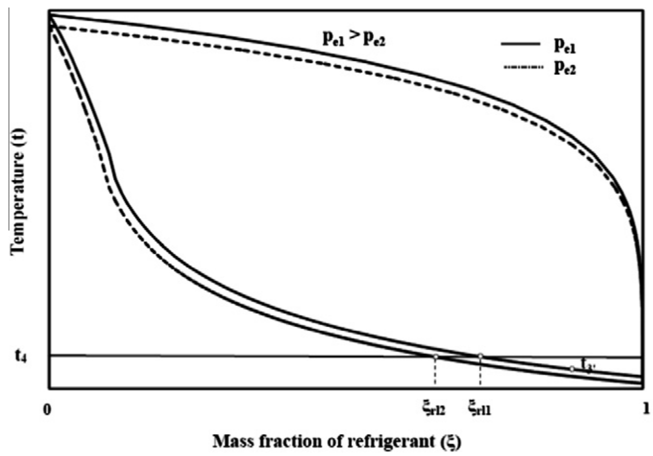

Fig. 2. The same cooling temperature at different evaporator pressures.

The evaporator (low side) pressure ($p_{3'} = p_4 = p_5 = p_6 = p_{9'} = p_e = p_a$) is selected (as a known operating parameter) with the maximum and minimum limits was follows:

(a) The strong solution mass fraction ξ_s decided by absorber temperature and low side pressure [$\xi_s = f(t_a, p_a)$] should be greater than ξ_w. Thus minimum value of low side pressure is decided as

$$P_{e,\min} = f(\xi_w, t_a) \tag{3}$$

(b) The equilibrium temperature of condensate at evaporator inlet, decided by its mass fraction (same as that at rectifier outlet) and evaporator pressure should be less than the desired cooling temperature. State 3′ in Fig. 2 represents inlet condition of condensate with mass fraction $\xi_{v'}$ at evaporator pressure p_{e1}, then corresponding inlet temperature becomes $t_{3'}$. At the outlet, its temperature increases to t_4 due to temperature gradient in the evaporator. t_4 is considered as the desired cooling temperature t_e. Hence, $t_{3'}$ needs to be less than t_4. But $t_{3'}$ increases with evaporator pressure for any fixed $\xi_{y'}$. Considering this aspect the maximum value of low side pressure is decided as,

$$P_{e,\max} = f(\xi_{v'}, t_e) \tag{4}$$

Rectifier efficiency is defined as,

$$\eta_r = (\xi_{v'} - \xi_{v''})/(1 - \xi_{v''}) \tag{5}$$

Thus, mass fraction of vapour after rectifier is obtained as,

$$\xi_{v'} = \eta_r(1 - \xi_{v''}) + \xi_{v''} \tag{6}$$

The corresponding temperature and enthalpy are found from p–t–ξ and h–t–ξ relations, respectively.

The circulation ratio (λ) is defined as the ratio of mass flow rates of weak solution to emitted refrigerant vapour (containing a small amount of DMAC). Thus,

$$\lambda = (\xi_{v'} - \xi_s)/(\xi_s - \xi_w) \tag{7}$$

Mass flow rate of vapour from rectifier outlet is $m_{v'}$

$$m_1 = m_2 = m_3 = m_{3'} = m_4 = m_5 = m_{v'} \tag{8}$$

By definition of circulation ratio

$$m_8 = m_9 = m_{9'} = m_w = \lambda m_{v'} \tag{9}$$

Similarly $m_6 = m_{6'} = m_7 = m_s = (\lambda + 1)m_{v'}$ (10)

Enthalpy of weak solution at outlet of generator is

$$h_8 = f(t_g, \xi_w) \tag{11}$$

Weak solution has always less thermal capacity and hence it can be cooled to absorber temperature, if efficiency of SHX is 1. Hence,

$$h_{9,\min} = f(t_a, \xi_w) \tag{12}$$

Enthalpy of strong solution at absorber outlet is

$$h_6 = f(t_a, \xi_s) \tag{13}$$

Ideal work input to the pump is given by

$$W = \frac{m_s}{\rho}(p_c - p_e)/100 \tag{14}$$

Enthalpy of strong solution at the inlet of SHX is

$$h_{6'} = h_6 + h_{eq}/m_s \tag{15}$$

Enthalpy of strong solution at the outlet of SHX (State 7) is found by enthalpy balance across SHX. Hence,

$$h_7 = h_{6'} + \eta_{shx}(h_8 - h_{9,\min})m_w/m_s \tag{16}$$

SHX loss is defined as,

$$\mathrm{loss}_{shx} = m_w(1 - \eta_{shx})(h_8 - h_{9,min}) \tag{17}$$

It is assumed that temperature of rising vapour (State 7v) from stripping section and falling liquid (State 7l) from the rectifier are the same which equals to the temperature of feed to generator, t_7. Heat released in the rectifier is obtained according to Gosney (1982) by mass, material (R134a) and enthalpy balance equations across the rectifier

$$m_{7v} = m_{7l} + m_{v'} \tag{18}$$

$$m_{7v}\xi_v = m_{7l}\xi_l + m_{v'}\xi_{v'} \tag{19}$$

$$m_{7v}h_v = m_{7l}h_l + m_{v'}h_{v'} + Q_r \tag{20}$$

By energy balance across generator, the heat input is found as

$$Q_g = m_{v'} \times h_{v'} + m_w \times h_g + Q_r - m_s \times h_7 \tag{21}$$

The mass fraction of residual liquid and evaporated vapour is found from evaporator temperature and pressure using p–t–ξ relations. The mass of residual liquid and evaporated vapour is found by total mass and material (R134a) balance across the evaporator

$$m_{rl} + m_{eV} = m_{v'} \tag{22}$$

$$m_{rl}\xi_{rl} + m_{eV}\xi_{eV} = m_{v'}\xi_{v'} \tag{23}$$

Solving Eqs. (22) and (23)

$$m_{rl} = m_{v'}(\xi_{eV} - \xi_{v'})/(\xi_{eV} - \xi_{rl}) \tag{24}$$

In LVHX considerable amount of R134a boils from the residual liquid of cold stream (residual liquid and vapour), supplementing the sub-cooling of the hot stream (condensate). So quality of the cold stream varies significantly. Depending on the quality of the residual liquid, stream of minimum heat capacity rate is decided. The pinch point shifts from one end to the other end of LVHX. For any operating condition, maximum possible heat transfer for both hot and cold streams has to be calculated assuming the terminal temperature difference of LVHX to be zero (when any one of the two streams attains the inlet temperature of the other depending on the minimum heat capacity rate). Minimum of these two amounts of heat transfer and LVHX efficiency decide the actual sub-cooling. These two amounts of heat transfer are obtained as follows:

If hot stream has minimum thermal capacity, it can be cooled to te when effectiveness of LVHX is unity. Maximum possible heat release by hot stream is

$$Q_{\mathrm{lvhx,max}} = c_{\mathrm{h}}(t_{\mathrm{c}} - t_{\mathrm{e}})m_{\mathrm{v}'} \tag{25}$$

The following procedure and set of equations are used to find maximum possible heat transfer when the cold stream has minimum thermal capacity. Here, it can be heated up to condenser temperature (t_{c}) at the exit (State 5) when effectiveness of LVHX is unity.

The mass fraction of residual liquid at LVHX outlet is found by condenser temperature and low side pressure using p–t–ξ relations. In LVHX, some R134a boils. At the outlet, the mass of residual liquid is found by material balance of DMAC. Hence,

$$\xi_{\mathrm{rl}'} = f(t_{\mathrm{c}}, p_{\mathrm{e}}) \tag{26}$$

$$m_{\mathrm{rl}}(1 - \xi_{\mathrm{rl}}) = m_{\mathrm{rl}'}(1 - \xi_{\mathrm{rl}'}) \tag{27}$$

At the exit (State 5) state the mass of R134a vapour is obtained by total mass balance across LVHX

$$m_{\mathrm{eV}'} = m_{\mathrm{rl}} + m_{\mathrm{eV}} - m_{\mathrm{rl}'} \tag{28}$$

Maximum possible heat absorbed by cold stream is obtained by total enthalpy balance across LVHX. The enthalpies of vapour and residual liquid at inlet and outlet of LVHX are found from h–t–ξ relations.

$$Q_{\mathrm{lvhx,max}} = (m_{\mathrm{rl}'}h_{\mathrm{rl}'} + m_{\mathrm{eV}'}h_{\mathrm{eV}'}) - (m_{\mathrm{rl}}h_{\mathrm{rl}} + m_{\mathrm{eV}}h_{\mathrm{eV}}) \tag{29}$$

Actual amount of heat transfer in LVHX is obtained as

$$Q_{\mathrm{lvhx,actual}} = \min(Q_{\mathrm{lvhx,25}}, Q_{\mathrm{lvhx,29}}) \times \eta_{\mathrm{lvhx}} \tag{30}$$

Enthalpy of condensate leaving LVHX (State 3) is given by

$$h_3 = h_2 - Q_{\mathrm{lvhx}}/m_{\mathrm{v}'} \tag{31}$$

Amount of heat transfer in evaporator is calculated by enthalpy balance among vapour, residual liquid and condensate across the evaporator. Enthalpy of vapour and residual liquid at the outlet of evaporator is found from h–t–ξ relations.

$$Q_{\mathrm{e}} = m_{\mathrm{rl}}h_{\mathrm{rl}} + m_{\mathrm{eV}}h_{\mathrm{eV}} - m_3h_3 \tag{32}$$

In the absence of LVHX, apparent cooling loss due to un-boiled R134a entrained in residual liquid is determined as

$$Q_{\mathrm{loss}} = m_{\mathrm{rl}}\xi_{\mathrm{rl}}h_{\mathrm{fg}} \tag{33}$$

Coefficient of performance (COP) of the system is defined as

$$\mathrm{COP} = \frac{Q_{\mathrm{e}}}{Q_{\mathrm{g}} + W} \tag{34}$$

4. Results and discussion

The results are obtained for operating parameters outlined in Table 1. Each of the variable operating parameters is varied in its range of variation keeping the rest at their respective mean values. The performance of the system has been found in the form of apparent cooling loss and COP with an emphasis on efficiencies of rectifier, LVHX and SHX and effect of evaporator pressure variation for the same cooling temperature.

4.1. Effect of operating temperatures

Fig. 3 represents apparent cooling loss (in the absence of LVHX) per unit refrigeration capacity with respect to generator temperature for various cooling and sink temperatures. At high generator temperature, a large amount of DMAC boils along with R134a. Hence, the mass fraction of the refrigerant of vapour at the bottom of generator ($\xi_{\mathrm{v}''}$) decreases and consequently, it diminishes the mass fraction of vapour at its outlet ($\xi_{\mathrm{v}'}$) for the same rectifier efficiency according to Eq. (6). As a result, at evaporator outlet, amount of residual liquid per unit mass of condensate increases with generator temperature, owing to corresponding increase of numerator term in Eq. (24). Consequently, apparent cooling loss per unit mass of condensate enhances. Hence, more condensate flow is required to achieve the same refrigeration capacity. As a result, apparent cooling loss per unit refrigeration capacity enhances with generator temperature owing to the combined effect of these two reasons. The figure also indicates that, the loss becomes less for a high cooling temperature. Mass fraction of residual liquid at evaporator outlet diminishes with an increase in the cooling temperature. This leads to a decrease in the mass of residual liquid according to Eq. (24), because the reduced value of ξ_{rl} increases the denominator term in the equation. Consequently, the apparent cooling loss diminishes with an increase in cooling temperature because of reduction of both the terms m_{rl} and ξ_{rl} in Eq. (33). The figure further reveals that, for any given generator temperature, the apparent cooling loss increases with a decrease in sink (condenser and absorber) temperature. At low sink temperature, condensing pressure and hence generator pressure is less. Therefore, mass fraction of R134a in vapour at generator bottom and, consequently, that at rectifier outlet reduce. Hence, in a similar

Table 1
Range of operating parameters.

Sl. no.	Parameter	Notation (unit)	Range	Mean
1	Generator temperature	t_g (°C)	110–170	Optimum or140[a]
2	Sink temperature	$t_a = t_c$ (°C)	25–35	30
3	Cooling temperature (temperature at the exit of evaporator)	t_e (°C)	−10 to 10	−10 or 0[b]
4	Evaporator (low side) pressure	$p_e = p_a$ (bar)	$p_{e,min}$ to $p_{e,max}$[c]	$p_{e,\ max}$
5	LVHX efficiency	η_{lvhx} (−)	0–1	0.7
6	Rectifier efficiency	η_{rec} (−)	0.3–1	0.9
7	SHX efficiency	η_{shx} (−)	0–1	0.7

[a] $t_g = 140$ °C for Figs. 12 and 13 and t_g = optimum for rest of the figures.
[b] $t_e = -10$ °C for Figs. 9–13 and $t_e = 0$ °C for rest of the figures.
[c] As described in Eqs. (3) and (4).

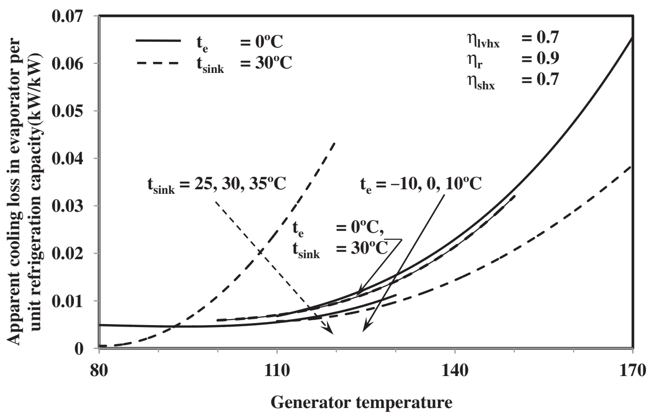

Fig. 3. Variation of apparent cooling loss in evaporator with respect to generator temperature for various cooling and sink temperatures.

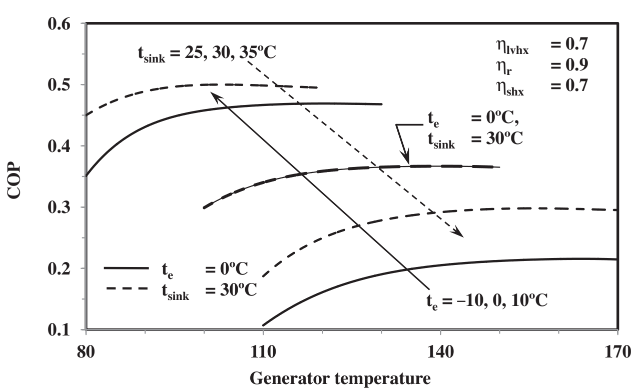

Fig. 4. Variation of COP with respect to generator temperature for various cooling and sink temperatures.

way apparent cooling loss per unit refrigeration capacity increases with a decrease in sink temperature at any fixed generator temperature.

Generator temperature is an important factor in determining system performance. Fig. 4 reveals that COP enhances with generator temperature initially, but decreases marginally after attaining a peak value. High generator temperature decreases the mass fraction of refrigerant of weak solution, which results in reduction of circulation ratio and mass flow rate of weak solution. Hence, according to Eq. (17) solution heat exchanger loss diminishes, leading to a requirement of less heat input. Simultaneously, with an increase in generator temperature, the amount of DMAC escaping to the condenser along with R134a vapour increases, which leads to an increase in the mass of residual liquid in the evaporator, resulting in more apparent cooling loss. At some intermediate temperature, these two losses attain an optimal balance. This intermediate temperature is referred to as optimum generator temperature hereafter in the discussion. As expected, the performance of the system deteriorates with a decrease in the cooling temperature and an increase in the sink temperature. Fig. 4 further indicates the rise of optimum generator temperature with a decrease in the cooling temperature. This can be justified by considering the requirement of higher generator temperatures at lower cooling temperatures. Similarly, the figure shows the increase of optimum generator temperature with sink temperature, because high sink temperature necessitates high generator temperature.

4.2. Effect of component efficiencies

Depending on the operating condition, some amount of DMAC boils along with R134a and the vapour from the generator gets enriched to some extent in the stripping sec-

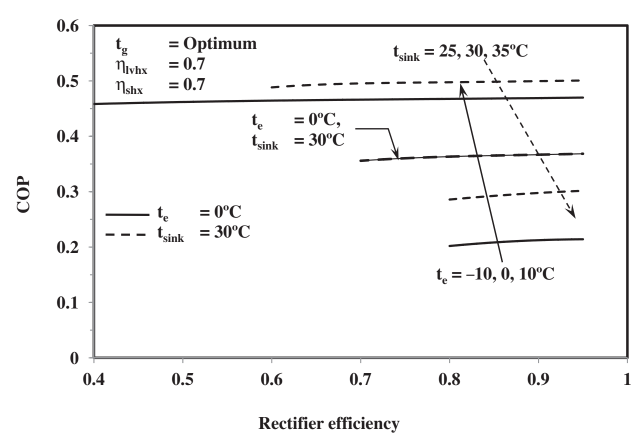

Fig. 5. Variation of COP with respect to rectifier efficiency for various cooling and sink temperatures.

tion by transferring heat to the incoming strong solution. In the rectification section of the rectifier, the vapour rejects some heat at the reflux condenser, partially condensing the vapour termed as reflux. The higher is the heat rejected at reflux condenser, the higher is the reflux and the mass fraction of R134a in vapour entering the condenser. Thus rectifier efficiency is more and wherein apparent cooling loss in the evaporator is less, due to higher concentration of R134a in the condensate. But it is achieved by releasing large amount of heat in the reflux condenser, which is to be supplied as additional heat input. Further, in LVHX this loss is recovered to some extent depending on LVHX efficiency. Hence, Fig. 5 shows that in the present operating conditions (LVHX efficiency = 0.7), system performance increases marginally with rectifier efficiency. It is interesting to note that each curve starts from a certain minimum rectifier efficiency, which in fact represent the scenario without the rectification section. Since enrichment of vapour also takes place in the stripping section, there exists a minimum efficiency as per Eq. (5) without the rectification section. According to Fig. 4, optimum generator temperature is high at low cooling temperature. Hence, a large amount of DMAC boils along with R134a. In addition, mass fraction of residual liquid increases with a decrease in cooling temperature, which enhances the mass of accumulated liquid and apparent cooling loss in the evaporator. The loss can be reduced, if mass fraction of R134a in the incoming condensate is increased. This is achieved with a high efficiency of the rectifier. Consequently, Fig. 5 indicates slightly higher significance of rectifier for low cooling temperature. Similarly, its importance increases slightly with sink temperature. Fig. 4 indicates that optimum generator temperature increases with sink temperature. So the amount of DMAC escaping along with R134a enhances. Hence, the apparent cooling loss in the evaporator increases. High efficiency of rectifier can compensate this to some extent.

Required optimum generator temperature enhances at low cooling and high sink temperature. As explained in Fig. 3, at high generator temperature larger amount of DMAC boils along with R134a, which results in an increase in the mass of the residual liquid and refrigerant entrained in it. Therefore, a large amount of R134a boils in LVHX, sub-cooling the condensate. So, the scope of heat transfer enhances correspondingly. Besides difference of inlet temperatures between the streams rises at low cooling and high sink temperature, as they cause decrease of cold stream and increase of hot stream inlet temperatures respectively. So, heat transfer in LVHX is more significant in these situations. Hence, Fig. 6 shows that system performance increases with LVHX efficiency at high rate at low cooling and high sink temperatures. In all operating conditions, LVHX enhances system COP to a significant extent, because of the low ratio of latent heat to specific heat of R134a vapour and liquid. At 0 °C, the ratios of latent heat to vapour and liquid for R134a are nearly one-sixth and one-fourth of corresponding values for water respectively.

In the present system, the role of rectifier in system performance is also influenced by the efficiency of LVHX. High rectifier efficiency reduces apparent cooling loss, but it is achieved at the cost of additional heat input to generator, whereas recovery of cooling enhances with the efficiency of LVHX without any increase in additional heat input to generator. Recovery of cooling decreases at high rectifier efficiency. Hence, Fig. 7 reveals that the role of the rectifier reduces gradually with an increase in LVHX efficiency. Its significance is almost nil when LVHX efficiency is as high as one. But when LVHX is not used, it enhances system COP from 0.29 to 0.32 only (around 10.3%) at mean operating condition.

The solution heat exchanger is very important in improving system COP owing to its large heat duty because of low latent heat of vapourisation of R134a. When cooling, sink and generator temperatures of the system are 0, 30 and 120 °C, respectively, the heat duty of the present system is nearly 3.5 kW per kW refrigeration capacity, whereas the corresponding value for H_2O–LiBr is about 0.5 kW per kW refrigeration. From Fig. 8, it is clear that the performance of the system enhances rapidly with an increase in SHX efficiency. It further shows that the importance of SHX increases at lower cooling temperatures because of higher values of circulation ratios, which result in enhancement of the amount of heat transfer in SHX. Similarly, its importance augments with the sink temperature. Owing to an increase in absorber temperature, the temperature difference between the two solutions decreases, hence the heat transfer tends to decrease, but this is more than counterbalanced by an enhancement in the circulation ratio. System COP increases from 0.123 to 0.432 if SHX efficiency is increased from 0.25 to 1 when cooling, sink and generator temperatures are −10, 30 and 150 °C, respectively.

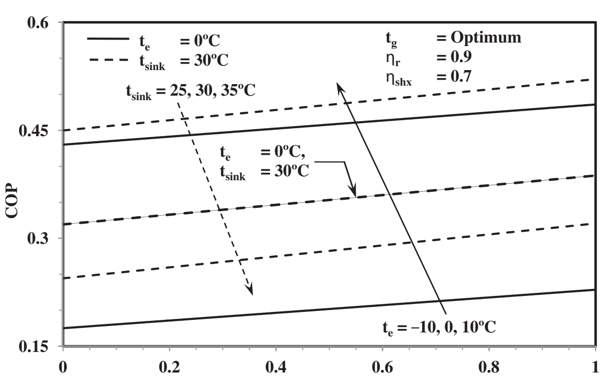

Fig. 6. Variation of COP with respect to LVHX efficiency for various cooling and sink temperatures.

4.3. Effect of evaporator pressure

Mass fraction of weak solution remains unchanged irrespective of evaporator pressure, as it is decided by genera-

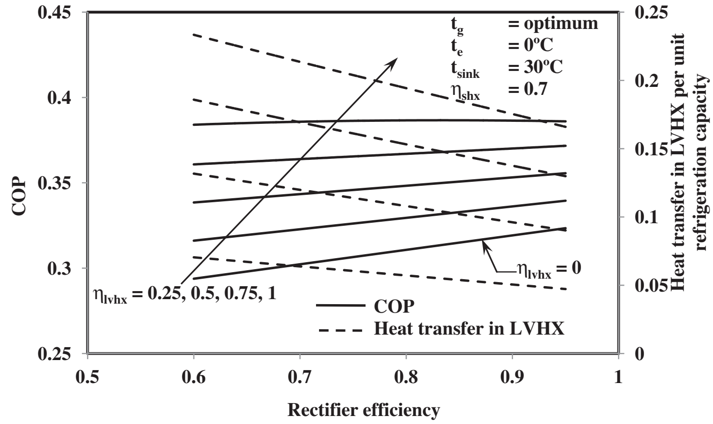

Fig. 7. Variation of COP with respect to rectifier efficiency for various LVHX efficiencies.

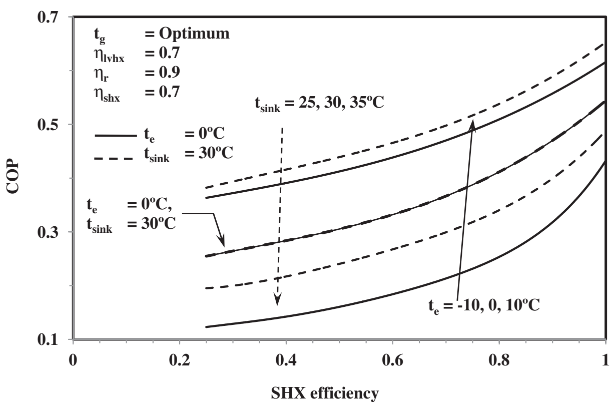

Fig. 8. Variation of COP with respect to SHX efficiency for various cooling and sink temperatures.

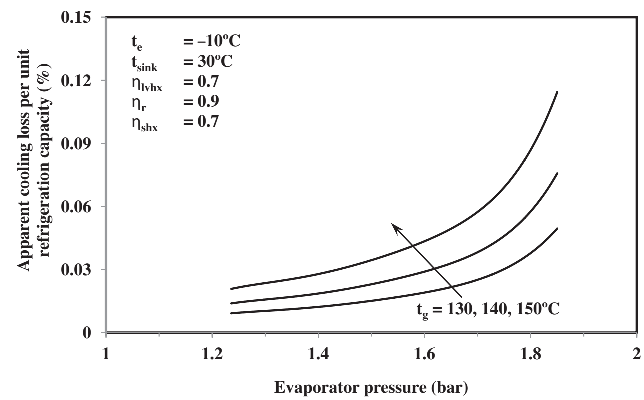

Fig. 10. Variation of apparent cooling loss with respect to evaporator pressure for various generator temperatures.

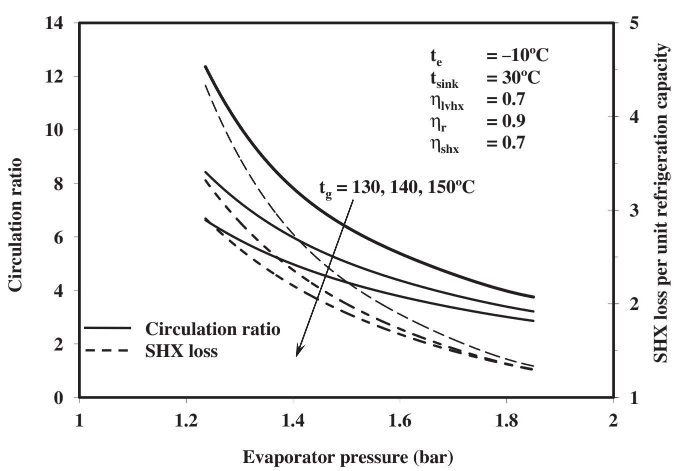

Fig. 9. Variation of circulation ratio and SHX loss with respect to evaporator pressure for various generator temperatures.

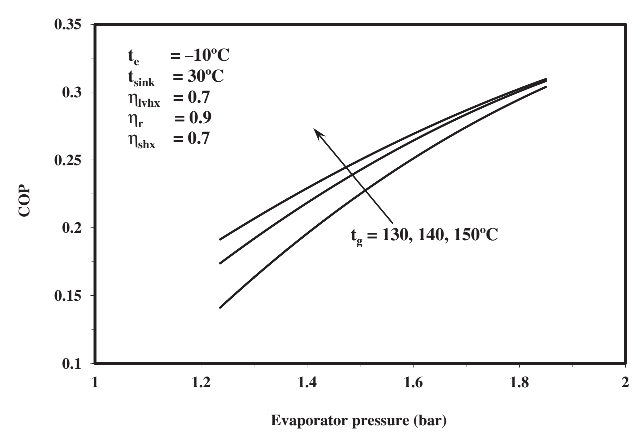

Fig. 11. Variation of COP with respect to evaporator pressure for various generator temperatures.

tor temperature and pressure only. However, that of strong solution increases with evaporator (low side) pressure. It results in an increase in the concentration difference between strong and weak solutions. Hence, Fig. 9 shows that circulation ratio decreases with an increase in evaporator pressure. Consequently, according to Eq. (17) reduction in SHX loss takes place.

Apart from the circulation ratio, apparent cooling loss is affected to some extent by evaporator pressure. Fig. 10 reveals that apparent cooling loss per unit heat input increases with an increase in evaporator pressure. Concentration of refrigerant in residual liquid reduces with reduction in the pressure. If cooling temperature t_4 is achieved at lower evaporator pressure p_{e2} instead of higher p_{e1}, the mass fraction of refrigerant of residual liquid reduces to ξ_{rl2} from ξ_{rl1} (Fig. 2). Mass and material balance equations show that, the mass of residual liquid reduces with a decrease in its mass fraction, because reduced value of $\xi_{rl'}$ increases the denominator term in Eq. (24). As a result, a small amount of R134a is entrained in residual liquid owing to a decrease in both of its amount and mass fraction. Further, the rate of increase of apparent cooling loss with evaporator pressure is more at higher generator temperatures.

Performance of the system depends on the simultaneous effects of evaporator pressure on cooling and SHX losses. Fig. 11 indicates that COP increases with an increase in evaporator pressure. Here the apparent cooling loss increases slightly, but the saving in the SHX more than compensates this loss. The circulation ratio decreases significantly leading to less SHX loss and consequently needing less heat input. So, with an increase in evaporator pressure COP of the system increases and rate of increase is faster when generator temperatures are lower. This phenomenon is attributed, to the facts that at low generator temperature, SHX loss increases more owing to high circulation ratio and saving in apparent cooling loss by means of low evaporator pressure becomes less significant owing to less amount of DMAC in the condensate.

Evaporator (low side) pressure is an important factor in determining apparent cooling loss. With an increase of rectifier efficiency, mass fraction of DMAC in the incoming condensate to evaporator reduces, so the apparent cooling loss reduces. Hence low evaporator pressure loses its importance for diminishing apparent cooling loss. However, irrespective of rectifier efficiency, circulation ratio and solution heat exchanger loss decrease with an increase in evaporator pressure. Consequently, owing to an increase of evaporator pressure system performance enhances at high rate, when rectifier efficiency is high. Fig. 12 reveals this aspect.

Significance of low evaporator pressure for reducing apparent cooling loss diminishes with an increase in LVHX efficiency. This phenomenon can be explained by considering large amount of recovery of apparent cooling loss at high LVHX efficiency. As discussed in the previous section, the negative effect of low evaporator pressure prevails irrespective of LVHX efficiency. Hence, Fig. 13 shows that, the rate of increase of system performance with evaporator pressure is more when LVHX efficiency is high.

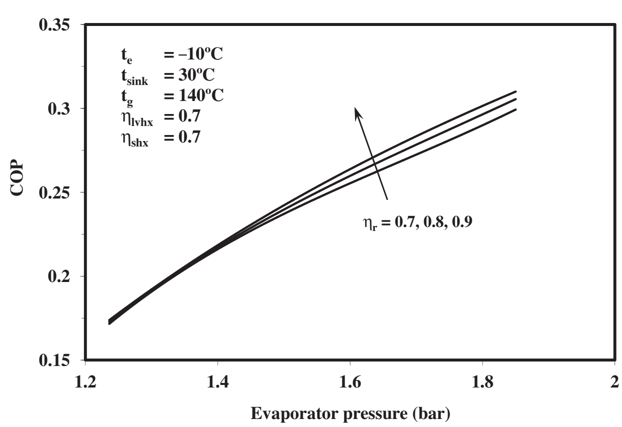

Fig. 12. Variation of COP with respect to evaporator pressure for various rectifier efficiencies.

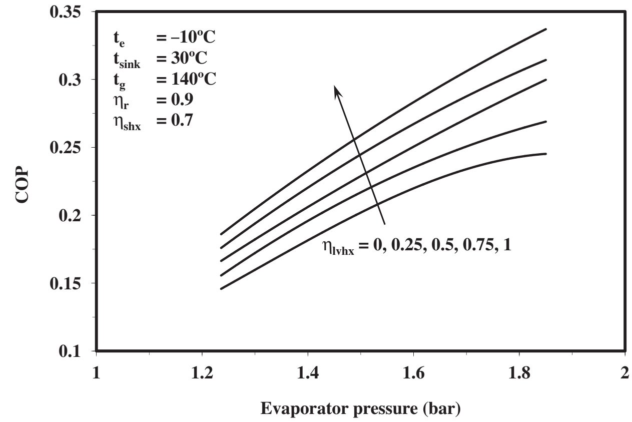

Fig. 13. Variation of COP with respect to evaporator pressure for various LVHX efficiencies.

5. Conclusions

Performance of R134a–DMAC VARS with rectifier and LVHX and at variable evaporator pressures for the same cooling temperature has been evaluated by theoretical studies at different operating conditions and component efficiencies. System performance peaks at some optimum generator temperature which increases at low cooling and high sink temperatures. Owing to temperature gradient in the evaporator, the same cooling temperature can be achieved at a range of evaporator pressures. For any specific cooling temperature, system COP increases with evaporator pressure. Rate of this increase is more when efficiencies of rectifier and LVHX are high. LVHX has a great role in enhancement of system COP, because of the low ratio of latent heat to specific heat of R134a vapour and liquid. It is found that, it increases the system COP up to 1.3 times in the range of operating conditions studied here. The significance of the rectifier is less in this respect and it looses its importance at high efficiency of LVHX. In the present operating conditions, maximum enhancement of COP by rectifier is around 14% (from 0.157 to 0.179) when LVHX is not used. For increasing system COP, SHX has the highest role among these three components. Significance of all the three components enhances at low cooling and high sink temperatures.

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