A Theoretical Study on the Overall Performance of a Falling Film Vertical Tube Absorber

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ABSTRACT

This work is an attempt to study theoretically the performance of a falling film multi vertical tube absorber. The governing heat and mass transfer equations were solved numerically using finite difference. The mass, cooling effectiveness, contact-space area ratio and the overall absorber performance were defined. Results were presented in terms of Lewis number, heat of absorption and thermal capacity ratio. The mass effectiveness was found to increase with increasing the Lewis number and reduce with increasing heat of absorption. The tube radius was found to have an insignificantly small effect on the mass effectiveness. The contact-space area ratio and the overall performance have greatly increased with decreasing the tube radius. An increase in overall performance means for the same absorber size, the mass flow rate of vapor increases and so does the cooling capacity in the evaporator. However, the overall performance has almost no effect on the system heat ratio

Keywords: Absorption; Refrigeration; heat and mass transfer; falling film.

1. Introduction

Vapor absorption refrigeration systems, VAS, differs from the vapor compression refrigeration system, VCS, by the means of extracting the refrigerant vapor from the evaporator. In absorption refrigeration systems, the absorber-generator replaces the compressor in VCS (Bahena and Romero, 2014). A strong solution enters the absorber where it absorbs vapor coming from the evaporator, during which heat is released. The lower is the absorbent temperature, the higher the affinity of the absorbent to vapor. This makes cooling of the absorbent becomes important. Therefore the description of the process requires the heat and mass transfer effects simultaneously (Sultana et al. 2015; Karami and Farhanieh 2009). To maximize the refrigeration capacity in vapor absorption refrigeration systems, the rate of vapor extracted from the evaporator must be maximized. This is achieved by making the concentration change of the absorbent as it passes through the absorber as large as possible. To achieve this goal, the absorbent must be cooled and the contact areas between the vapor and absorbent and the coolant and absorbent must be as large as possible. Such requirement can be achieved by various absorber arrangements. One of the widely used arrangements is the falling film type (Ventas et al. 2012; Harikrishnan 2011). Generally, practical absorbers are constructed using a bank of horizontal tubes (Yoon et al. 2013; Auracher et al. 2008). Strong solution falls down due to gravity forming a film around the outer surface of the tubes where it absorbs vapor and release heat that must be transferred to the coolant that passes through the inside of tubes. The performance of a falling film over a vertical tube and flat surface in vapor absorption refrigeration systems is being used quite often by researchers (Tomforde and Luke 2012; Tsem et al. 2001). As mentioned above the larger the contact areas between the absorbent and vapor, the larger the concentration change across the absorber. With a vertical tube, the larger the diameter, the larger is the contact area. However, the space occupied by a large vertical tube, which reduces the number of vertical tubes per unit absorber space area and so may reduce the absorber performance. Ibrahim and Khalaf (2013) studied the performance of a falling film absorber using thin wires. Results showed that such system is unfeasible due to low capacity of the wire and high cooling requirement. The current research is an attempt to study theoretically the performance of a falling film multi vertical tube absorber.

2. Theory

The system geometry is shown in Fig. 1. The strong absorbent from the generator enters the absorber at “I” and return to the generator at “E”. Then coolant is shown in counter-flow, but the analysis applies for both counter-flow and parallel-flow. The analysis covers the fully developed smooth laminar flow in the absorbent. There is no general agreement among researchers at what Reynolds number the falling films appears to be smooth. Hirshburg and Florschuetz (1982) reported a smooth film at $Re < 40$ where as Hounkanlin and Dumargue (1986) observed smooth film at $Re <= 80$. Saber (1993) compared the wavy film to smooth one and showed that the enhancement of mass transfer at Reynolds number of 400 drops significantly along the film. But reducing Reynolds number below a critical value, the falling film breaks and causes dry spots on the tube surface. Kim and Lee (2003) conducted experimental work on falling film over horizontal tube absorber. The absorber consisted of six tubes arranged in one row. They reported a minimum Reynolds number of 75 for full wettability of surface. More research on the critical Reynolds number is required. However, in this current work the theoretical model will be solved assuming smooth film at $Re = 80$. Further assumptions are:

- The vapor is single phase
- One component of the absorbent is non-volatile

- The absorbents are Newtonian and physical properties are constant with respect to temperature and concentration.
- The rate of vapor absorbed is small compared to the absorbent flow rate. Therefore the absorbent flow rate is constant and therefore its film thickness and mean velocity are also constant.

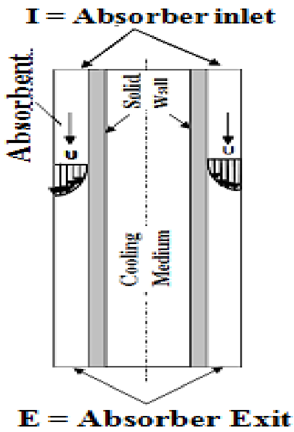

Fig. 1: is schematic diagram of falling film absorber along the outside of vertical tube of absorber.

Under the above mentioned conditions the governing equations in cylindrical coordinates are:

The velocity profile in the absorbent falling film is

$$u = \frac{g r_o^2}{2\upsilon}\left[(1+\varepsilon)^2 \ln\frac{r}{r_o} - 0.5\left(\left(\frac{r}{r_o}\right)^2 - 1\right)\right] \qquad \text{Eq. 1}$$

Where the mean velocity is:

$$\bar{u} = \frac{g r_o^3}{4\upsilon(1+0.5\varepsilon)\delta}\,\mathrm{E}_\varepsilon \qquad \text{Eq. 2}$$

Where

$$E_\varepsilon = \left\{(1+\varepsilon)^4\left[\ln(1+\varepsilon) - 0.75\right] + (1+\varepsilon)^2 - 0.25\right\}$$

The film thickness, δ, and the dimensionless film thickness, ε are derived in terms of Reynolds number, Re, and Galileo number, Ga, (Sinkunas et al 2005) as

$$\varepsilon = 1.67\left(\sqrt{1+1.09(\mathrm{Re}/Ga)^{1/3}} - 1\right) \qquad \text{Eq. 3a}$$

and the $\delta = \varepsilon\, r_o$ Eq. 3b

The diffusion and energy governing equations

$$u\frac{\partial C}{\partial x} = D\left[\frac{1}{r}\frac{\partial C}{\partial r} + \frac{\partial^2 C}{\partial r^2}\right] \qquad \text{Eq. 4}$$

$$u\frac{\partial T}{\partial x} = \alpha\left[\frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial r^2}\right] \qquad \text{Eq. 5}$$

2.1 Boundary Conditions

2.1.1 Absorbent-Refrigerant Interface

$$\frac{\partial T}{\partial r} = \frac{\dot{M}_v}{k}\delta h \qquad \text{Eq. 6}$$

Where

$$\delta h = h_v - \left[h + (1 - C)\frac{\partial h}{\partial C}\right] \qquad \text{Eq. 7}$$

$$\dot{M}_v = \frac{\rho D}{1 - C}\frac{\partial C}{\partial r} \qquad \text{Eq. 8}$$

The absorbent concentration and temperature are assumed to be in equilibrium at the interface at the operating vapor pressure.

$$T_i = f(C_i) \qquad \text{Eq. 9}$$

The actual equilibrium relationship between the absorbent concentration and temperature within the practical application is approximately linear (Ibrahim and Vinnicombe 1993)

$$T_i = a + bC_i \qquad \text{Eq. 10}$$

2.1.2 Absorbent-Coolant

$$\frac{\partial C}{\partial r} = 0 \qquad \text{Eq. 11}$$

$$\frac{\partial T}{\partial r} = \pm\frac{(\dot{m}Cp)_c}{k\,dA}dT_c \qquad \text{Eq. 12a}$$

$$\frac{\partial T}{\partial r} = \pm\frac{h_t}{k}(T_w - T_c) \qquad \text{Eq. 12b}$$

Due to the equilibrium state at the vapor-absorbent conditions, a sudden change in absorbent conditions at absorber inlet is most likely to occur and a point of singularity will therefore exist and precaution should be taken to avoid problem with solution.

The governing equations can be written in the following dimensionless form
By using the dimensionless parameters the governing equations will be:

$$X = \frac{x\alpha}{\bar{u}\delta^2}; \qquad \gamma = \frac{C - C_{in}}{C_e - C_{in}}; \qquad \theta = \frac{T - T_{in}}{T_e - T_{in}};$$

$$R = \frac{r - r_o}{\delta}; \qquad L_e = \frac{D}{\alpha}; \qquad U = \frac{u}{\bar{u}}$$

Where Tin and Cin are the absorbent conditions at the inlet, Ce is the equilibrium concentration at Tin and Te is the equilibrium temperature at Cin, and by analyzing the X components it may look as follow;

$$X = (k\,\Delta T/\delta)(x/(Cp\,\Gamma\,\Delta T)).$$

Therefore X may represent the heat transfer as a result of conduction relative to thermal capacity across the film along the x-axis. The governing equations can be written in the following dimensionless form

$$U = \frac{(2+\varepsilon)\varepsilon}{E_{\varepsilon}}\left[(1+\varepsilon)^2 \ln\frac{r}{r_o} - 0.5\left(\left(\frac{r}{r_o}\right)^2 - 1\right)\right] \qquad \text{Eq.13}$$

$$U\frac{\partial\gamma}{\partial X} = L_e\left[\frac{1}{R + 1/\varepsilon}\frac{\partial\gamma}{\partial R} + \frac{\partial^2\gamma}{\partial R^2}\right] \qquad \text{Eq. 14}$$

$$U\frac{\partial\theta}{\partial X} = \frac{1}{R + 1/\varepsilon}\frac{\partial\theta}{\partial R} + \frac{\partial^2\theta}{\partial R^2} \qquad \text{Eq. 15}$$

2.2 Boundary Conditions

2.2.1 Absorbent-Refrigerant Interface (R = 1)

$$\frac{\partial\theta}{\partial R} = H\frac{\partial\gamma}{\partial R} \qquad \text{Eq. 16}$$

Where

$$H = \frac{\rho D}{(1 - C_{in})k}\frac{C_e - C_{in}}{T_e - T_{in}}\delta h \qquad \text{Eq. 17}$$

Where interface equilibrium is governed by

$$\theta + \gamma = 1 \qquad \text{Eq. 18}$$

2.2.2 Coolant absorbent interface (R = 0)

$$\frac{\partial \gamma}{\partial \mathrm{R}} = 0 \qquad \text{Eq. 19}$$

$$\frac{\partial \theta}{\partial \mathrm{R}} = \pm \ \mathrm{Cr} \frac{\mathrm{d}\theta_{\mathrm{c}}}{\mathrm{dX}} \qquad \text{Eq. 20}$$

$$\frac{\partial \theta}{\partial \mathrm{R}} = \pm Nu_c \left(\theta_w - \theta_c\right) \qquad \text{Eq. 21}$$

Positive sign is for concurrent flow and negative sign is for counter flow.

3. Solution:

Explicit finite difference method-Matlab codes (Self written program) have been used to solve the above governing equations. However, it was found that because of the singularity within vapor-absorbent interface at entry and small Lewis number, the finite difference method is mesh size dependent and requires a distance to converge to the true solution. The numerical domain is divided into N and M nodes in the axial and transverse coordinates. The grid size ΔX is related to the grid size ΔR by the stability criteria for the explicit finite difference method and calculated for the selected M-nodes. The grid size was investigated by running the program with M = 80, 120, 160 and 300. It was found that, the discrepancy of average concentration produced with M = 80 and M = 300, for LitBr (Le = 0.0083 and H = 0.0934), was less than 1%, for Le = 0.001 was less than 3.2% and for Le = 0.0001, was less than 35%. Therefore, M = 80 was chosen to perform all calculation except for Le = 0.0001, M = 120 was applied to ensure accuracy. Ibrahim and Vinnicombe (1993) showed that the linear relation between concentration and temperature at the vapor-absorbent interface deviates from the nonlinear relation by about 0.8% at X = 10 and 3.4% at X = 1000. In the present study, the analysis is applied to a dimensionless length, X up 10.

A vertical absorber is mainly used for research purposes. In such absorber, the absorbent forms a falling film either on the inside of the tube and the cooling medium flows on the outside (Tsem et al 2001), or on the outside surface of the tube and cooling medium flows through the inside (Saber 1993). The size of the tube with the absorbent flowing on the inside should be large enough to accommodate the water vapor extracted from the evaporator with minimum pressure drop. For the case where the absorbent flows on the outside, the size of a vertical tube is also important in terms of contact area between vapor and absorbent, and space occupied. Although large tubes provide large contact (preferred), it occupies larger space (un-preferred). That raises the question, whether there is an optimum tube size that could balance between the contact area and absorber space. It may be more useful to implement the contact to space area ratio as

$$A_r = \frac{2\pi r_o x}{\pi r_o^2} = \frac{2x}{r_o}$$

The Ar is plotted versus the tube radius in Fig. (2) of 1 meter absorber length. From the figure, it can be seen that the smaller the tube radius the larger the contact-space area ratio. This implies that an absorber with smaller tube radius will be able absorb more water vapor. However, the need for simultaneous cooling, it puts a limit to how small a tube size can be. In the present work, the effect of tube radius and the Ar on the overall absorber performance will be investigated.

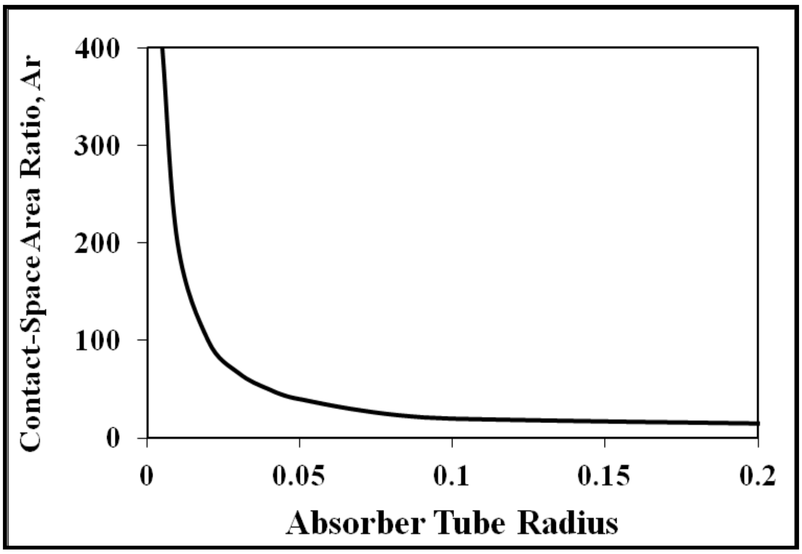

Fig. 2: The contact-Space area ratio, Ar, versus the tube radius.

4. Results and Discussion:

In addition to contact-space area ratio, the cooling capacity ratio, Cr is another convenient parameter for the presentation of results. The cooling capacity ratio, Cr is

$$Cr = (\dot{m}Cp)_c / (\dot{m}Cp)_a \qquad \text{Eq. 22}$$

The concept of effectiveness to describe the performance of heat exchanger is quite useful and so it has been adopted. Saber 1993 and Ibrahim (1991) used three different effectiveness; concentration effectiveness, heat effectiveness and mass effectiveness. The mass effectiveness seems to be more appropriate and descriptive of an absorber, since it relates the actual vapor mass flow rates absorbed to the maximum vapor mass flow rate absorbed. Since the absorber is neither a heat exchanger nor a condenser, but somewhere in between, the heat effectiveness was based on the coolant although the thermal capacity of absorbent is smaller than the coolant thermal capacity

.

$$(\dot{m}Cp)_a < (\dot{m}Cp)_c$$

So a practical absorber will have a Cr < 1, so the heat effectiveness was then defined as

$$\varepsilon_h = \frac{T_{c,o} - T_{c,in}}{T_{a,in} - T_{c,in}} \qquad \text{Eq. 23}$$

Since the objective of an absorber is to enhance the absorption rate by removing the heat of absorption and reducing the absorbent temperature, the cooling effectiveness is defined here as the ratio of the total energy (Latent and sensible) to be removed from the absorbent in the actual absorber to the maximum total energy that can be removed in an ideal absorber.

$$\varepsilon_c = \frac{(\dot{m}Cp)_a (T_{a,in} - T_{a,o}) + \dot{m}_v \delta h}{(\dot{m}Cp)_a (T_{a,in} - T_{c,in}) + \dot{m}_{v,\max} \delta h} \qquad \text{Eq. 24}$$

And the mass effectiveness is defined as the actual vapor flow rate absorbed divided by that maximum vapor flow arte obtained by an ideal absorber where the exit concentration is at equilibrium at cooling medium temperature.

$$\varepsilon_m = \frac{\dot{m}_v}{\dot{m}_{v,\max}} \qquad \text{Eq. 25}$$

Using the dimensionless parameters, the contact-space area ratio is related to the Reynolds Number and Prandtl number in the following form:

$$A_r = \frac{2x}{r_o} = 2X(\mathrm{Re}/\,\mathrm{Pr})\varepsilon \qquad \text{Eq. 26}$$

The heat effectiveness based on the coolant thermal capacity becomes

$$\varepsilon_h = \frac{\theta_{c,in} - \theta_{c,o}}{\theta_{c,in}} \qquad \text{Eq. 27}$$

The cooling effectiveness based on the absorbent is then

$$\varepsilon_c = \frac{H_f / \lambda - \theta_{a,o}}{H_f / \lambda_{\max} - \theta_{c,in}} \qquad \text{Eq. 28}$$

Where

$$H_f = (H / Le)\lambda_{\mathrm{r}}$$

Where is λ and λmax are the circulation factors of an actual absorber and ideal absorber where the concentration at the ideal absorber exit is at equilibrium with inlet coolant temperature respectively.

$$\lambda_r = (1 - C_{in})/(C_e - C_{in})$$

$$\lambda = \left(\lambda_r / \gamma_o\right) - 1$$
$$\lambda_{\max} = \left[\lambda_r /(1-\theta_{c,in})\right] - 1$$

The mass effectiveness may be expressed in terms of circulation factor as

$$\varepsilon_m = \frac{\dot{m}_v}{\dot{m}_{v,\max}} = \frac{\dot{m}_{a,in} / \lambda}{\dot{m}_{a,in} / \lambda_{\max}} = \frac{\lambda_{\max}}{\lambda} \qquad \text{Eq. 29}$$

From the design perspective, for a given heat effectiveness value, the actual exit cooling medium can be obtained easily. On the contrary the cooling effectiveness contains two unknowns, the actual absorbent exit temperature and exit concentration. Therefore, a second design parameter must be specified prior to use of the cooling effectiveness. The second parameter can be the circulation factor, which provides the exit concentration. However, the usefulness of the cooling effectiveness that it provides information about the absorbent exit conditions, rather than the cooling medium conditions.

In Practice, a counter-flow sensible heat transfer is always more effective than a parallel-flow heat transfer and therefore it is preferred, whereas in an absorber the heat transfer is due to both latent and sensible, which suppose to reduce the effect of flow direction on the cooling process. However, all results in this work are produced with counter-flow. Fig. 3 shows that for an absorber dimensionless length of 10 the absorbent wall temperatures at thermal capacity ratio, Cr, of 20, 40, 100 with inlet cooling temperature of -1 falls along the absorber linearly keeping constant temperature difference between the absorbent and cooling medium. Near the entry region, up to X = 0.1, the effect of heat of absorption on has not yet penetrated to the wall. Therefore the absorbent wall temperature is affected only by the cooling medium..

In Fig.4 the mass effectiveness is plotted against the absorber length at different Lewis numbers. It is obvious that to maximize the effectiveness, the absorbent should have a Lewis number as large as possible. This can also be seen from Fig. 5 where the effect of Lewis number on the concentration change along the absorber is quite obvious. The heat of absorption is assumed to be H = 0.01 except for LitBr-water absorbent the H = 0.0934. In absorption refrigeration systems, an absorbent with a large Lewis number is favored because it produces large mass effectiveness and concentration changes. The larger the Lewis number, the larger the absorbent concentration rate of change. Since the Lewis number for LitBr-Water (Le = 0.0083) is within the range 0.001 and 0.01, its average concentration falls between the average concentration for Le = 0.01 and Le = 0.001. But at the interface where the H is quite influential, the interface concentration of LitBr-Water is lowest due to its high heat of absorption (H = 0.0934) compared to the other working fluids (with H = 0.01 for $0.0001 \leq Le \leq 0.01$).

The effect of heat of absorption at Le = 0.01 on the mass effectiveness is shown in Fig. (6). It can be clearly seen that the smaller the heat of absorption, the lager the mass effectiveness. For H = 0.001, the mass effectiveness is the largest (at X = 10, it is greater than the mass effectiveness for H = 0.01 by 0.7% and for H = 0.1 by 8.5%). The mass effectiveness of LitBr-Water is the lowest (at X = 10, it is less than the mass effectiveness for H = 0.1 by 2.22%), although its heat of absorption (H = 0.0934) is not the highest. This is because, its Lewis number (Le = 0.0083) is smaller than that of the absorbent with H = 0.1 and Le = 0.01.

Since the LitBr-water pair is most used in the air-conditioning application, the heat, cooling and mass effectiveness for this pair will be presented to see how they are affected by the Cr. The heat and cooling effectiveness are plotted against the Cr.

From Fig. (7), it can be seen that the heat effectiveness, εh, decreases with increasing the thermal capacity ratio, Cr. This is expected since the coolant temperature slope decreases with increasing the Cr (Fig. 3), i.e. the change in cooling temperature (Tco – Tcin) in equation 23 decreases with increasing Cr. On the contrary, the cooling effectiveness increases only marginally with Cr. Fig.8 shows the increase in mass effectiveness with increasing Cr. However, the rate of increase of Em with Cr is diminishing relatively fast at Cr > 40. For example, increasing Cr from 20 to 40, the mass effectiveness increases by 0.5%. But by increasing Cr from 40 to 60, the mass effectiveness increases only by 0.186%. The effect of tube radius on the mass effectiveness is insignificantly small. For example, the increase in mass effectiveness by reducing the radius from 100 mm to 5 mm is maximum 0.07% at Cr = 100. This increase is due to the slight reduction in film thickness, where δ = 0.4135 mm at ro = 5 mm and 0.42mm at ro = 100 mm (equation 3b). This in turn will have a minor effect on circulation factor. Consequently, the heat ratio of the absorption refrigeration system almost remains the same. In other words, the effect of tube size should lie in the capacity of the absorber with respect to its size. To see the effect of the tube size on multi tube absorber, the overall absorber performance, Eff, may be defined as the actual vapor flow rate absorbed by a multi-tube absorber that occupies one unit space-area divided by the maximum vapor flow arte absorbed by an ideal one-tube absorber that occupies the same one unit space-area with the exit concentration at equilibrium with the inlet cooling medium temperature. This will take into account the total contact area of all absorber tubes that occupy one unit space area. i.e. the number of tubes forming the absorber of one unit space area decreases with increasing the tube radius. The overall absorber performance, Eff, is

$$E_{ff} = \frac{\dot{m}_{v,t}}{\dot{m}_{v,smx}} \qquad \text{Eq. 30}$$

Where $\dot{m}_{v,t}$ is the actual total vapor mass flow rate and $\dot{m}_{v,smx}$ is the vapor flow rate that is obtained from an absorber of a single tube that occupies one unite space area where the absorbent leaves the absorber with an equilibrium concentration at the inlet coolant temperature. With $\dot{m}_{v,t} = 2\pi r_o \Gamma No / \lambda$, $\dot{m}_{v,smx} = 2\pi r_{o,s} \Gamma No / \lambda_{\max}$ and the number of tubes is No = π ro,s 2 /π ro2 , the overall performance becomes

$$E_{ff} = \frac{r_{o,s}\lambda_{\max}}{r_o \lambda} = \frac{r_{o,s}}{r_o}\varepsilon_m \qquad \text{Eq. 31}$$

The absorber overall performance (Fig.9), unlike the mass effectiveness, increases largely with decreasing the tube radius. It also increases with increasing the thermal capacity ratio, Cr. Eff increases linearly with increasing the contact-space area ratio, (Fig. 10). However, at very low Ar, the absorber overall performance have almost the same value at any thermal capacity ratio, Cr. But with increasing contact space area ratio, the lines of Eff begins to separate having a larger slope at large Cr. On the other hand, it should be well understood that, reducing the tube size in the multi vertical tube leads to an increase in the overall performance, Eff, i.e. an increase in the mass flow rate of vapor absorbed in the absorber, and so does the cooling capacity in the evaporator. Consequently, larger absorbent mass flow rate needed in the absorber, which has to be accommodated in the generator. To illustrate the meaning of the absorber overall performance, considers two absorbers with one unit space area; using Fig. 9, the first one, which is a multi-tube absorber with a radius of 5 mm, has an overall performance of approximately Eff = 19 whereas the second one, which is a multi-tube absorber with a radius of 20 mm, has an overall performance of approximately Eff = 5. Therefore the absorber of a smaller tube size will absorb about 3.8 times more water vapor than the absorber of the larger tube size does.

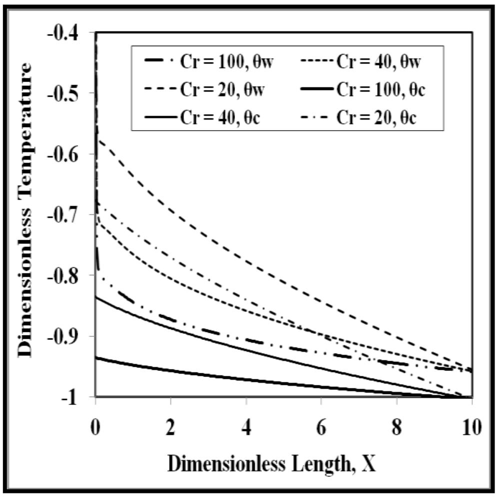

Fig. 3: shows the behavior of the absorbent wall temperature along the tube length at various thermal capacity ratio. θc,in = -1, Re = 80, ro = 5 mm, LitBr-Water (H = 0.0934 and Le = 0.0083).

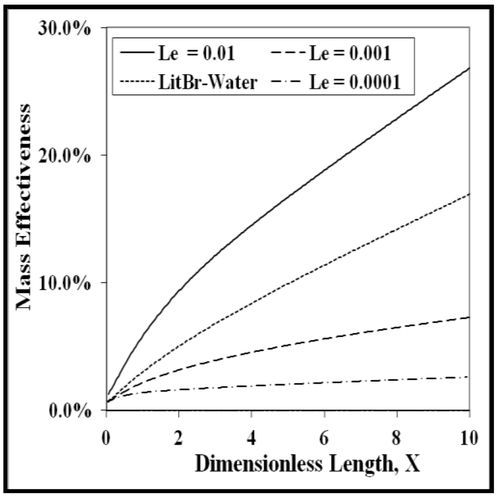

Fig.4 the mass effectiveness is plotted against the absorber length at different Lewis numbers. H = 0.01, θc,in = -1, Re = 80, Cr = 100, ro = 5 mm, LitBr-Water (H = 0.0934 and Le = 0.0083)

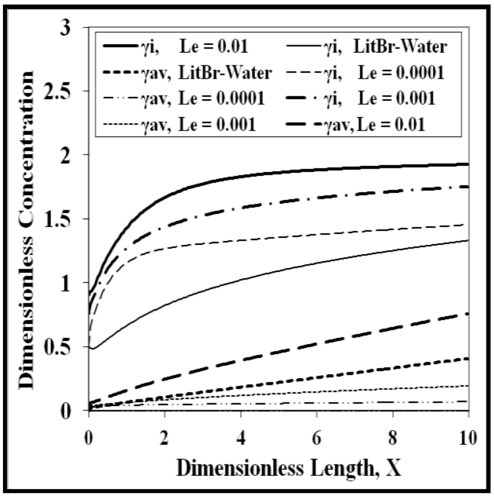

Fig. 5 shows the effect of Lewis number on the concentration change along the absorber. H = 0.01, θc,in = -1, Re = 80, Cr = 100, ro = 5 mm, LitBr-Water (H = 0.0934 and Le = 0.0083),i: vapor-absorbent interface: ave: average values.

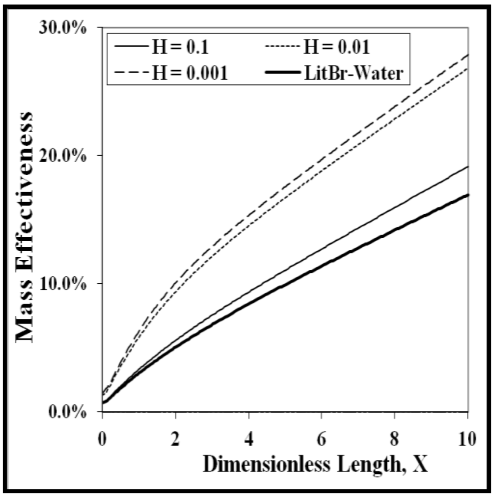

Fig. 6 presents the effect of heat of absorption on the mass effectivness. Le = 0.01, θc,in = -1, Re = 80, Cr = 100, ro = 5 mm, LitBr-Water (H = 0.0934 and Le =

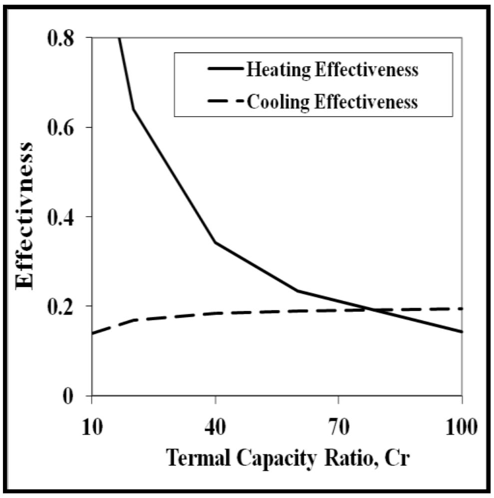

Fig. 7 the heat and cooling effectiveness are plotted versus thermal capacity ratio. θc,in = -1, Re = 80, ro = 5 mm, LitBr-Water (H = 0.0934 and Le = 0.0083).

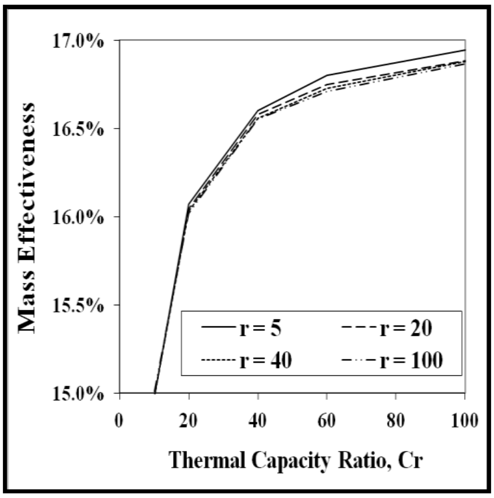

Fig.8 Effect of thermal capacity ratio and radius on the mass effectiveness. θc,in = -1, Re = 80, LitBr-Water(H = 0.0934 and Le = 0.0083)

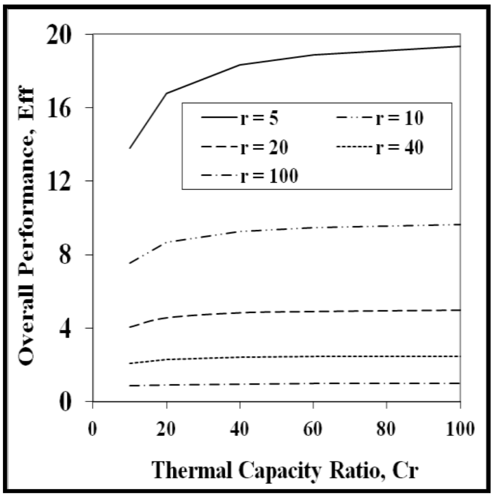

Fig. 9 The effect of tube radius and thermal capacity ratio on the absorber overall performance. θc,in = -1, Re = 80, LitBr-Water (H = 0.0934 and Le = 0.0083).

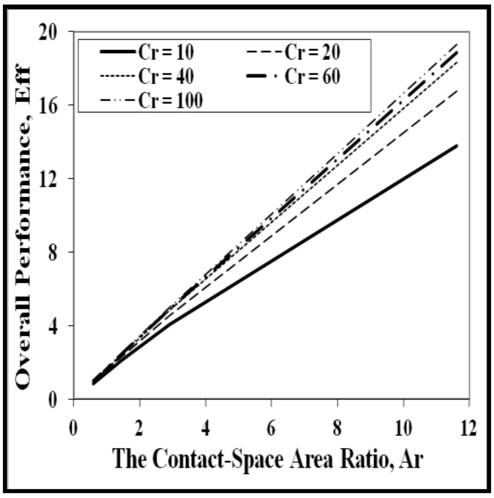

Fig. 10 the overall performance as function of contact-space area ratio at various thermal capacity ratios. θc,in = -1, Re = 80, LitBr-Water (H = 0.0934 and Le = 0.0083).

. Conclusion

Results were presented in terms of Lewis number, heat of absorption and thermal capacity ratio. The important results are concluded as follows:

- The heat effectiveness decreases with increasing thermal capacity ratio, due to the reduction in the change of cooling temperature between absorber inlet and outlet, (Tco – Tcin).
- The mass effectiveness increases with increasing Lewis number and reduces with increasing heat of absorption.
- The effect of tube radius on the mass effectiveness is insignificantly small (maximum 7% at Cr = 100).
- The contact-space area ratio has greatly increased with decreasing the tube radius.
- The overall performance of a multi-vertical tube absorber increases with decreasing tube radius. i.e. for the same absorber size, the mass flow rate of vapor absorbed increases and so does the cooling capacity in the evaporator. However, the overall performance has almost no effect on the system heat ratio.

Nomenclature

Ar	Contact-Space are ratio
a	Constant in equation (10)
b	Constant in equation (10)
C	Concentration
CP	Specific heat[J kg^{-1} K^{-1}]
Cr	Heat capacity ratio (mCp)c/(mCp)a
D	Diffusion coefficient [m^2 s^{-1}]
Eff	Absorber overall performance
Ga	Galileo number ($g\, r_o^3/\nu^2$)
g	Gravitational constant [ms^{-2}]
h	Enthalpy [J kg^{-1}]
ht	Heat transfer coefficient [W m^{-2} K^{-1}]
δh	Heat of absorption [Jkg^{-1}]
H	Dimensionless heat of absorption
Hf	Heat factor (H/Le)λr
k	Absorbent thermal conductivity[Wm-1K^{-1}]
Le	Lewis number
$\dot{m}$	mass flow rate [kg s^{-1}]
$\dot{M}_v$	vapor mass flux [kg $m^{-2}$$s^{-1}$]
$\dot{m}_v$	vapor mass flow rate [kg s^{-1}]
No	Number of tubes
Nuc	Nusselt Number ($h_t\delta$)/k
Pr	Prandtl number
Re	Reynolds number (4Γ/ν)

r	Radial coordinate [m]
ro	Tube radius [m]
R	Dimensionless radial coordinate
T	Temperature [^oC]
u	Velocity [ms^{-1}]
$\bar{u}$	Mean velocity [$m\,s^{-1}$]
U	Dimensionless velocity
x	coordinate in flow direction [m]
X	Dimensionless coordinate in flow direction

Greek symbols

Γ	Mass flow rate per wetted perimeter [$kgm^{-1}s^{-1}$]
α	Thermal diffusivity (k/ρ Cp)
γ	Dimensionless concentration
δ	Film thickness [m]
ε	Film thickness ratio (δ/ro)
ε_h	Heat effectiveness
ε_c	Cooling effectiveness
ε_m	Mass effectiveness
λr	Concentration factor (1 - Cin)/(Ce - Cin)
λ	Circulation factor (λr/γo – 1)
θ	Dimensionless temperature
ρ	Density [$kg\ m^{-3}$]
ν	Kinematic viscosity [$m^2\ s^{-1}$]

Subscripts

a	absorbent
av	average

c	coolant
i	Absorbent-vapor interface
in	inlet
max	maximum state
s	single tube
smx	single tube maximum state
o	exit
t	total
v	vapor
w	absorbent at the wall

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