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Numerical Studies On The Performance Of Methanol Based Air To Air Heat Pipe Heat Exchanger

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Abstract: Heat transfer has an important role in many engineering applications. Recently heat pipe heat exchangers have been extensively used in industries for the application of waste heat recovery from exhaust gasses. This paper presents numerical simulation of an air to air Gravity Assisted Heat Pipe Heat Exchanger (GAHPHE) which uses methanol as a working fluid based on effectiveness – number of transfer units method. Ambient air temperature of 30ºC has been considered at condenser inlet. The effects of mass flow rate ($\dot{m}$) and hot air temperature at evaporator inlet ($T_{H,in}$) on heat transfer (Q), overall heat transfer coefficient (U), overall effectiveness (ε) and temperature range ($T_{H,in} – T_{H,out}$) for copper pipe diameters of 8 mm and 16 mm have been studied.

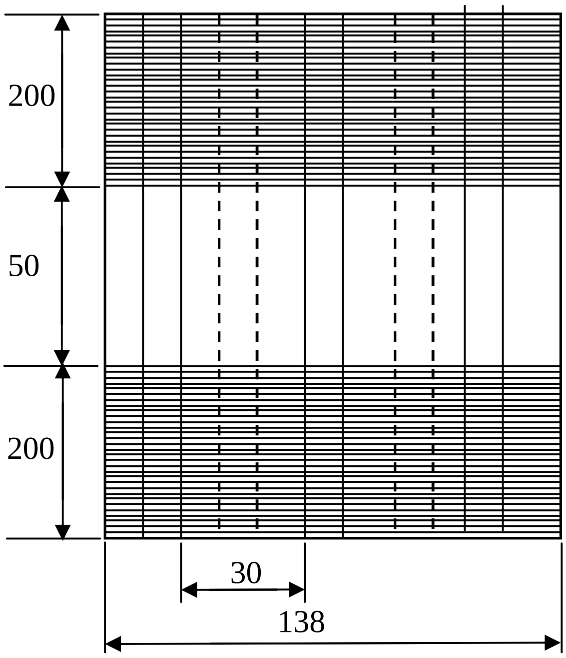
Keywords: heat pipe; thermosyphon heat exchanger; ε-NTU method; methanol.

Introduction

Heat pipes are heat transfer devices which use the principles of thermal conduction and latent heat of vaporisation to transfer heat effectively at very fast rates. There are two types of heat pipes depending on how the condensed working fluid is returned to the evaporator section, namely gravity assisted heat pipes and heat pipes with wicks. In gravity assisted heat pipes the condensed fluid is returned due to gravitational forces whereas in heat pipes with wicks the condensed fluid is returned due to the capillary action through wicks. The use of heat pipes in heat exchangers results in many advantages such as compactness, low weight, high heat recovery effectiveness, no moving parts, pressure tightness, complete separation of hot and cold fluids and high reliability[1]. These advantages led to usage of heat pipe heat exchangers in various industries as waste heat recovery systems[2] waste heat from the exhaust gasses released from the industrial plants. An important application is the recovery of waste heat from the industrial exhaust gasses[3] which not only saves energy but also protects the environment.

Azad and Gibbs presented a theoretical study of an air to water heat pipe heat exchanger[4] in which the variation of effectiveness and with ratio of cold to hot flow-stream capacity rate for 8, 10 and 12 rows was analysed. An experimental study of the performance of air to air heat pipe heat exchanger utilizing R-22 as the working fluid was carried out by Wadowski et al.[5] to investigate its behaviour under different operating conditions. An experimental and theoretical research was carried out by Noie to investigate the thermal performance of an air to air thermosyphon heat exchanger in which distilled water was used as the working fluid[6]. The overall effectiveness of the heat exchanger varied between 37% and 65%.

In the present study, a simulation program has been developed to predict the performance of an air to air gravity assisted heat pipe heat exchanger with eight heat pipes in a staggered arrangement as shown in Fig. 1. The working fluid considered is methanol and copper pipes are used. The total length of each heat pipe is taken as 450 mm. The length of the condenser section and evaporator section is each 200 mm and the length of the insulated section is taken as 50mm. The length and the breadth of the heat exchanger were considered to be 138mm each. The eight heat pipes were arranged in three rows with a surface to surface distance between rows being 30 mm. As heat is being transferred between hot air and cold air, both the evaporator and condenser section are assumed to have finned surfaces. Both the sections are assumed to have eighty aluminium fins each.



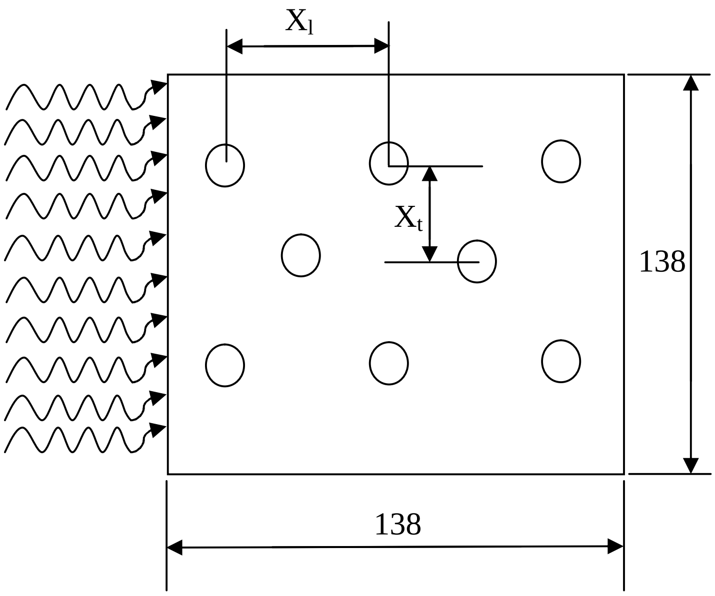


Fig. 1 Front and top view of GAHPHE

Mathematical Model

In the present study for the analysis of an air to air Gravity Assisted Heat Pipe Heat Exchanger (GAHPHE) the method of effectiveness of transfer units has been used[7]. The GAHPHE considered for this study consists of methanol heat pipes in staggered arrangement with continuous aluminium finned circular tubing are used. By neglecting heat conduction in the axial direction of heat pipes the equations given below were used for the simulation.

Evaporator side

The resistances for the outer, pipe and inner section of the evaporator side are given by Eq.1, Eq.2 and Eq.3 respectively.

$$R_o = \frac{1}{\eta_o\, h_o\, A_o} \tag{1}$$

$$R_p = \frac{t}{k_p A_i} \tag{2}$$

$$R_i = \frac{1}{h_i \ A_i} \quad (3)$$

The overall heat transfer coefficient of the evaporator side based on the outer area is given by Eq.4.

$$U = \frac{1}{A_o(R_o + R_p + R_i)} \quad (4)$$

The correlations for continuous fin with circular tubing [8] are given by Eq.5 and Eq.6.

$$Re_L = \frac{G \ X_l}{\mu} \quad (5)$$

$$j = 0.195 \ Re_L^{-0.35} = St \ Pr^{2/3} \quad (6)$$

The Stanton number is given by-

$$St = \frac{h_o}{c_p \ \rho \ u} \quad (7)$$

By solving Eq.5, Eq.6 and Eq.7 for h_o we can find the value of heat transfer coefficient of the outer section of the evaporator side.

The Air side surface efficiency is given by Eq. 8

$$\eta_o = 1 - \frac{A_{fin}}{A_o} (1 - \eta_{fin}) \quad (8)$$

The fin efficiency η_{fin} of a non-circular fin is given by Eq.9, where R_e is the effective radius of the fin

$$\eta_{fin} = \frac{\tanh(m_{es} R_e \emptyset)}{m_{es} R_e \emptyset} \quad (9)$$

The standard extended surface parameter, m_{es} is given by Eq. 10 by assuming that the thickness of the fin is much less than the length of the fin.

$$m_{es} = \sqrt{\frac{2h_o}{k \ t_f}} \quad (10)$$

The effective fin radius R_e is calculated using the correlations given in Eq. 11 to Eq.15

$$\frac{R_e}{r} = 1.27 \ \Psi \ (\beta - 0.3)^{1/2} \quad (11)$$

Where r is the outer radius of the tube and,

$$\Psi = \frac{B}{r} \quad (12)$$

$$\beta = \frac{H}{B} \quad (13)$$

In this analysis

$$B = X_l \ if \ X_l < X_t/2 \quad otherwise \ B = X_t/2 \quad (14)$$

$$H = \frac{1}{2}\sqrt{\left(\frac{X_t}{2}\right)^2 + X_l^2} \tag{15}$$

The fin efficiency parameter is found by using the effective radius in Eq.16

$$\emptyset = \left(\frac{R_e}{r} - 1\right)\left(1 + 0.35\, ln\left(\frac{R_e}{r}\right)\right) \tag{16}$$

A modified version of Rohsenow equation is used to determine the boiling heat transfer coefficient i.e. the heat transfer coefficient (h_i) inside the pipe in the evaporator section.

$$Nu = C_1 Re^{C_2} Pr^{C_3} \tag{17}$$

Where,

$$Nu = \frac{h_i}{k_l}\left(\frac{\sigma}{g(\rho_l - \rho_g)}\right)^{1/2} \tag{18}$$

$$Re = \frac{q}{\lambda \mu_l}\left(\frac{\sigma}{g(\rho_l - \rho_g)}\right)^{1/2} \tag{19}$$

For methanol as working fluid C_1= 24.09, C_2= 0.2 and C_3= 0.4. By using Eq. 17 to Eq.19 and substituting appropriate values the value of h_i is determined.

Condenser side

Eq.1 to Eq.16 are used to determine the heat transfer coefficient on the outside of the condenser section and the air side surface efficiency on the condenser side. The property values to be substituted are taken at the temperature on the condenser side.

For the condenser side the heat transfer coefficient inside the pipe $h_{i,}$ is found by using the equation given below.

$$h_i = C_4\left(\frac{\rho_l(\rho_l - \rho_g) g k_l^3 \lambda}{\mu_l L \Delta T_{cd}}\right)^{C_5} \tag{20}$$

For methanol C_4=0.96, C_5=0.27 and ΔT_{cd} is the temperature difference between inside surface temperature and inside temperature of the condenser.

Ɛ-NTU method

There are three methods available for predicting the performance of heat exchangers using heat pipes[9]. The method used in the present study is effectiveness-number of transfer units[7]. The effectiveness-NTU equations for a single row heat pipe heat exchanger are given as follows[10].

For evaporator section,

$$(NTU)_H = \frac{(UA_o)_H}{C_H} \tag{21}$$

$$C_H = (\dot{m}c_p)_H \tag{22}$$

$$\varepsilon_{H1} = 1 - \exp\,(-NTU)_H \tag{23}$$

For condenser section,

$$(NTU)_C = \frac{(UA_o)_C}{C_C} \tag{24}$$

$$C_C = (\dot{m}c_p)_C \tag{25}$$

$$\varepsilon_{C1} = 1 - \exp\,(-NTU)_C \tag{26}$$

For a heat pipe heat exchanger with n rows in the direction of flow, the effectiveness- NTU equations are given below

For evaporator section,

$$\varepsilon_{Hn} = \frac{\left(\frac{1-\frac{C_H}{C_L}\varepsilon_{H1}}{1-\varepsilon_{H1}}\right)^n - 1}{\left(\frac{1-\frac{C_H}{C_L}\varepsilon_{H1}}{1-\varepsilon_{H1}}\right)^n - \frac{C_H}{C_L}} \tag{27}$$

For condenser section,

$$\varepsilon_{Cn} = \frac{\left(\frac{1-\frac{C_C}{C_L}\varepsilon_{C1}}{1-\varepsilon_{H1}}\right)^n - 1}{\left(\frac{1-\frac{C_C}{C_L}\varepsilon_{C1}}{1-\varepsilon_{C1}}\right)^n - \frac{C_C}{C_L}} \tag{28}$$

Since vapour inside the heat pipe is at constant temperature the specific heat c_p and the capacity rate C_L become equal to infinity. So C_H/C_L and C_C/C_L become equal to zero. Therefore Eq.27 and Eq.28 reduce to the following equations.

$$\varepsilon_{Hn} = 1 - (1 - \varepsilon_{H1})^n \tag{29}$$
$$\varepsilon_{Cn} = 1 - (1 - \varepsilon_{C1})^n \tag{30}$$

The overall effectiveness of the heat exchanger is given as,

If $C_H > C_C$,

$$\varepsilon_T = \frac{1}{\left(\frac{1}{\varepsilon_{Cn}} + \frac{C_C/C_H}{\varepsilon_{Hn}}\right)} \tag{31}$$

If $C_C > C_H$

$$\varepsilon_T = \frac{1}{\left(\frac{1}{\varepsilon_{Hn}} + \frac{C_H/C_C}{\varepsilon_{Cn}}\right)} \tag{32}$$

The outlet temperatures of the air from the evaporator and the condenser side are given by the equations given below,

$$T_{H,out} = T_{H,in} - \varepsilon_T \frac{(\dot{m}c_p)_{min}}{(\dot{m}c_p)_H}\left(T_{H,in} - T_{C,in}\right) \tag{33}$$

$$T_{C,out} = T_{C,in} + \varepsilon_T \frac{(\dot{m}c_p)_{min}}{(\dot{m}c_p)_C}\left(T_{H,in} - T_{C,in}\right) \tag{34}$$

Results And Discussion

A computer simulation program has been written based on the above analysis and results obtained are discussed in this section. Methanol has been used as refrigerant and its thermodynamic properties have been calculated using REFPROP. The following range of input data is used for the analysis.

Air inlet temperature at evaporator	:	60 to 90ºC
Air inlet temperature at condenser	:	30ºC
Refrigerant saturation temperature	:	50ºC
Mass flow of air across evaporator	:	0.1 to 0.5 kgs^{-1}
Mass flow of air across condenser	:	0.1 to 0.5 kgs^{-1}
Diameter of the tube	:	8 and 16 mm

Effect of mass flow of air

Figure 2 shows the effect of mass flow of air at evaporator inlet on overall heat transfer coefficient (U) across the heat pipe heat exchanger for 8 and 16 mm tubes. As mass flow of air increases, air-side heat transfer coefficient increases resulting in increase in overall heat transfer coefficient.

Figure 3 shows the effect of mass flow of air on heat transfer rate across the heat pipe heat exchanger for 8 and 16 mm tubes. Heat transfer rate is a function of overall heat transfer coefficient. Hence, as mass flow of air increases, heat transfer rate also increases due to increase in overall heat transfer coefficient (Fig. 2)

Figure 4 shows the effect of mass flow of air on effectiveness of heat pipe heat exchanger for 8 and 16 mm tubes. Effectiveness is a function of number of transfer units (NTU), which is directly proportional to 'U' and inversely proportional to heat capacity rate of air (C_{air}). As mass flow of air increases, 'U' as well as 'C' increase. However, increase in 'C' is more than that of 'U', resulting in decrease in NTU. This leads to decrease in effectiveness.

Figure 5 shows the effect of mass flow of air on temperature range across heat pipe heat exchanger for 8 and 16 mm tubes. Temperature range is a function of heat pipe heat exchanger effectiveness. Hence as mass flow of air increases, temperature range decreases due to decrease in effectiveness (Fig. 4).

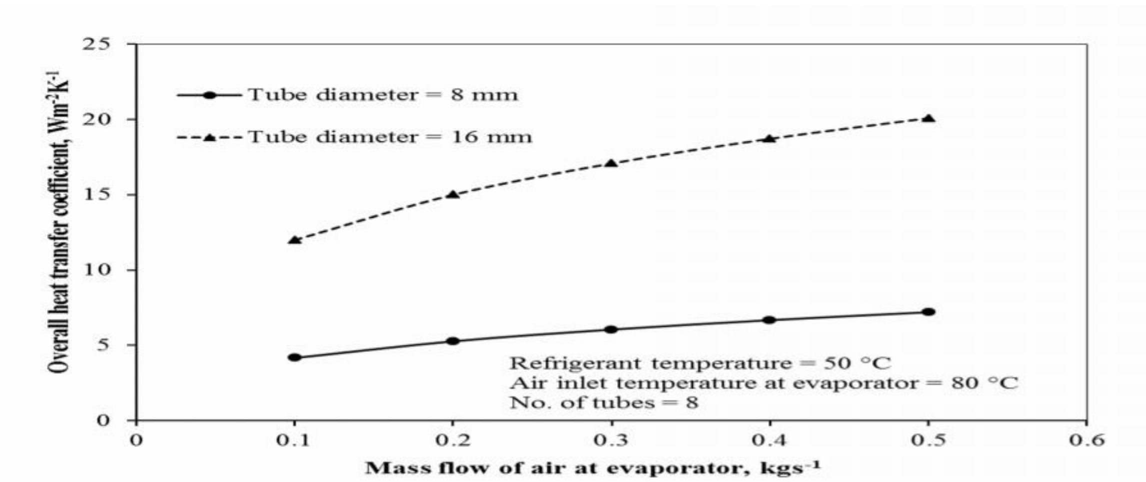


Fig.2 mass flow of air at evaporator vs. overall heat transfer coefficient

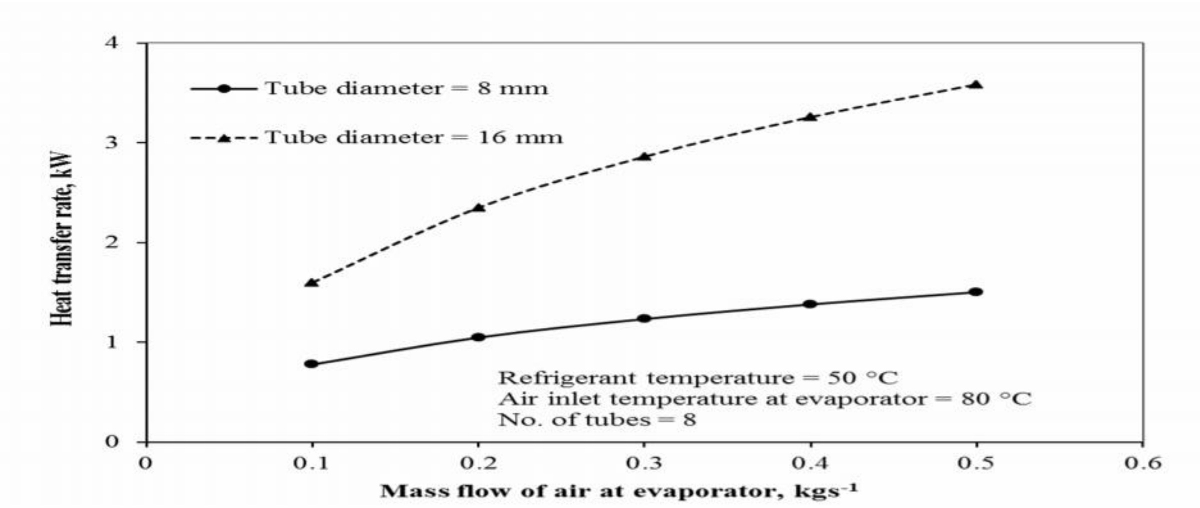


Fig.3 mass flow of air at evaporator vs. heat transfer rate

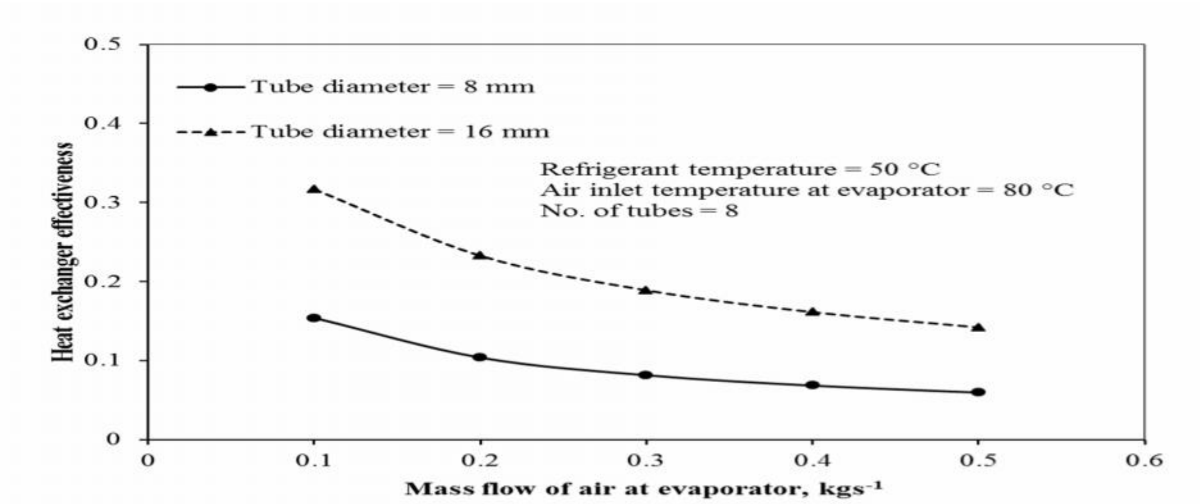


Fig.4 mass flow of air at evaporator vs. heat exchanger effectiveness

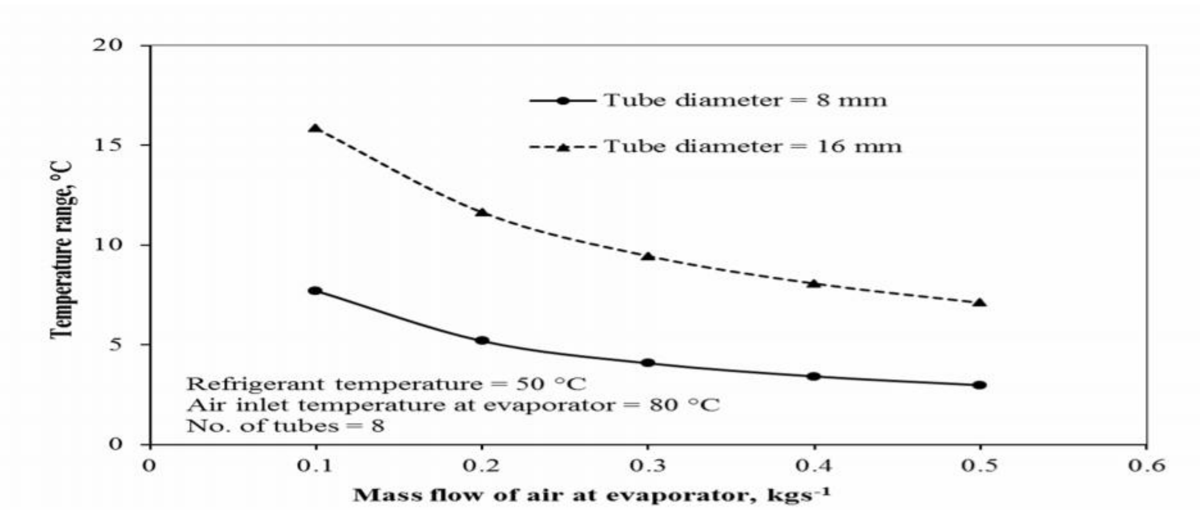


Fig.5 mass flow of air at evaporator vs. temperature range

Effect of air inlet temperature

Figure 6 shows the effect of air inlet temperature at evaporator on overall heat transfer coefficient (U) across the heat pipe heat exchanger for 8 and 16 mm tubes. As temperature of air increases, refrigerant-side free convective heat transfer coefficient marginally increases due to increase in temperature potential. This results in marginal increase in overall heat transfer coefficient.

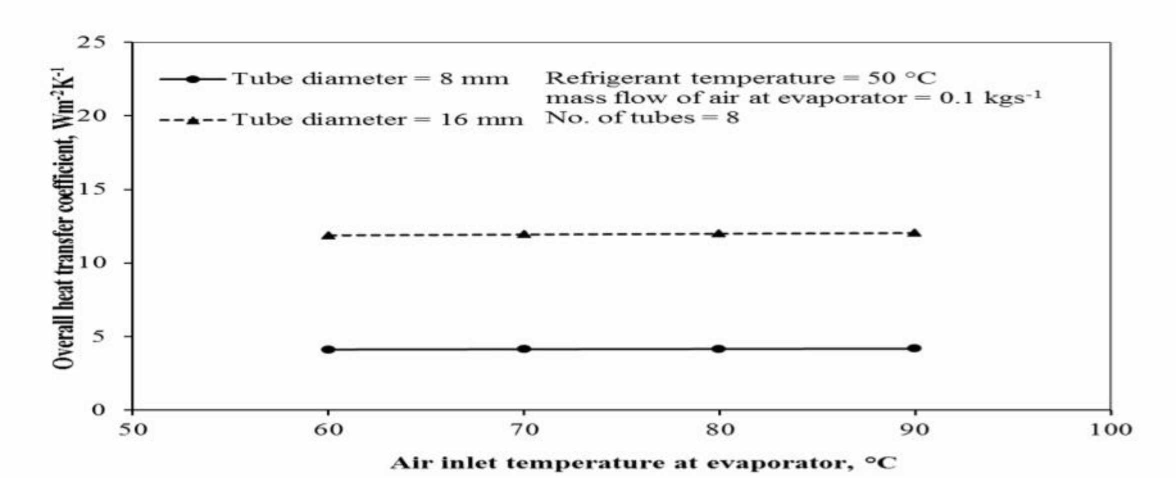


Fig.6 air inlet temperature at evaporator vs. overall heat transfer coefficient

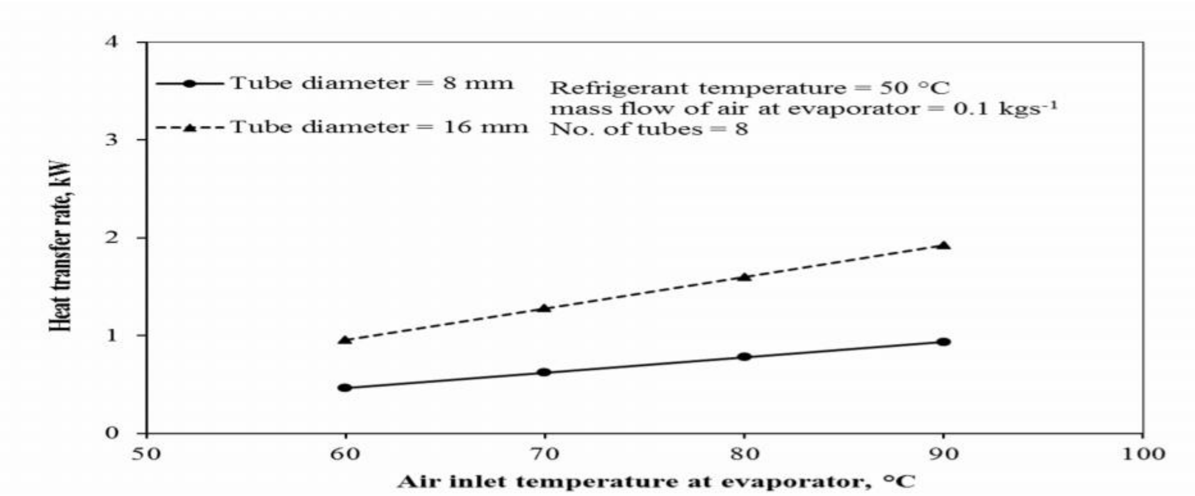


Fig.7 air inlet temperature at evaporator vs. heat transfer rate

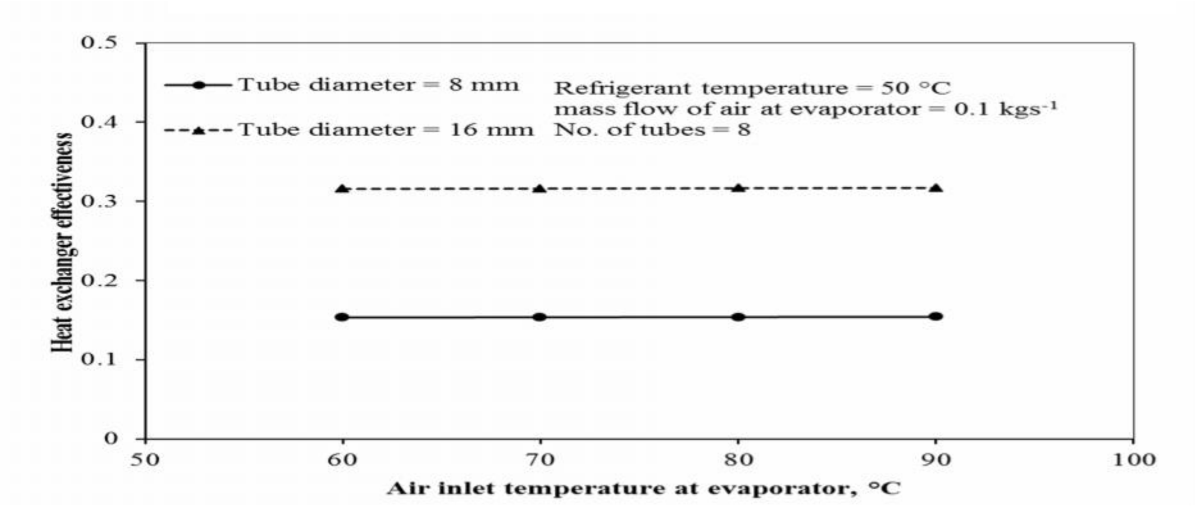


Fig.8 air inlet temperature at evaporator vs. heat exchanger effectiveness

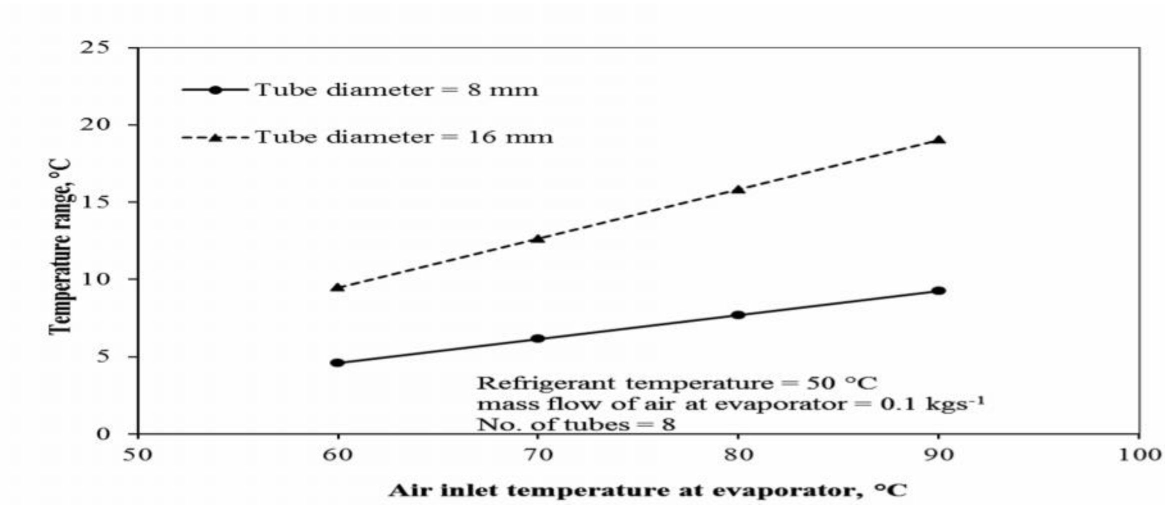


Fig.9 air inlet temperature at evaporator vs. temperature range

Figure 7 shows the effect of air inlet temperature on heat transfer rate across the heat pipe heat exchanger for 8 and 16 mm tubes. Heat transfer rate is a function of overall heat transfer coefficient. Hence, as air inlet temperature increases, heat transfer rate also increases due to increase in overall heat transfer coefficient (Fig. 6).

Figure 8 shows the effect of air inlet temperature on effectiveness of heat pipe heat exchanger for 8 and 16 mm tubes. Effectiveness is a function of number of transfer units (NTU), which is directly proportional to 'U' and inversely proportional to heat capacity rate of air (C_{air}). As air inlet temperature increases, 'U' increases and 'C' remains constant. This results in increase in NTU, leading to marginal increase in effectiveness.

Figure 9 shows the effect of air inlet temperature on temperature range across heat pipe heat exchanger for 8 and 16 mm tubes. Temperature range is a function of heat pipe heat exchanger effectiveness. Hence as air inlet temperature increases, temperature range also increases due to increase in effectiveness (Fig. 8).

Overall heat transfer coefficient, heat transfer rate, effectiveness and temperature range are more for 16 mm tube compared to 8 mm tube. The reason is that as tube diameter increases, minimum flow area across heat pipe heat exchanger decreases, resulting in increase in mass velocity. This leads to increase in air-side heat transfer coefficient and 'U'. Due to increase in 'U', heat transfer rate, effectiveness and temperature range are more for 16 mm tube.

Conclusion

Numerical studies have been carried out on the performance of methanol based air to air heat pipe heat exchanger for varying operating conditions. A computer simulation model has been developed to predict the thermal performance using ε-NTU method. Effects of air mass flow rate, inlet temperature and tube diameter on system performance have been investigated and the following conclusions been drawn from the simulation studies.

- Overall heat transfer coefficient and heat transfer rate across heat pipe heat exchanger increase as mass flow of air, inlet temperature of air and tube diameter increase
- Effectiveness and temperature range across heat pipe heat exchanger increase as inlet temperature of air and tube diameter increase whereas they decrease as mass flow of air increases.

These numerical studies provide a simple and effective approach for optimum sizing of heat pipe heat exchanger to give better performance.

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