THERMODYNAMIC PERFORMANCE EVALUATION OF AN AIR-AIR HEAT PIPE HEAT EXCHANGER

by

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In this paper, heat transfer analysis for an air-air heat exchanger was experimentally carried out to find its thermal performance and effectiveness. Air-air heat exchanger equipped with finned heat pipes was considered for the experimentation. Mass flow rate of air was considered in between 0.24 to 0.53 kg/s. The temperature at the condenser side of the heat pipe heat exchanger was kept constant at around 23 °C and at the evaporator part it was varied from 88 to 147 °C. The performance of heat pipe heat exchanger was evaluated at different mass flow rate of air, in terms of effectiveness and compared with its corresponding value found by theoretical analysis.

Key words: *heat pipe heat exchanger, thermodynamic performance, heat transfer coefficient, effectiveness, Reynolds number*

Introduction

With the advent of the modern hi-tech era, the demand for energy is increasing at a rapid rate. Because of this, in a few years' time the world is bound to face an energy crisis. In lieu of this fact it becomes our utmost duty to save as much of energy as possible. Hence, we should always try to identify areas for possible energy saving. One such possible area of work lies in air-conditioning systems as, in these systems, the waste heat transfer rate is poor. Thus, by working in this area, we would be able to contribute towards energy saving. In air-conditioning applications the incoming fresh air can be pre-cooled using the outgoing exhaust air to reduce the cooling load. But the amount of heat transfer between hot and cold fluids in air-air waste heat recovery systems is not high due to the fact that the air side heat transfer coefficient is low. Hence, many heat transfer devices have been developed for increasing the air side heat transfer coefficient. The chief of these is heat pipe.

The heat pipe concept was first put forward by Gaugler [1] of the General Motor corporation, Ohio, USA. However, the heat pipe as proposed by Gaugler [1] was not developed beyond the patent stage, and it was not patented until Grover [2] rediscovered the heat pipe. The work proposed by Grover [2] includes theoretical analysis and presents results of experiments carried out on stainless steel heat pipes with a wire mesh wick and sodium, lithium and silver as

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working fluids. In due course of time, interest in heat pipes as a means of heat transfer increased with considerable amount of work done. As a result, these devices began to be used in applications such as in rotary heat exchanger [3], waste heat recovery systems in automobiles [4] and in hospitals [5], latent heat storage systems [6-9], energy storage in wind power systems [10], aeroponic system [11], nuclear reactor [12, 13], thermionic converters [14, 15], and electronics cooling applications [16]. Review of the application of heat pipes in modern heat exchangers can be found in Vasiliev[17].

The thermal performance of the heat recovery systems using heat pipe heat exchanger is dependent on the effectiveness of the heat pipe heat exchanger which in turn depends on input heat transfer rate, the evaporator length, working fluid and its filling ratio [18-20]. Investigations were carried out to find the thermal performance and effectiveness of heat recovery systems using heat pipe heat exchanger [21-25]. Heat pipe heat exchangers were theoretically designed and validated with experimental analysis to find their optimum effectiveness for low temperature field applications [5] and air-air waste heat recovery systems [21-23]. It was found that the optimum effectiveness was obtained when the fresh air inlet temperature is near the operating temperature of the heat pipes. Heat pipe heat exchanger was modeled analytically using NTU method [24] and by numerical simulations using CFD [25].

This paper focuses on heat pipe as a means of waste heat recovery as it is not only highly effective but also economically viable. It presents experimental testing of a horizontal air-air heat pipe exchanger carried out at different mass flow rates of hot and cold fluids. The effectiveness of air-air heat pipe heat exchanger in recovering waste heat was explored. The measured performance of the heat pipe heat exchanger has been compared with the theoretical results obtained by its sizing.

Performance evaluation of heat pipe heat exchanger

Performance evaluation of heat pipe heat exchanger is necessary to show that the heat pipe meets the requirements laid down during the design. A considerable number of variables including effectiveness, orientation with respect to gravity, vapor temperature, evaporator heat flux, *etc*. can be investigated experimentally by benchmark testing.

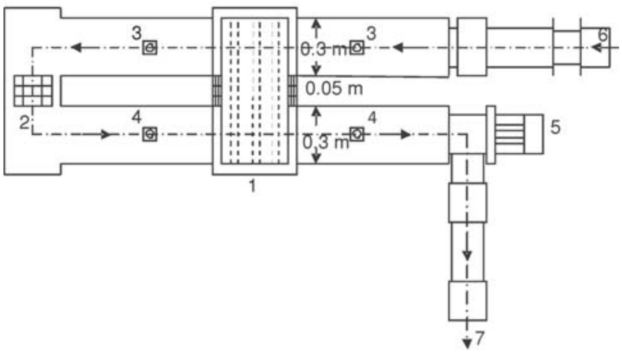

Figure 1. Experimental set-up (top view)
(1) Heat pipe heat exchanger, (2) Electric heater, (3) Pressure tapings at condenser section side, (4) Pressure tapings at evaporator section side, (5) Air blower, (6) Inlet, (7) Outlet

Experimental set-up

The experimental set-up used is shown in figs. 1 and 2. The set-up consists of a circumferentially finned tube heat pipe heat exchanger installed in a two sided rectangular air duct of size $0.65 \times 0.3 \times 0.196$ m^3. Heat pipe heat exchanger consisted of condenser, adiabatic, and evaporator sections. Condenser section (0.3 m × 0.3 m) of the heat pipe exchanger is placed on one side of the air duct and evaporation section (0.3×0.3 m^2) on the other side. The two sides of the duct were connected at one end by a U bend where an electric coil heater was fixed. At the exit section of the duct, air blower with variable speed control was provided to create air movement through the air duct. Pressure difference in the duct across the condenser and evaporator sections of the heat pipe heat exchanger was measured with the help of pressure gauges. At the exit section of the duct, air blower with variable speed control was provided to create air movement through the air duct. Pressure difference in the duct across the condenser and evaporator sections of the heat pipe heat exchanger was measured with the help of pressure gauges.

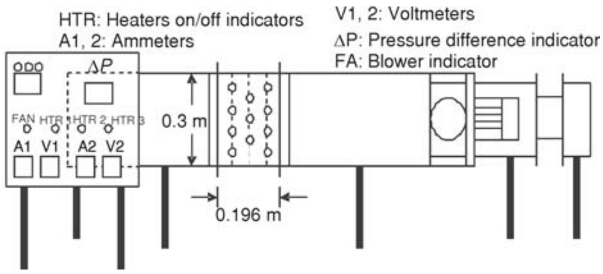

Figure 2. Experimental set-up (front view)

The heat pipe heat exchanger consisted of 22 identical and individually circumferentially finned aluminum heat pipes with water as the working fluid. The aluminum annular fins were mechanically bonded to the aluminum tubes and the contact resistance is negligible. The wick structure was a copper wire mesh of 2 mm thick and porosity, ε_w of 0.676. The wick consisted of 200 layers with permeability of 0.000077 m^2. The heat pipe heat exchanger was housed in a box made up of 2 mm GI plates with a separator plate of 3 mm thickness. The heat pipes were arranged in staggered grid arrangement. The horizontal and vertical spacing of the finned heat pipes were 38.1 and 44.45 mm, respectively. The outer diameter of the heat pipe including the fin was 25.4 mm and excluding the fin was 12.7 mm; the inner diameter of the heat pipe was 11.7 mm and wick structure diameter 9.7 mm. The total length of the heat pipe would be the total length of the duct including evaporator, condenser, and adiabatic sections. The heat pipes arrangement in the heat pipe heat exchanger is shown in fig. 3.

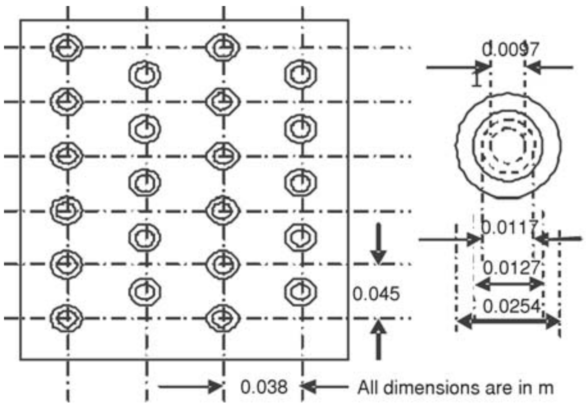

Figure 3. Cross-sectional view of the heat pipe heat exchanger

Mathematical formulation

The effectiveness of heat pipe heat exchanger for counter flow can be calculated by:

$$\varepsilon_{\text{the}} = \frac{1-\exp[-NTU(1-C)]}{1-C\exp[-NTU(1-C)]} \tag{1}$$

In eq. (1), NTU, the number of transfer units, and C, a constant, are given by:

$$NTU = \frac{UA}{C_{\min}} \quad \text{and} \quad C = \frac{C_{\min}}{C_{\max}} \tag{2}$$

In eq. (2):

$$C_{\min} = \min(\dot{m}_{\mathrm{hf}} c_{\mathrm{hf}}, \dot{m}_{\mathrm{cf}} c_{\mathrm{cf}}) \quad \text{and} \quad C_{\max} = \max(\dot{m}_{\mathrm{hf}} c_{\mathrm{hf}}, \dot{m}_{\mathrm{cf}} c_{\mathrm{cf}}) \tag{3}$$

The suffixes hf and cf represent air on evaporator and condenser side, respectively.

In eq. (2), UA, the product of overall heat transfer coefficient and overall heat transfer area, is given by:

$$UA = \frac{1}{\Sigma R} \tag{4}$$

In eq. (4), ΣR, the total thermal resistance exiting across the heat pipe heat exchanger, is given by:

$$\Sigma R = (R_{\mathrm{conv}} + R_{\mathrm{sca}} + R_{\mathrm{cond}})_{\mathrm{hf}} + (R_{\mathrm{conv}} + R_{\mathrm{sca}} + R_{\mathrm{cond}})_{\mathrm{cf}} + R_{\mathrm{whf}} + R_{\mathrm{wcf}} \tag{5}$$

$$R_{\mathrm{conv}} = \frac{1}{\eta_{\mathrm{o}} h A} \quad \text{where} \quad \eta_{\mathrm{o}} = 1 - \frac{A_{\mathrm{f}}}{A}(1 - \eta_{\mathrm{f}}) \tag{6}$$

$$R_{\mathrm{sca}} = \frac{R_{\mathrm{conv}}}{\eta_{\mathrm{o}} A} \tag{7}$$

$$R_{\mathrm{cond}} = \frac{\ln \dfrac{r_{\mathrm{o}}}{r_{\mathrm{i}}}}{2\pi k L_{\mathrm{t}} N_{\mathrm{t}}} \tag{8}$$

In eq. (6), the convective heat transfer coefficient, h, can be calculated from eq. (6.4) given in [26]:

$$\mathrm{Nu} = \frac{h d_{\mathrm{h}}}{k} = 0.248\,\mathrm{Re}^{0.6}\,\mathrm{Pr}^{1/3} \left(\frac{A}{A_{\mathrm{p}}} \right)^{-0.15} F^{-0.075} \left(\frac{S_{\mathrm{t}}}{S_{\mathrm{d}}} \right)^{1.06} \tag{9}$$

In eq. (9), the Reynold's number, the Prandtl number, and F, a constant, are given by:

$$\mathrm{Re} = \frac{\rho u_{\mathrm{a}} d_{\mathrm{h}}}{\mu}, \quad \mathrm{Pr} = \frac{\nu}{\alpha}, \quad F = \frac{1}{\dfrac{A'}{A_{\mathrm{ff}}} + 1}, \quad \text{and} \quad d_{\mathrm{h}} = \frac{4 d_{\mathrm{c}} A_{\mathrm{ff}}}{A'} \tag{10}$$

In eq. (10), A' is the heat transfer area of unit length of the finned tube and A_{ff} – the minimum free flow area of the finned tube per unit length; they are given by:

$$A' = \pi d_{\mathrm{o}} \left(1 - \frac{\delta_{\mathrm{f}}}{S_{\mathrm{f}}} \right) + \frac{\pi}{S_{\mathrm{f}}} \left(\frac{d_c^2 - d_o^2}{2} + d_{\mathrm{c}} \delta_{\mathrm{f}} \right) \quad \text{and} \quad A_{\mathrm{ff}} = S_{\mathrm{t}} - d_{\mathrm{c}} + (d_{\mathrm{c}} - d_{\mathrm{o}})(1 - \delta_{\mathrm{f}} N_{\mathrm{f}}) \tag{11}$$

In eq. (6), A and A_{f}, are given by:

$$A = A_p + A_{\mathrm{f}} = [\pi d_{\mathrm{o}} L_{\mathrm{t}} (1 - \delta_{\mathrm{f}} N_{\mathrm{f}})] N_{\mathrm{t}} + \left(\pi L_{\mathrm{t}} \frac{d_c^2 - d_o^2}{2} + d_{\mathrm{c}} \delta_{\mathrm{f}} \right) N_{\mathrm{f}} N_{\mathrm{t}} \tag{12}$$

In eq. (6), the fin efficiency, η_{f}, is calculated from:

$$\eta_{\mathrm{f}} = \frac{\tanh(m l_{\mathrm{e}})}{m l_{\mathrm{e}}}, \quad \text{where} \quad m = \sqrt{\frac{2h}{k_{\mathrm{f}} \delta_{\mathrm{f}}}} \quad \text{and} \quad l_{\mathrm{e}} = L_{\mathrm{hf}} \ \text{or} \ L_{\mathrm{cf}} \tag{13}$$

In eq. (5), R_{whf} and R_{wcf} are the wick resistances on the evaporator and condenser side of the heat pipe heat exchangers. Assuming that the heat fluxes on condenser and evaporator sections are low and the heat transfer through the wick is due to conduction process only, R_{whf} and R_{wcf} are given by:

$$R_{\mathrm{whf,\,wcf}} = \frac{\ln \dfrac{r_{\mathrm{i}}}{r_{\mathrm{v}}}}{2\pi l_{\mathrm{e}} k_{\mathrm{e}} N_{\mathrm{t}}} \quad \text{where} \quad k_{\mathrm{e}} = \frac{k_1[(k_1 + k_{\mathrm{w}}) - (1-\varepsilon_{\mathrm{w}})(k_1 - k_{\mathrm{w}})}{(k_1 + k_{\mathrm{w}}) - (1-\varepsilon_{\mathrm{w}})(k_1 - k_{\mathrm{w}})} \tag{14}$$

In eq. (14), k_{l} and k_{w}, the thermal conductivities of heat pipe liquid (water) and wick material (copper), are assumed to be 0.6616 and 396.78 W/mK, respectively.

Experimental procedure

Air blower was allowed to operate at different speeds, so that it created variable pressure difference within the unit to cause changes in the mass flow rate of the air. Variation of mass flow rate of air was considered between 0.23-0.54 kg/s. Atmospheric air (at an average temperature of 22.5 °C) entered the set-up at the inlet side of the duct, and flew over the condenser section of the heat pipe heat exchanger. After that the air flew over the electric heater, where its temperature increased, and moved to the evaporator section of the heat pipe heat exchanger. Temperature of the air at the inlet to the evaporator section was adjusted (between 80 and 150 °C) by varying the energy input to the electric heater. Experiment was carried out and the following observations were recorded in tab. 1.

Table 1. Experimental reading: pressure and temperatures

S. no.	Pressure drop, Δp, [mm of H_2O]	Velocity, u_{a}, [ms^{-1}]	Exhaust air temperature, [°C]		Fresh air temperature, [°C]	
			Evaporator in, T_{hi}	Evaporator out, T_{ho}	Condenser in, T_{ci}	Condenser out, T_{co}
1	4	3.5052	147	84	23	60
2	5	3.9624	130	80	23	55
3	6	4.5212	117	76	23	49
4	7	4.6736	111	75	23	48
5	8	5.1308	107	72	23	46
6	9	5.4559	102	70	23	46
7	10	5.7759	94	68	22	42
8	11	6.0452	92	66	22	40
9	12	6.3195	90	65	22	40
10	13	6.5735	88	64	22	38

Mass flow rate of air across the heat pipe heat exchanger was calculated using:

$$\dot{m}_{\mathrm{a}} = \rho A_{\mathrm{duct}} (u_{\mathrm{a}})_{\mathrm{STP}} = \rho \frac{P_{\mathrm{a}} V_{\mathrm{a}} T_{\mathrm{STP}}}{P_{\mathrm{STP}} T_{\mathrm{a}}} \tag{15}$$

In eq. (15), $(u_{\mathrm{a}})_{\mathrm{STP}}$ is the velocity of air reduced to STP (temperature equal to 15 °C and pressure 1 atmospheric bar) conditions. P_{a} and V_{a} $\{= u_{\mathrm{a}}.A_{\mathrm{duct}}\}$, the pressure and the volume flow rate of air, were measured from the experiment. T_a is the average of temperatures at the inlet of evaporator and condenser sections. Property values of air were considered corresponding to the temperature, T_a.

Assuming that the heat pipe heat exchanger configuration is reduced to a double pipe concentric counter flow type of heat exchanger, the actual quantity of heat transfer between evaporator and condenser side can be calculated by:

$$Q_{\mathrm{act}} = UA\Delta T_{\mathrm{LMTD}} = UA\frac{\Delta T_{\mathrm{in}} - \Delta T_{\mathrm{out}}}{\ln\dfrac{\Delta T_{\mathrm{in}}}{\Delta T_{\mathrm{out}}}};\ \Delta T_{\mathrm{in}} = T_{\mathrm{hi}} - T_{\mathrm{co}} \text{ and } \Delta T_{\mathrm{out}} = T_{\mathrm{ho}} - T_{\mathrm{ci}} \tag{16}$$

The maximum quantity of heat that can be transferred between evaporator and condenser sides, is given by:

$$Q_{\max} = C_{\min}(T_{\mathrm{hi}} - T_{\mathrm{ci}}) \tag{17}$$

Effectiveness of heat pipe heat exchanger is given by:

$$\varepsilon_{\exp} = \frac{Q_{\mathrm{act}}}{Q_{\max}} \tag{18}$$

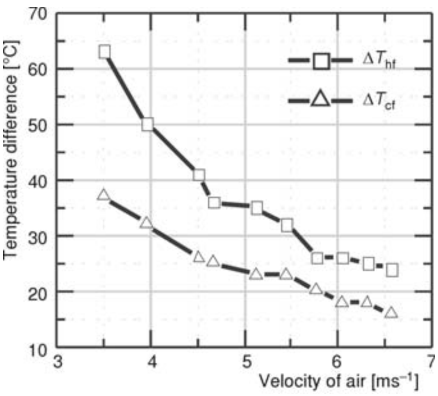

Figure 4. Variation of temperature difference with the velocity of air

Results and discussion

It was observed from the experimental results that, as the velocity of air changed from 3.5 to 6.5 m/s, temperature difference (ΔT_{hf}) between inlet and exit points across the evaporator section changed from 63 to 24 °C, and that across the condenser section, ΔT_{cf} changed from 37 to 16 °C. Variation of ΔT_{hf} and ΔT_{cf} with velocity of air is shown in fig. 4. ΔT_{hf} was found to be higher than ΔT_{cf} at all the values of velocity of air. It was also found that, as the value of velocity of air increased, the value of ΔT_{hf} decreased at a faster rate than the value of ΔT_{cf}. Further, the difference between ΔT_{hf} and ΔT_{cf} decreased with the increase in velocity of air.

The value of mass flow rate of air was calculated using eq. (15). Convection heat transfer coefficient and Reynolds number at different mass flow rates of air were calculated using eq. (9) and eq. (10), respectively. The property values of air were considered corresponding to the mean temperature of inlet and exit of the evaporator and condenser sections for studying related Reynolds number and the heat transfer coefficient. The variation of convection heat transfer coefficient and Reynolds number with varying mass flow rate of air is shown in fig. 5 (a) across evaporator section and (b) across condenser section.

It was found from fig. 5(a) and (b) that, when the mass flow rate of air on evaporator side increased from 0.285 to 0.572 kg/s, Re increased from 2915 to 6090, and h increased from 206 to 321 W/m^{2}K. For the condenser side, Re increased from 4694 to 9145 and h increased from 275 to 411 W/m^{2}K. As the Reynolds number and heat transfer coefficient are linearly proportional to the velocity of air, both Reynolds number and heat transfer coefficient increased linearly with mass flow rate of air (fig. 5). Further, for a given mass flow of air, both the Reynolds

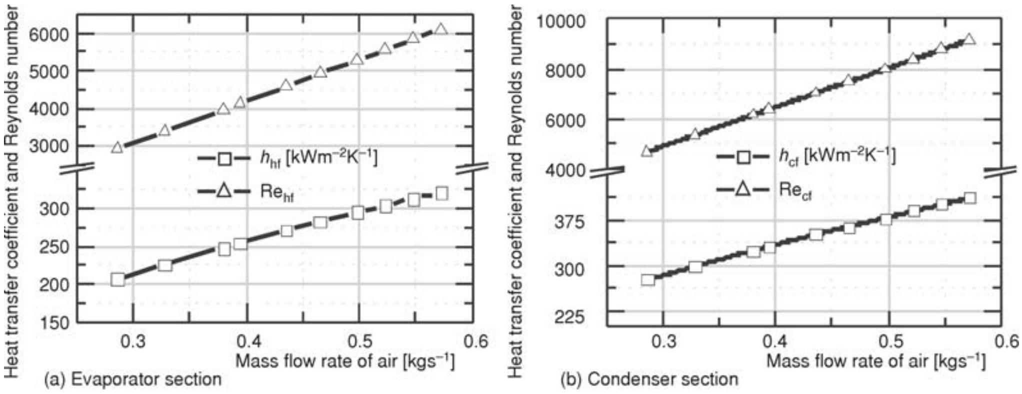

Figure 5. Variation of heat transfer coefficient and Reynolds number with mass flow rate of air

number and heat transfer coefficient values for air on the condenser section side were higher than those of air on the evaporator section. As the temperature range on the cold air side (*i. e.* at the condenser section) is lower than that on the hot air side (*i. e*. at the evaporator section), the more for the cold air. As a result, for the same mass flow rate of air, the Reynolds number and heat transfer coefficient values are more on the condenser section.

The value of *UA*, the product of overall heat transfer coefficient and overall surface area, is calculated using eq. (4), and total thermal resistance for heat flow was calculated using eq. (5). Figure 6(a) shows the variation of various thermal resistances across the heat pipe heat exchanger and the total thermal resistance for different mass flow rate of air for hot air section and (b) cold air section.

It was found on the evaporator section side that, as the mass flow rate of air increased, R_{conv} decreased from $0.13 \cdot 10^{-4}$ to $0.93 \cdot 10^{-4}$ K/W, R_{sca} decreased from $0.37 \cdot 10^{-4}$ to $0.28 \cdot 10^{-4}$ K/W. However, on the condenser side, as the mass flow rate of air increased, R_{conv} decreased from $0.11 \cdot 10^{-4}$ to $0.075 \cdot 10^{-4}$ K/W, R_{sca} decreased from $0.31 \cdot 10^{-4}$ to $0.23 \cdot 10^{-4}$ K/W. It can be seen from fig. 5 that R_{cond} (= $0.0942 \cdot 10^{-4}$ K/W) and R_{whf} (= $34.1 \cdot 10^{-4}$ K/W) were independent of the mass flow rate of air. The convection thermal resistance decreased with the increasing mass

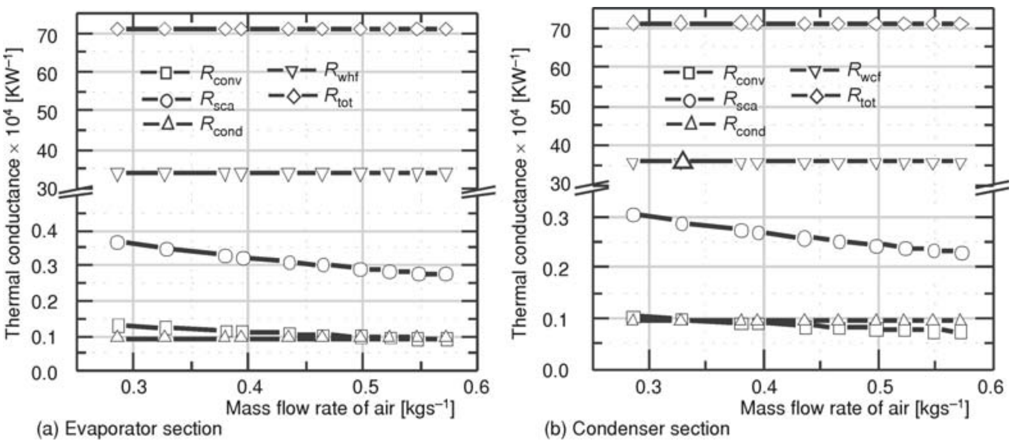

Figure 6. Variation of thermal resistances with mass flow rate of air

flow rate of air for both condenser and evaporator section of the heat pipe heat exchanger. This is true as the convection heat transfer coefficient increases with the mass flow rate; hence, the convective thermal resistance decreased. It can also be found from fig. 6 that values of R_{conv} and R_{sca} are more for the evaporation section of the heat pipe exchanger compared to those of condenser section. This is because of the fact that temperature limits on the evaporator section are higher than those on the condenser section.

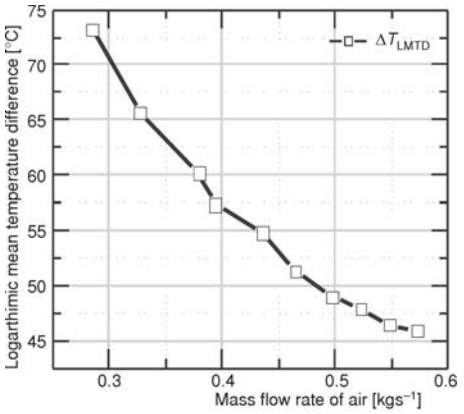

Figure 7. Variation of logarithmic mean temperature difference with mass flow rate of air

The value of logarithmic mean temperature difference, ΔT_{LMTD}, was determined using eq. (16). Figure 7 shows the variation of ΔT_{LMTD} with mass flow rate of air. As the mass flow rate of air increased from 0.285 to 0.572 kg/s, ΔT_{LMTD} decreased from 73.23 to 45.88 °C. This is due to the fact that the values of both ΔT_{hf} and ΔT_{cf} decreased with increasing mass flow rate of air.

The values of Q_{act} and Q_{max} were also calculated using eqs. (16) and (17). Figure 8 shows the variation of Q_{act} and Q_{max} with mass flow rate of air. Q_{act} decreased (from 10.30 to 6.47 kW) with increasing value of mass flow rate of air (from 0.285 to 0.572 kg/s). This is due to the decreasing value of ΔT_{LMTD} with mass flow rate of air. However, the value of Q_{max} increased (from 29.35 to 32.65 kW) with respect to the increase in mass flow rate of air. Even though the value of $T_{hi} - T_{ci}$ decreased with the mass flow rate of air, the value of C_{min} increased. The rate at which C_{min} increased was higher than the rate at which the value of $T_{hi} - T_{ci}$ decreased. As a result, Q_{max} increased with respect to the increase in mass flow rate of air.

The values of theoretical and experimental effectiveness of the heat pipe heat exchanger were calculated using eqs. (1) and (18), respectively. Figure 9 shows the variation of effectiveness of heat exchanger for different mass flow rate of air across the heat pipe heat

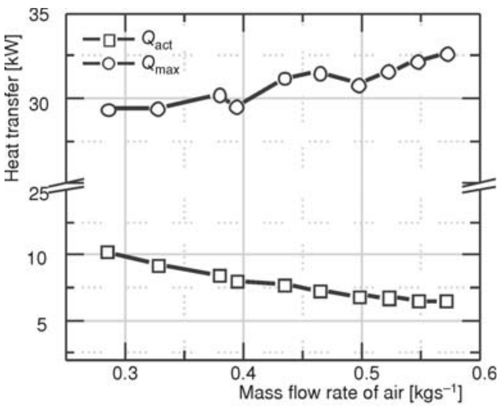

Figure 8. Variation of heat transfer with mass flow rate of air

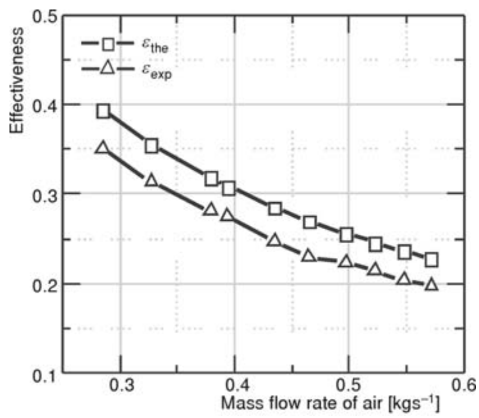

Figure 9. Variation of effectiveness with the mass flow rate of air

exchanger. ε_{the} decreased (from 0.39 to 0.23) with increasing value of mass flow rate of air (from 0.285 to 0.572 kg/s). Similarly, the value of ε_{exp} decreased (from 0.35 to 0.2) with respect to the increase in mass flow rate of air. It was found from fig. 8 that rate at which Q_{max} increased was higher than rate at which Q_{act} decreased, hence the value of ε_{exp} decreased with the increase in mass flow rate of air.

The theoretical and experimental values of effectiveness were well in agreement with each other. The difference between the two was decreasing with increasing mass flow rate of air, which is due to at higher mass flow rates the rate at which ε_{exp} decreased was lower than the rate which ε_{the} decreased. The average percentage difference between the theoretical and experimental values of effectiveness was found to be 12.5%.

Conclusion

In this paper, experimental heat transfer analysis for an air-air heat exchanger has been carried out to find its thermal performance and effectiveness. The measured performance of the heat pipe heat exchanger has been compared with the theoretical results obtained by its sizing. It was observed from the experimental results that, as the velocity of air was increased, the temperature difference between inlet and exit sides across the evaporator and condenser sections decreased.

As the Reynolds number and heat transfer coefficients are directly proportional to the velocity of air flow, it was found that convective heat transfer coefficient and Reynolds number increased linearly with increasing mass flow rate of air for both condenser and evaporator sections of the heat pipe heat exchanger. As the mass flow rate of air increased, the thermal resistance for the convective heat transfer and thermal resistance offered by the scaling decreased. However, the thermal resistance for conduction heat transfer and thermal resistance offered by the wick structure in the heat pipe are independent of the mass flow rate of air as they are dependent on the material property. Even though mass flow rate of air affects the convective thermal resistance and resistance offered by the scaling factor, the overall thermal resistance across the heat pipe heat exchanger did not significantly change with the mass flow rate of air. As the temperature difference available at the condenser and evaporator sections decreased, the value of logarithmic mean temperature difference decreased with corresponding increase in mass flow rate. The actual quantity of heat transferred decreased with increasing mass flow rate of air.

Nomenclature

A – the total surface area of the heat pipe heat exchanger, [m^2]
A_{duct} – cross-sectional area of the duct, [m^2]
A_f – the fin surface area, [m^2]
A_{ff} – the minimum free flow area of the finned tube per unit length, [m^2]
A_p – the primary surface area available, [m^2]
A' – the heat transfer area of unit length of the finned tube, [m^2]
C – a constant equal to the ratio of specific heat capacities, [–]
C_{min} – minimum heat capacity of either hot or cold fluid, [W°C^{-1}]
C_{max} – maximum heat capacity of either hot or cold fluid, [W°C^{-1}]
$c_{hf,cf}$ – specific heat of hot and cold fluids, [Jkg^{-1}K^{-1}]
d_c – diameter of the heat pipe with fin, [m]
d_h – hydraulic diameter, [m]
d_o – outer diameter of the heat pipe, [m]
F – constant, [–]
H – convective heat transfer coefficient, [Wm^{-2}K^{-1}]
K – thermal conductivity of the heat pipe and fin material, [Wm^{-1}K^{-1}]
k_l – thermal conductivity of heat pipe working fluid, [Wm^{-1}K^{-1}]
k_w – thermal conductivity of heat pipe wick material, [Wm^{-1}K^{-1}]

k_e – equivalent thermal conductivity of heat pipe, [$Wm^{-1}K^{-1}$]
L_t – length of the heat pipe, [m]
$L_{hf,cf}$ – length of the evaporator and condenser sections, [m]
l_e – length of the heat pipe on either condenser or evaporator side, [m]
M – constant, [–]
$\dot{m}_a$ – mass flow rate of air, [kgs^{-1}]
Nu – Nusselt number ($= hd_h/k$), [–]
NTU – number of transfer units ($= UA/C_{min}$), [–]
N_f – number of fins per unit length of the heat pipe, [–]
N_t – number of tubes in the heat pipe heat exchanger, [–]
Pr – Prandtl number ($= \upsilon/\alpha$), [–]
Q_{act} – actual quantity of heat transferred, [W]
Q_{max} – maximum possible quantity of heat that can be transferred, [W]
r_i – inner radius of the heat pipe, [m]
r_o – outer radius of the heat pipe, [m]
Re – Reynolds number ($\rho u_a d_h/\mu$), [–]
R_{conv} – convective thermal resistance, [KW^{-1}]
R_{cond} – conduction thermal resistance, [KW^{-1}]
R_{sca} – thermal resistance offered by the scaling, [KW^{-1}]
$R_{whf,wcf}$ – thermal resistance by the wick at evaporator or condenser section, eq. (14), [KW^{-1}]
S_d – diagonal pitch of the tubes, [m]
S_F – fin pitch, [m]
S_t – transverse pitch of the tubes, [m]
T_{hi}, T_{ho} – temperature of the air at inlet and outlet points of the evaporator section of the heat pipe heat exchanger, [K]
T_{ci}, T_{co} – temperature of the air at inlet and outlet points of the condenser section of the heat pipe heat exchanger, [K]
ΔT_{in} – temperature difference between hot and cold fluid at inlet of the heat exchanger, [K]
ΔT_{LIMD} – logarithmic mean temperature difference, [K]
ΔT_{out} – temperature difference between hot and cold fluid at the outlet of the heat exchanger, [K]
U – overall heat transfer coefficient, [$Wm^{-2}K^{-1}$]
u_a – velocity of air, [ms^{-1}]

Greek symbols

α – thermal diffusivity of air, [m^2s^{-1}]
δ_f – thickness of fin, [m]
η_f – efficiency of fin, [–]
η_o – efficiency factor, [–]
ε_w – porosity of wick material in the heat pipe, [m]
ε_{exp} – experimental value of effectiveness of heat pipe heat exchanger, [–]
ε_{the} – theoretical value of effectiveness of heat pipe heat exchanger, [–]
υ – kinematic viscosity of air, [m^2s^{-1}]
ρ – density of the air, [kgm^{-3}]

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