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Passive Cooling Method of Energy Reduction Using Fins-Type Heat-Pipe Heat Exchanger

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Abstract

A passive cooling method has turned the comfort cooling technology inexpensive to the operating cost and contributes to lower the threat of environmental pollution. Current cooling technology relies on electricity sources to operate within its cycles by manipulating temperature difference. A passive cooling method using a heat-pipe heat exchanger changes the phase of a refrigerant fluid by evaporating and condensing, transfers the heat within its tube. The study investigated the air passing through the heat-pipe heat exchanger before entering a room. The effort comes from a gravity assisted R134a heat-pipe heat exchanger temperature reduction ability as a pre-cooled component that does not need power sources to operate. The heat-pipe heat exchanger is a row of 3 by 3 tubes, 9 mm ID straight copper tube with inner grooves structured, length of 500 mm fitted with square type aluminum fins. Each of the copper tubes is quarterly filled with liquid R134a as a refrigerant medium. Acrylic box in the size of 1000 mm x 820 mm x 620 mm, modeled as a room, used to collect air diffusion and the temperature distributions. The location of the heat-pipe heat exchanger had been set at the top and the side wall of the acrylic box to measure the best heat transfer conditions. A computational fluid dynamic software CFD-ACE and ANSYS-Fluent are being used to run simulations of the experiment. It is found that the heat-pipe heat exchanger has the capability of pre-cooling a room fresh air up to 9 K different. The cooling energy consumption had been calculated to save up between 90 W/m^3 to 273 W/m^3 or about 25% to 33% to the box volume.

Keywords: *computational fluid dynamics, heat-pipe heat exchanger, passive cooling*

Introduction

Researchers [1], [2], [3] and [4] have proposed a passive cooling system using methanol as a cooling medium and the proposal is mostly in a heat recovery system application [5], [6], [7]. This study proposed a method of pre-cooling of an air intake using a straight heat-pipe heat exchanger to reduce ambient heat. By decreasing the temperature, the energy consumption could be reduced while increasing the work done of the room's air-conditioner evaporator coil. Simulation process of the heat-pipe heat exchanger, effectiveness and the declining temperature is experimentally tested.

Problem Description: R134a Heat-Pipe Heat-Exchanger

The purpose of this study is to investigate the possibility of reducing the high outside air temperature entering a room, using heat-pipe heat exchanger as a passive cooling equipment. Copper tubes with grooves inner structure are being used to show the capabilities of a fin-type heat-pipe heat exchanger. Refrigerant R134a as a safe, zero global warming potential (GWP) index is exploited as a heat transfer medium. Figure 1 shows the working principle of the heat-pipe. Table 1 shows the temperature difference comparison for the cooling medium.

Figure 1. The working principle of a heat-pipe heat exchanger.

Methodology

The experiments study on the heat sink of a heat-pipe heat exchanger evaporator section ability that absorbed heat from the surrounding air. As the liquid evaporates, and by the tube void factor, the vapor pressurizes and transfers heat to the condenser section.

Parameters	Water	Acetone	R134a
Density kg/m^3	998.2	791	1207.3
Spec. Heat J/kg.K	4182	2160	1424.41
Therm. Cond. W/m. K	0.6	0.18	0.08119
Viscos kg/m-s	1.003x10^{-3}	3.31x10^{-4}	1.9525x10^{-4}
Inlet Temp, K	293	293	293
Outlet Temp, K	295	296	298
Diff In-Out Temp, K	**2**	**3**	**5**

Table 1. The cooling medium properties are taken from the ANSYS-Fluent material database and the result shows the difference temperature at the outlet that the fluids can achieve.

The experiment set-up is shown in Figure 2. Figure 3 shows the configurations of the air inlet and outlet openings. Suggested by [4], the author used the hot ambient temperature in the range of 303K (30°C) to 318K (45°C) for the inlet temperature. A heat-pipe heat exchanger is installed on the façade of the box before the supply air enters the acrylic box. Suggested by [8], heat-pipe inclination position should be at least 6º with respect to the horizontal; this experiment had taken a position of 10º inclination for the refrigerant gravitational-return purpose.

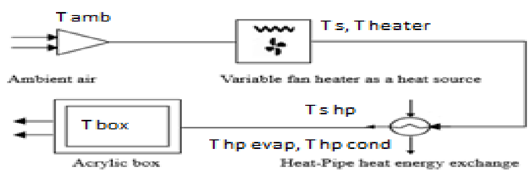


Figure 2: The work-flow of the experimental project.

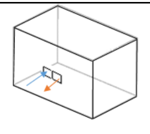
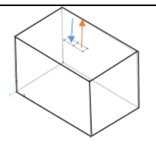
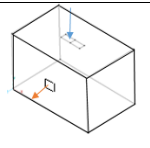
Case	Flow	Specification
Case 1		Incoming air at 303K and 318K flows on the surface of a heat-pipe heat exchanger from the side and exit on the side wall.
Case 2		Air at 303K and 318K flows on the surface of a heat-pipe heat exchanger from the top and exit on the top side.
Case 3		Air at 303K and 318K flows on the surface of a heat-pipe heat exchanger from the top and exit at the side of the box.

Figure 3: Models of acrylic boxes configuration where fresh air intake locations and openings that best suits the domain.

CFD Equation and Boundary Conditions

The CFD used standard 3D Navier-Stokes equation, continuity and energy equations using the RANS method of simulation. Table 2 shows the boundary conditions and the cells used to run the models. When the fundamental laws of mechanics applied to a fluid, the conservation of mass is,

$$\frac{\partial U_j}{\partial x_j} = 0 \qquad (1)$$

And the momentum equations are,

$$\frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} = -\frac{1}{\rho}\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[v\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right) - \overline{u_i u_j}\right] - \beta(T - T_{ref})g_i \qquad (2)$$

$$-\overline{u_i u_j} = v_t\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right) - \frac{2}{3}K\delta_{ij}$$

$$\rho c\left(\frac{\partial T}{\partial t} + U_j\frac{\partial T}{\partial x_j}\right) = k\frac{\partial^2 T}{\partial x_j \partial x_j} - \rho c\frac{\partial}{\partial x_j}\left(\overline{u_j T'}\right) + \Phi + \emptyset \qquad (3)$$

$$\overline{u_j T'} = \frac{v_t}{\sigma_t}\left(\frac{\partial T}{\partial x_j}\right)$$

$$\Phi = \mu\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)\frac{\partial U_i}{\partial x_j} \qquad \emptyset = \mu\left[\overline{\left(\frac{\partial u_i}{\partial x_j}\right)\left(\frac{\partial u_i}{\partial x_j}\right)} + \overline{\left(\frac{\partial u_i}{\partial x_j}\right)\left(\frac{\partial u_i}{\partial x_j}\right)}\right]$$

$$\frac{\partial K}{\partial t} + U_j\frac{\partial K}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(v + \frac{v_t}{\sigma_K}\right)\frac{\partial K}{\partial x_j}\right] + v_t\left[\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)\frac{\partial U_i}{\partial x_j} + \frac{\beta}{\sigma_t}g_j\frac{\partial T}{\partial x_j}\right] - \varepsilon \qquad (4)$$

$$K = \frac{1}{2}\overline{U_i U_j}$$

$$\varepsilon = v\,\overline{\left(\frac{\partial u_i}{\partial x_j}\right)\left(\frac{\partial u_i}{\partial x_j}\right)}$$

$$\frac{\partial \varepsilon}{\partial t} + U_j\frac{\partial \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(v + \frac{v_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_1\frac{\varepsilon}{K}v_t\left[\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)\frac{\partial U_i}{\partial x_j} + \frac{\beta}{\sigma_t}g_j\frac{\partial T}{\partial x_j}\right] - C_2\frac{\varepsilon^2}{K} \qquad (5)$$

Where subscript t refers to turbulence,

$$\mu_t = {\rho C_\mu K^2}/{\varepsilon}\,;\; v_t = {\mu_t}/{\rho};$$

C_μ=0.09; C_1=1.44; C_2 =1.92; σK =1.0; σ_ε=1.3; reference temperature = 300K. For the simulation, the pressure assumed to be constant, the K and ε value at inlet used;

$$K = \frac{3}{2}(U_{ave}T_i)^2 \qquad (6)$$

$$\varepsilon = \frac{C_\mu^{3/4} K^{3/2}}{KL} \qquad (7)$$

Where U_{ave} is inlet velocity, T_i is turbulence intensity, L is taken to be a “reasonable length” of 1m and κ = 0.41 is the Von Karman constant.

Domain	Heat Pipe
General	Pressure based, Absolute velocity, Time: steady, 3D planar, Gravity: Y direction -9.81 m/s^2
Model	Energy: on, Viscous: laminar
Material	Fluid: air, Solid: copper
Boundary conditions	Velocity magnitude: 1m/s, Thermal: 300K. Pressure outlet, Back pressure: 300K. Inlet temp 303K to 318K Heat pipe tube temp 295K
Wall	Surface body: stationary wall, no slip
Solution Methods	Scheme: simple, Gradient: Least square cell based, Pressure: Standard, Momentum: Power law, Energy: Power law
Total cells	655360
Total Nodes	595820

Table 2: Boundary condition, cells and nodes runs on ANSYS Fluent and CFD-ACE.

Result and Discussion

Figure 4 and 5 show the temperature changes and the air distributions. Table 3, 4 and 5 show the results of the experiments, for all cases.

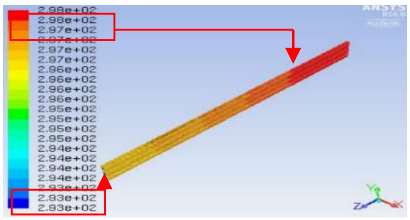


Figure 4: Simulation on heat pipe heat exchanger for R134a by ANSYS-Fluent**.**

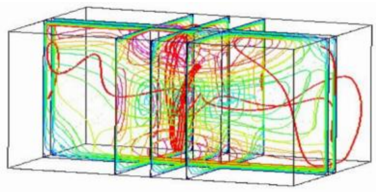


Figure 5: Simulation of air distribution by CFD-ACE.

Case 1

Parameters	Case 1	
Supply Air Temp, K	303	318
Heat-pipe Evaporator Section Temp K	301	332
Heat-pipe Condenser Section Temp K	295	311
Average Box Temp K	293	302
Different Temp of Heat-pipe Evaporator and Condenser Section, K	6	21
Different of Supply Air Temp and Heat-pipe Condenser Section Temp K	**8**	**7**

Table 3: The parameters for Case 1.

Case 2

Parameters	Case 2	
Supply Air Temp, K	303	318
Heat-pipe Evaporator Section Temp K	299	331
Heat-pipe Condenser Section Temp K	296	310
Average Box Temp K	294	301

Different Temp of Heat-pipe Evaporator and Condenser Section, K	3	21
Different of Supply Air Temp and Heat-pipe Condenser Section Temp K	**7**	**8**

Table 4. The parameters for Case 2.

Case 3

Parameters	Case 3	
Supply Air Temp, K	303	318
Heat-pipe Evaporator Section Temp K	297	333
Heat-pipe Condenser Section Temp K	294	311
Average Box Temp K	294	301
Different Temp of Heat-pipe Evaporator and Condenser Section, K	3	22
Different of Supply Air Temp and Heat-pipe Condenser Section Temp K	**9**	**7**

Table 5. The parameters for Case 3.

Estimated Energy Saving by the Heat-pipe Heat Exchanger

ASHRAE 55 specifies that a design temperature of 295.8 K (22.8°C) to 298 K (25°C) as a comfortable indoor temperature for domestic air conditioning systems. Taken the average temperature of 297 K (24°C) as a design indoor temperature and 303 K (30°C) outdoor temperature for domestic air conditioning systems, the sensible cooling load for the box can be calculated as Wan [9],

$$Q = 1.213 \times qv \times (t_r - t_o) \qquad (1)$$

where Q is the estimated sensible cooling load at standard sea level kW, 1.213 kJ/m^3. K (1.025 kJ/kg. K / 0.845 m^3/kg), qv is the air flow rate m^3/s and (t_r - t_o) is the delta temperature of the box and the supply air. The face area is taken as 0.0325 m^2.

The rate of energy saving can be calculated by,

$$\propto = \frac{Q_o' - Q_o}{Q_o'} \qquad (2)$$

Where $\propto$ is the rate of energy saving by the heat-pipe %, Q_o is the sensible load at constant indoor, (that is 0.280kW) and Q_o'is the calculated sensible load with heat-pipe heat exchanger.

Parameters	Case 1		Case 2		Case 3	
	303	**318**	**303**	**318**	**303**	**318**
SA-HP$_{cond}$, K	8	7	7	8	9	7
Q_o', kW	0.373	0.327	0.327	0.373	0.420	0.327
Energy saving Q_o'-Q_o, kW	0.93	0.47	0.47	0.93	**0.140**	0.47
$\propto$, % x 100	0.25	0.14	0.17	0.33	**0.33**	0.14

Table 5. The energy saving comparison for all cases.

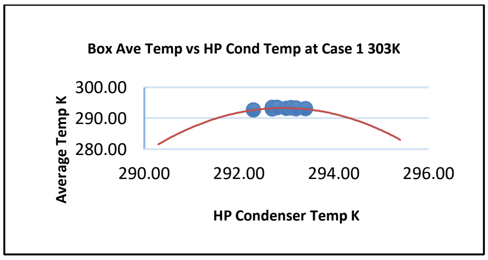


Figure 6: The box vs condenser side temperature for Case 1, 303 K.

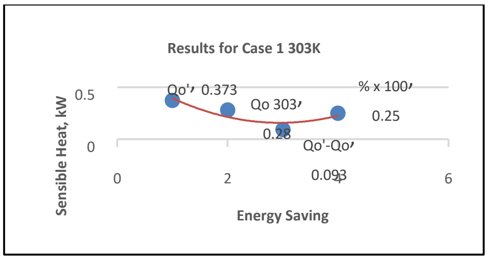


Figure 7: Energy saving at 303K for Case 1.

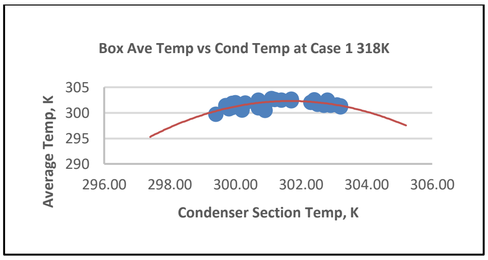


Figure 8: The box vs condenser side temperature for Case 1, 318 K.

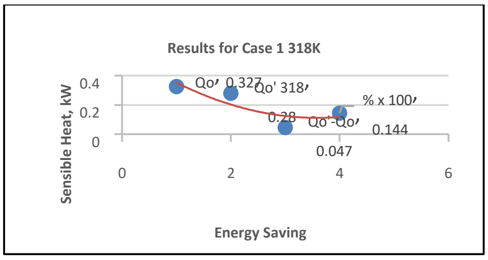


Figure 9: Energy saving at 318K for Case 1.

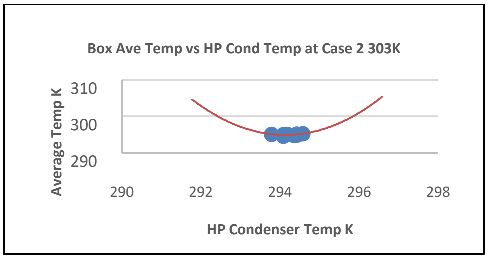


Figure 10: The box vs condenser side temperature for Case 2, 303 K.

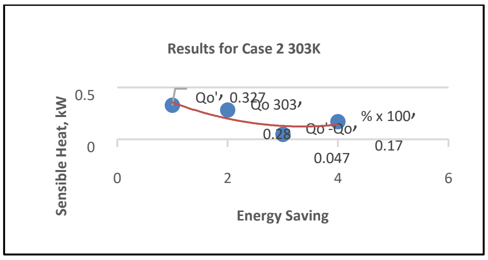


Figure 11: Energy saving at 303K for Case 2.

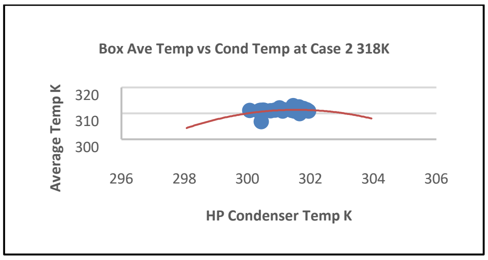


Figure 12: The box vs condenser side temperature for Case 3, 303 K.

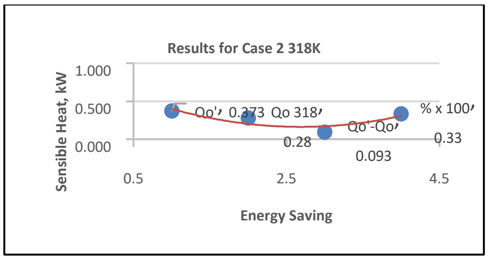


Figure 13: Energy saving at 318K for Case 2.

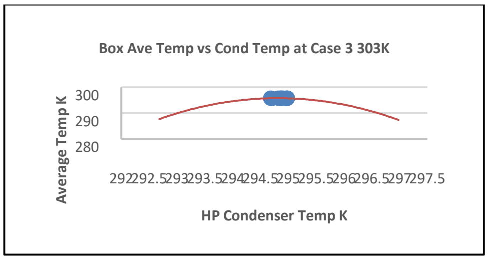


Figure 14: The box vs condenser side temperature for Case 3, 303 K.

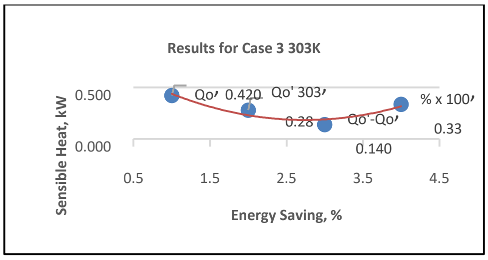


Figure 15: Energy saving at 303K for Case 3.

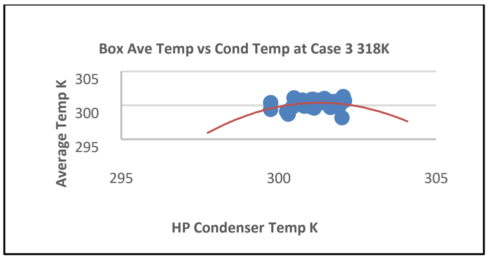


Figure 16: The box vs condenser side temperature for Case 3, 318 K.

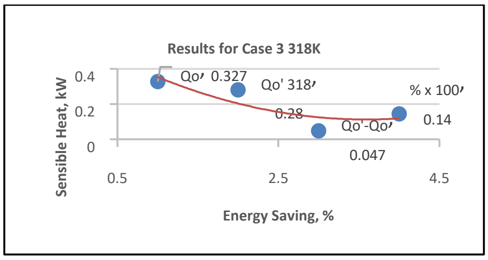


Figure 17: Energy saving at 318K for Case 3.

Conclusions

Simulations using computational fluid dynamics and experimental study on energy-saving of a heat-pipe heat exchanger ability are presented in this paper. The results show that the heat-pipe heat exchanger is capable to transfer 3 to 5 K differential temperature from end-to-end and achieved a 9K different of air inlet-outlet. The results found that the designed fins-type heat-pipe heat exchanger is capable to save up to 33% energy consumption. Figure 10 shows that between 327W and 420W of sensible heat gain had been determined and an energy saving between 93W to 140W achieved from the studies. For an equipment that did not require any energy sources, heat-pipe heat exchanger performed promising tools for an energy-saving application.

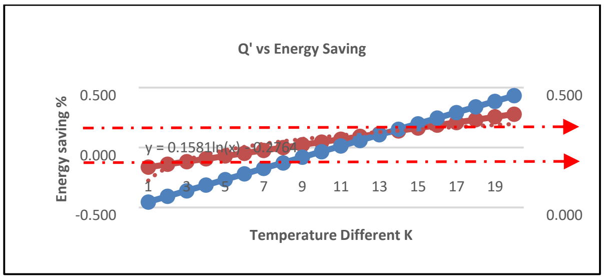


Figure 10: The sensible gain is between 327W to 420W and the energy saving is about 33%.

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