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The Utilization of Heat Exchangers for Energy Conservation in Air Conditioning

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ABSTRACT

This paper investigates the characteristics of heat exchanger (HPHE) as an efficient coolness recovery unit in air conditioning through experimental studies. It was conducted under a multiple-nozzle code tester based on the ASHRAE standards. The wind tunnel was subjected to airflow with considerable variation in its inlet air temperature. Among the factors being investigated are the air velocity, inlet and outlet air temperatures, overall efficiency and the number of rows in longitudinal direction. The data obtained were compared with the results predicted by previous theoretical studies. Good agreement was observed.

Keywords: Utilization, heat exchangers, multiple-nozzle code tester

INTRODUCTION

A heat is a device that allows very high rates of heat transfer over medium distances with low temperature differences. A heat pipe may have effective conductivity up to several hundred times that of an equivalent solid copper bar. It requires no external driving force other than a small temperature difference. It is lightweight and responds quickly to changes in heat load. A heat pipe heat exchanger composed of an array of individual heat pipes installed in a metal casing as illustrated in Fig.1.

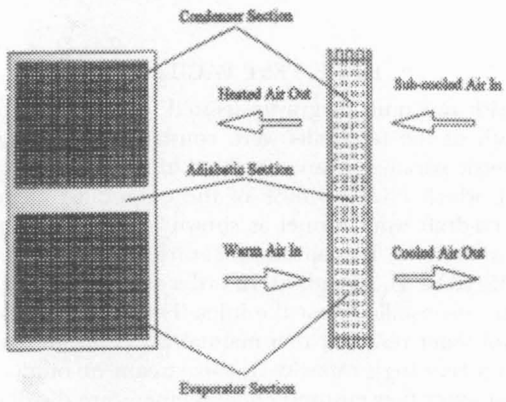


Fig. 1. Heat pipe heat exchanger

The special features of heat pipe have made them attractive for use as heat exchangers, especially in the recovery of waste energy. HPHE may also served for energy savings in commercial air conditioning system under the following forms:

(i) Recovering energy from the cold exhaust air to precool the incoming fresh air.
(ii) Provide precooling down to 50% RH and reheat to 75°F leading to smaller refrigeration plants.

This study would lead to smaller size refrigeration units in providing a more comfortable and healthy environment. Various attempts to predict the performance of heat pipe heat exchangers have been proposed. Attempts to predict the performance of heat pipe heat exchangers using a conductance model were made by Amode and Feldman, and Lee and Bedrossian.

The effectiveness-NTU method had been widely used to predict the performance of HPHE. Expressions of effectiveness for single heat pipe, and for n-rows of a HPHE were reported by Krishman and Rao, and Chaudrone. However, most of the work published earlier concentrated on theoretical studies and lacked of experimental data.

Wadowski *et al.* indicated that a minimum temperature difference of 5^{0}C between the two air streams was required to initiate operation of a six-row heat exchanger at 0.17kg/s. When full operating power is reached, the effectiveness became independent of the temperature difference of the two air streams. Effectiveness of the heat exchanger installed in the stream of moist exhausted air could be estimated on the basis of calculations of an ideal effectiveness for a particular condition. It had been found that the performance of six row heat exchanger, with equal supply and exhaust air stream mass flow rates, did not improve with change in air stream density and with condensation.

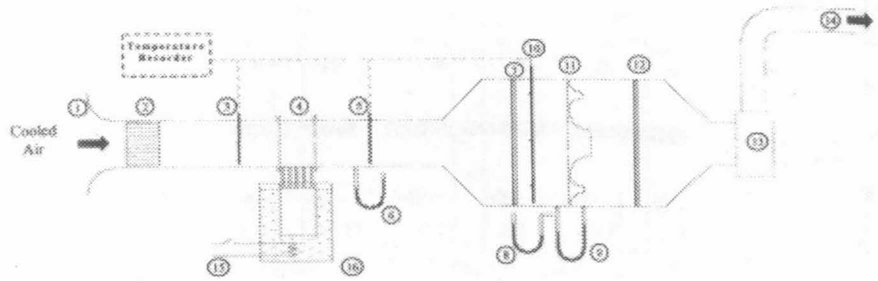
Good agreement of experimental and simulation results of a seven-row (HPHE), consisting of 3 pipes per row with pipe ID of 21mm, proved that the computation scheme using heat balance in each flow can be conveniently used for thermal calculations of HPHE of various configurations involving different number of rows of tubes and diffrence types of the outer transfer surfaces.

Azad *et al.* presented a theoretical analysis to predict the performances of a HPHE with 16.38 x 10^{-3} m tube diameter, fin density 275.59 fins per meter, 6.68 x 10^{-3} m flow passage hydraulic diameter and 2.54 x 10^{-4} m fin thickness. He showed that for HPHE with six and ten rows and for $C_e>C_c$ or $C_e>C_c$, the overall effectiveness increased with row and at $C_e=C_c$, the overall effectiveness was minimum. Also, the overall effectiveness was minimum. Also, the overall effectiveness increased with the number of fins perunit length.

HPHE TEST FACILITY

In this study, tests with two units of gravity-assisted wickless heat pipe heat exchanger were conducted. Both of the test units were constructed in staggered configuration. Their detailed geometric parameters are tabulated in Table 1. Refrigerant 134a was used as the working fluid, which filled up 60% of the evaporator section. Experiment was conducted in a forced-draft wind tunnel as shown in Fig. 2, equipped with a 1.1 kW centrifugal fan and an inverter. This air-flow measuring station is a multiple-nozzle code tester based on the ASHRAE 41.2 standard. In order to obtain good laminar flow, a wire-mesh air straightener was installed near the inlet. The evaporator section of HPHE was submerged into a hot water reservoir that maintains a constant temperature. This heat bath is analogous to a very high capacity hot-air stream or outdoor fresh air. A good internal circulation of water flow ensured equal temperature distribution in the bath, as well as continuous flow across the evaporator section. The wind tunnel's inlet is connected to an air-conditioning unit that controlled the inlet is connected to an analogous to the exhaust air from a refrigerated space passes through the condenser section of HPHE and is heated by heat transferred from the evaporator section.

The inlet and outlet temperatures across the test units are measured by two K-type thermocouple meshes. Each of this mesh consists of 9 thermocouple, having an accuracy of approximately 0.5°C. The pressure drop across the test units and the nozzle section



1. Inlet
2. Flow straightener
3. T/C inlet temperature measuring station
4. Test unit (HPHE)
5. T/C outlet temperature measuring station
6. Out pressure tap
7. Settling means
8. Nozzle pressure tap (inlet)
9. Nozzle pressure tap (outlet)
10. T/C inlet temperature measuring station (nozzle)
11. Multi nozzles plate
12. Settling means
13. Variable exhaust fan system
14. Discharge
15. Heater 4.5 kW
16. Heat bath with stirrer

Fg. 2. Schematic diagram of the experiment setup

was measured by a vernier scale manometer, reading up to 0.5 Pa. The air in each heat pipe of HPHE was evacuated by a vacuum pump to a pressure of 10^{-6} bar prior to charging. Two parameter variations were used to investigate different operating conditions during the experiments:

- Air flow rate was varied between 1.0 m/s and 3.0 m/s
- The cooled inlet air temperature was varied from 7^0C to 20^0C

Temperatures and operating pressure was recorded only when steady state was achieved. Both 4 dan 8 rows HPHE were tested under the same experimental conditions.

TABLE 1
Dimension of the HPHE tested

	4 Rows	8 Rows
Outer Tube Diameter (mm)	9.55	9.55
Inner Tube Diameter (mm)	7.55	7.55
Transverse tube spacing (mm)	25.4	25.4
Longitudinal tube spacing (mm)	22.0	22.0
Fin thickness (mm)	0.33	0.33
Fin density (fins per meter)	315	315
Face dimension (Evaporator and Condenser) (mm^2)	200 x 200	200 x 200
Tube material	Copper (pure)	(Copper (pure)
Fin material	Aluminum 2024	Aluminum 2024

Consider a HPHE with n-rows of heat pipes in the direction of flow as shown in Fig. 3.

The evaporator effectiveness for the jth row, $(e_e)_j$, may be obtained as follows:
For $C_e > C_c$

$$(\varepsilon_e)_j = \frac{(T_{e,in})_j - (T_{e,in})_{j+1}}{(T_{e,in})_j - (T_s)_j} \tag{1}$$

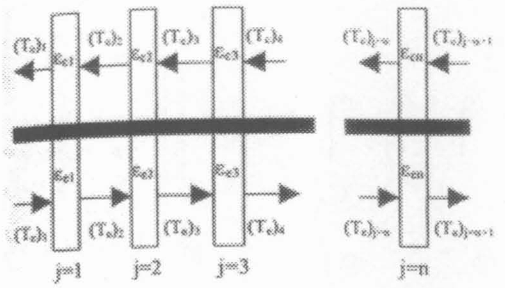


Fig. 3. A n-row Heat Pipe Heat Exchanger

Submersion of evaporator section into the water bath resulting a higher overall coefficient of heat transfer for the evaporator, U_e, as compared to the condenser, U_c. In addition, the specific heat, C_p, and density, p, of water is much greater than air, thus causing the evaporator effectiveness $(\varepsilon_e)_j$ to close to unity. As a result, the saturation temperature, T_s, is almost equivalent to the water bath temperature, T_w.

The condenser effectiveness for the jth row, $(\varepsilon_c)_j$ and the overall effectiveness of the HPHE for the jth row, $(\varepsilon_o)_j$, are respectively given by:

$$(\varepsilon_c)_j = \frac{(T_{c,in})_j - (T_{c,in})_{j+1}}{(T_s)_j - (T_{c,in})_{j+1}} \tag{2}$$

$$(\varepsilon_o)_j = \frac{1}{\frac{1}{\varepsilon_{cn}} + \frac{C_c / C_e}{\varepsilon_{en}}} \tag{3}$$

Since $C_c/C_e \approx 0$, it can be assumed that the overall effectiveness of the HPHE, $(\varepsilon_o)_j$, is identical to the condenser effectiveness, ε_c.

The temperature $(T_e)_j$, $(T_c)_j$, and $(T_s)_j$ may be calculated as follows:

$$(T_o)_{j+1} = \left(\varepsilon_0 \frac{C_c}{C_e}\right)(T_c)_{j+1} + \left(1 - \varepsilon_0 \frac{C_c}{C_e}\right)(T_e)_j \tag{4}$$

$$(T_c)_j = \left(1 - \varepsilon_c + \varepsilon_0 \frac{C_c \varepsilon_c}{C_e \varepsilon_e}\right)(T_c)_{j+1} + \left(\varepsilon_c + \varepsilon_0 \frac{C_c \varepsilon_c}{C_e \varepsilon_e}\right)(T_e)_j \tag{5}$$

$$(T_s)_j = \left(\frac{C_c \varepsilon_0}{C_e \varepsilon_c}\right)(T_c)_{j+1} + \left(1 - \frac{C_c \varepsilon_0}{C_e \varepsilon_c}\right)(T_e)_j \tag{6}$$

RESULTS AND DISCUSSION

Fig. 4 and 5 show the condenser outlet air temperature distribution in reponse to different inlet air temperature for both 4 and 8 rows HPHE.

In general, the test unit outlet air temperature increased with the inlet temperature for both 4 and 8-row HPHE. However, the magnitude of increment decreases with the inlet temperature. A considerable temperature difference between the condenser and evaporator section of the HPHE is essential in initiating the refrigerant boiling process.

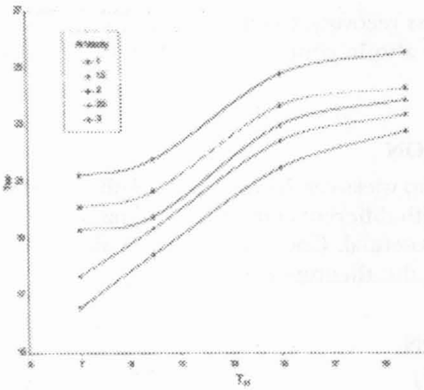


Fig. 4. Condenser outlet air temperature distribution vs inlet air temperature for 4-row HPHE

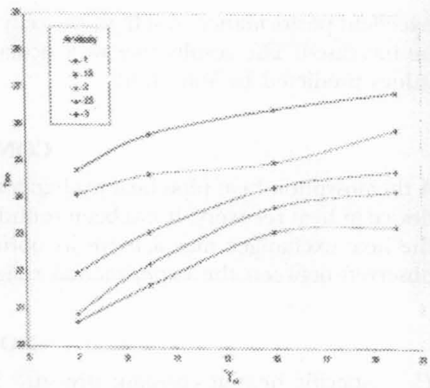


Fig. 5. Condenser outlet air temperature distribution vs inlet air temperature for 8-row HPHE

Higher inlet air temperature resulted in lower temperature differences and, therefore, reducing the outlet air temperature as well as the heat transfer rate.

The effect of air velocities are such that the outlet air temperature experienced a descending profile as air velocity increased. Lower air velocity means more time is taken in flowing across the HPHE, thus allows the air to be higher temperature.

The outlet air temperatures taken from the experiment was compared with the theoretical values calculated from equation (5). The results are shown in Fig. 6. Note that good agreement is obtained in the lower region of the 4-row HPHE.

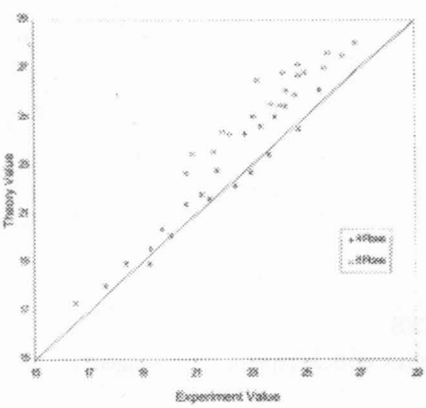


Fig. 6. Comparison of the theoretical and experimental temperature distribution

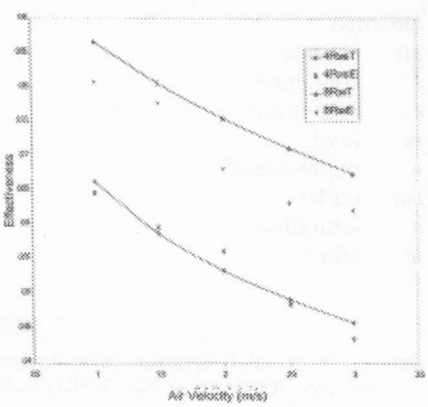


Fig. 7. Comparison of the overall effectiveness

Equations 2 and 3 give the condenser effectiveness and consequently the overall effectiveness of the HPHE. Fig. 7 presents the comparison of the overall effectiveness. Generally, e_o increases about 0.2 when the number of rows is doubled. However, the percentage of increment is much more significant, which varies from 25% to 45% as the air velocity changes between 1.0 m/s and 3.0 m/s respectively. This means that to obtain excellent performance in a high capacity coolness recovery, the number off rows should be increased. The results that we calculated are also in conformity with the theoretical values predicted by Yau et. al.

CONCLUSION

A themosyphon heat pipe heat exchanger, with no wicks can be used as a highly efficient device in heat recovery. It has been found that with different combination of parameters, the heat exchanger may achieve its optimum potential. Good agreement is also being observed between the experimental results and the theoretical values.

NOTATION

C_p	specific heat at constant pressure [$Jkg^{-1}K^{-1}$]
C_c	flow-stream capacity of cold-side fluid [WK^{-1}]
C_e	flow-stream capacity of hot-side fluid [WK^{-1}]
m	air mass flow rate [kgs^{-1}]
n	number of rows in flow direction [dimensionless]
NTU	number of heat-transfer units of an exchanger [dimensionless]
Q	heat transfer rate [W]
T	temperature [°C]
U	overall heat transfer coefficient [$Wm^{-2}K^{-1}$]
V	face velocity [ms^{-1}]

Greek letter

ε	exchanger effectiveness
ρ	density [kgm^{-3}]

Subscripts

air	air side
c	condenser
e	evaporator
in	inlet
o	outlet overall
out	outlet
s	saturation
w	water

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