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A method of analysis for heat pipe heat exchangers

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Abstract—A method of analysis for the thermal performance of heat pipe heat exchangers based on the conductance model was developed in the present study. In the analysis the specific heat conductance of the heat pipe was obtained from a performance test of a single heat pipe described in the present paper and the well-known universal correlations were used to calculate the convective heat transfer coefficients in tube banks. A computer program based on the finite difference equations of the model was then developed to calculate the thermal performance of heat pipe heat exchanger. The analysis was finally validated by an experiment and shown to be applicable in engineering applications.

INTRODUCTION

HEAT EXCHANGERS made of heat pipes have now become one of the most effective and economic devices for the recovery of waste heat energy. Although the thermal performance characteristics of a single heat pipe have been extensively studied and clearly understood during the past twenty years, the study of the prediction of the overall performance of heat exchangers using a bank of heat pipes as the heat transfer elements appears to be very limited. A typical theoretical study but without experimental verifications on the performance calculation of heat pipe heat exchanger was recently carried out by Amode and Feldman [1]. In this study, the basic theories for heat pipes and conventional heat exchangers were employed to perform a pure theoretical calculation.

More recently, Lee and Bedrossian [2] developed a simple analytical model to study the characteristics of heat exchangers using heat pipes or thermosyphons. In this model, the thermal resistances inside the heat pipes or thermosyphons were theoretically calculated by the well-known correlations for the boiling and condensing processes. However, the two constants in the empirical correlation used to calculate the convective heat transfer coefficients outside the heat pipes or thermosyphons had to be determined by a curve fitting to the experimental results of an actual heat exchanger. It is well known that the universal empirical correlation for convective heat transfer in tube banks is of the form

$$Nu = C\, Re^{n}\, Pr^{1/3} \tag{1}$$

where the universal constants, C and n, depend on the geometric arrangement of the tube bank and have been universally tabulated according to a great deal of experimental results [3]. For the thermosyphon heat exchanger used by Lee and Bedrossian, C and n should be 0.415 and 0.581, respectively. The curve fitting by the experiment of a thermosyphon heat exchanger in [2], however, gave 0.009 and 1.050 for C and n. Such a large discrepancy could be attributed to two facts. First, the actual boiling and condensing processes in the thermosyphons were not exactly identical with those described by the empirical correlations and the fouling inside the thermosyphons was not taken into account in Lee's model. Second, there was insufficient quality control in making the heat pipes or thermosyphons so that the thermal performance of each one was different.

To develop a method for the prediction of overall thermal performance of a heat pipe heat exchanger, a modified method of analysis is therefore proposed in the present study. In this method, the thermal performance of a single heat pipe or thermosyphon is experimentally, instead of theoretically, determined by a testing method. With the testing results and the universal empirical correlation of convective heat transfer for tube banks, the overall thermal performance of the heat exchanger can be calculated by use of the proposed method of analysis.

PERFORMANCE TEST OF A SINGLE HEAT PIPE

Heat pipes without fins and using water as the working fluid were carefully made in the laboratory for the present study. The specifications of the heat pipes are listed in Table 1. Since the present experiment was carried out in a short time period (within a month) as compared to commercial operation (usually for several years) and the major concern of this study was on the thermal performance analysis, the deterioration in thermal performance due to the generation of non-condensible gases caused by the use of a carbon steel–water combination was minimal and could be neglected.

The thermal performance of a heat pipe was measured by the testing equipment as shown in Fig. 1. As the heat pipe was inserted into the water jacket, it was automatically divided into the evaporation and the condensation sections with equal lengths but with negligible adiabatic section. This design was adopted in order to meet the general requirement in some practical

NOMENCLATURE

A total area of heat pipe heat exchanger [m^2]
A_c surface area of condensing section of a heat pipe [m^2]
A_e surface area of evaporating section of a heat pipe [m^2]
C_c specific heat of cold air flow [KJ kg^{-1} °C^{-1}]
C_h specific heat of hot air flow [KJ kg^{-1} °C^{-1}]
D_p characteristic diameter of heat pipe heat exchanger [m]
D heat pipe O.D. [m]
k_f thermal conductivity of air [W m^{-1} °C^{-1}]
h convective heat transfer coefficient [W m^{-2} °C^{-1}]
M_c mass flowrate of cold air [kg s^{-1}]
M_h mass flowrate of hot air [kg s^{-1}]
n_j number of heat pipes in jth row
Nu Nusselt number in tube banks, defined in equation (11)
Pr Prandtl number
Q total energy transfer rate of heat exchanger [W]
q_p energy transfer rate of a heat pipe [W]
q_{pj} energy transfer rate in the jth row of heat pipes [W]
R_j overall heat transfer resistance in the jth section [°C W^{-1}]
Re Reynolds number in tube banks, defined in equation (11)
T temperatures [°C]
u characteristic velocity in tube banks [m s^{-1}].

Greek symbols
ρ density of air [kg m^{-3}]
μ viscosity of air [N s m^{-2}].

Subscripts
c condenser side; cold air
h evaporator side; hot air
w outside wall surface of heat pipe
j section notation of heat exchanger.

applications for which the adiabatic sections of the heat pipes in the heat pipe heat exchanger had to be as small as possible. The condensing section of the heat pipe was then sealed by use of a flange and o-rings in the water jacket through which a cooling water was allowed to pass. In the evaporating section of the heat pipe, a 2.5 kW electric heating wire was uniformly wrapped over the pipe surface as the heating element. The rate of heating was controlled by a power transformer. To eliminate the heat losses, the tested heat pipe including the water jacket was carefully insulated by 6 cm thick calcium silicate. Therefore, the overall energy transfer rate through the heat pipe can be measured either by measuring the rate of energy added by the electric heater or by measuring the rate of energy removed by the water jacket. These two measurements also provided a check for the insulation condition during the experiment. It was observed in the present experiments that the two values obtained from the above two measurements all coincided within $\pm 5\%$ deviations.

As the present method of testing for the thermal performance of a single heat pipe is non-destructive and

Table 1. Specifications of heat pipes

Effective length	610 mm
Outside diameter	33.7 mm
Wall thickness	1.6 mm
Wall material	carbon steel
Wick structure	wrapped screen (15 layers)
Wick material	100-mesh bronze
Working fluid	distilled water (115 g)

can be applied to commercial heat pipes, it was not attempted in the present study to measure the temperatures or other properties in the interior of the heat pipes. Instead, only temperatures over the outside surface of the heat pipe were measured using eleven T-type thermocouples which were fixed equally spaced onto the pipe surface and read by an Omega 2176A recorder to within ± 0.5°C. Temperature measurements were also made at the inlet and outlet of the water jacket. To determine the rate of energy removed in the water jacket, the mass flowrate of the cooling water was also measured by a rotameter which was calibrated to within $\pm 5\%$.

Figure 2 shows the temperature distributions vs energy transfer rates obtained from one of the test runs for a performance test of a single heat pipe. At this stage, it is very important to derive a method to correlate the testing results. As a first-order approximation, it is assumed that the longitudinal heat conduction in pipe wall and wick material is negligible. Thus, there are five thermal resistances for the transfer of energy from the hot to the cold side of the heat pipe as shown in Fig. 3, where $R_{p,e}$ and $R_{p,c}$ are the thermal resistances of the pipe wall at the evaporating and the condensing sections, respectively; $R_{w,e}$ and $R_{w,c}$ are the thermal resistances in the wick materials at the evaporating and the condensing sections, respectively and R_v is the overall thermal resistance in the vapor core region including interphase condensing and evaporating resistances.

For heat pipes operating below the sonic limit, the thermal resistance in the vapor core, R_v, can be ignored

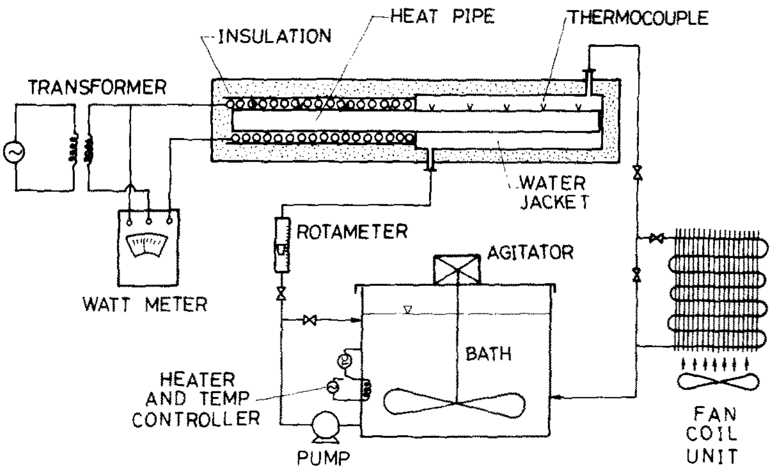

Fig. 1. Testing equipment of a single heat pipe.

in comparison with the others. Therefore it can be theoretically shown in accordance with the heat pipe theory [4] that the vapor core temperature, T_v, can be expressed in terms of the lengths and the mean surface temperatures of the condensing and the evaporating sections

$$T_v = \frac{T_{wh}L_e + T_{wc}L_c}{L_e + L_c} \tag{2}$$

where T_{wc} and T_{wh} are the mean temperatures on the outside surface of the condensing and the evaporating sections, respectively, and L_c and L_e are the lengths of condensing and evaporating sections. If L_c and L_e are set equal as in the present experiment, the vapor core temperature is just the arithmetic mean of T_{wh} and T_{wc}. Furthermore, the overall energy transfer rate in a heat pipe, q_p, can be expressed as

$$q_p = (UA)_p(T_{wh} - T_{wc}) \tag{3}$$

where $(UA)_p$ is the specific heat conductance from the surface of the evaporating section to that of the condensing section. Therefore, the specific conductance of a heat pipe can be experimentally determined according to the definition shown in equation (3).

Since the temperature distributions on the surfaces of condensing and evaporating sections are generally very uniform, the mean temperatures on the outside surfaces of the condensing and evaporating sections, T_{wc} and T_{wh}, can be experimentally determined by averaging the thermocouple readings on them. However the readings for the positions at which the axial wall heat conduction

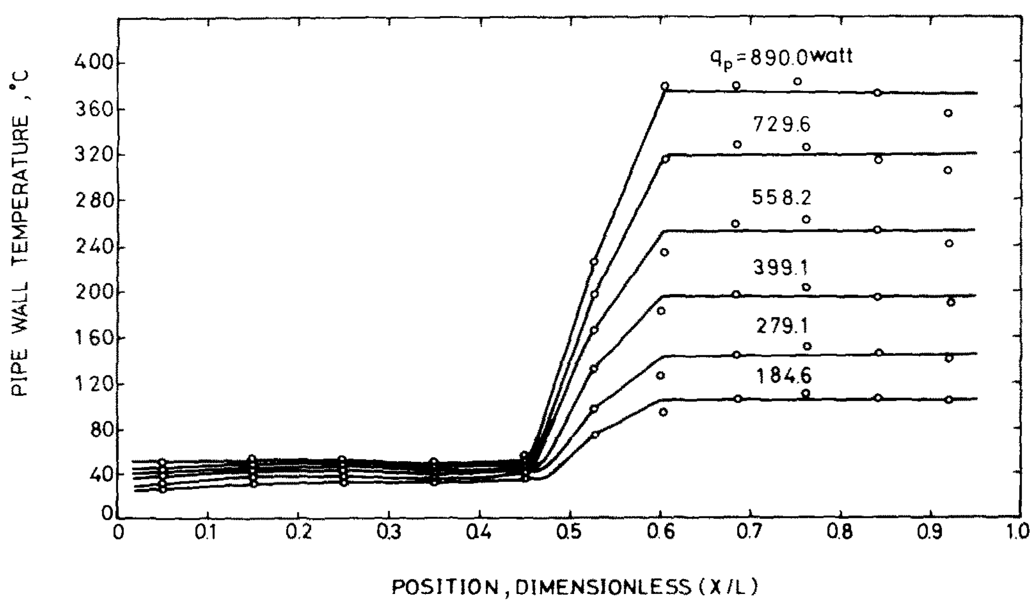

Fig. 2. Wall temperature distributions vs energy transfer rates.

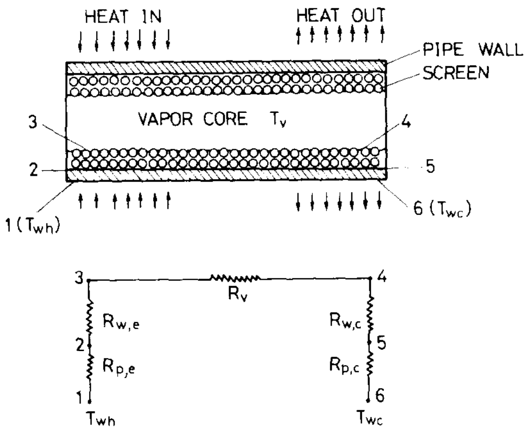

FIG. 3. Thermal resistances network of a heat pipe.

toward the condensing section is not negligible cannot be included. Therefore, care should be taken in averaging the thermocouple readings on the pipe surface to preclude these points.

For the heat pipes made in the present study, the largest deviation of the surface temperature in the evaporating section, precluding the point nearest to the pipe mid-point, was within $\pm 12°C$ for temperatures around 360°C and even much less for temperatures below this, as can be seen from Fig. 2. Thus the averaged surface temperatures could give reasonably good approximations.

In addition, poor quality control in making the heat pipes can lead to small deviations in heat conductance between each heat pipe. This necessitates a certain amount of testing of samples to determine a mean value for the heat conductance which can statistically represent the averaged thermal performance of the batch of heat pipes to be used to make a heat pipe heat exchanger.

In the present study, five out of 32 heat pipes made in the laboratory were randomly selected to be separately tested according to the above testing method. It can be seen from Fig. 4 that the specific conductance of each heat pipe is approximately independent of the vapor core temperature as can be expected from the heat pipe theory. Therefore, combining all the data, the mean heat conductance for this batch of heat pipes was determined as 3.36 W $°C^{-1}$ with a standard deviation of ± 0.59 W $°C^{-1}$. The standard deviation will apparently decrease with improving quality control. When the quality control is excellent, as in commercial mass production, this sampling test can be omitted and a single performance test for a heat pipe of the same kind using the same procedure can directly give the heat conductance.

It also can be seen that the heat transfer behavior of the heat pipes made in the present study is not good. This is attributed to the fact that no other means which can reduce the thermal resistance of the wick material were employed in the present heat pipes. In addition, a tight contact between the wrapped bronze screen and the pipe wall was produced by the elastic force induced merely by the screen itself when carefully pressing the wrapped screen into the pipe. Since the major purpose of the present study is to develop a new method for the thermal performance analysis of heat pipe heat exchanger, the study on the fabricating techniques which can produce highest efficiency heat pipes was skipped over.

THERMAL ANALYSIS OF HEAT PIPE HEAT EXCHANGER

A finite difference method based on the conductance model is employed in the present study to analyze the overall heat transfer of the heat pipe heat exchanger. Assume that the heat pipe heat exchanger can be

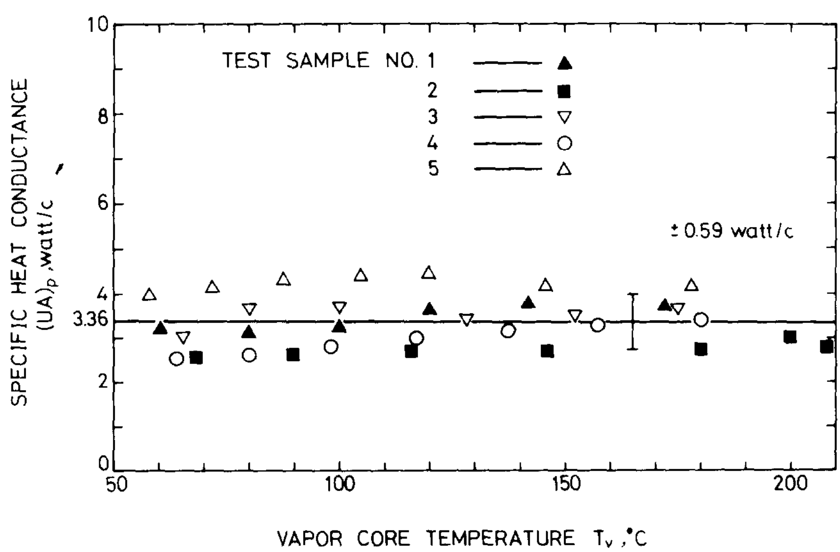

FIG. 4. Thermal performance test results of heat pipes.

divided into N sections along the flow direction and each section consists of a row of heat pipes of the same kind, as shown in Fig. 5. Neglecting the heat transfer across the partition and the heat loss to the ambient, the total energy transfer in the jth section can be expressed as

$$q_{pj} = n_j(1/R_j)(\bar{T}_{hj} - \bar{T}_{cj}) \tag{4}$$

where n_j is the number of heat pipes in the jth section, $\bar{T}_{hj}$ and $\bar{T}_{cj}$ are the mean fluid temperatures of the hot and the cold flows, respectively, in the jth section, which is defined as

$$\bar{T}_{hj} = (T_{h,j} + T_{h,j-1})/2 \quad \text{and} \quad \bar{T}_{cj} = (T_{c,j} + T_{c,j-1})/2.$$

R_j in equation (4) represents the overall heat transfer resistance from the hot flow to the cold flow for a heat pipe in the jth section and can be expressed by the following equation, ignoring fouling resistance in the pipe wall

$$R_j = (1/h_e A_e)_j + (1/UA)_p + (1/h_c A_c)_j \tag{5}$$

where h_e and h_c are the convective heat transfer coefficients outside the heat pipe wall in the hot and the cold flow, respectively; A_e and A_c are the outside surface areas of a heat pipe in the evaporating and the condensing sections. Therefore the overall energy transfer rate delivered by the whole heat pipe heat exchanger can be calculated by an integration over the N sections

$$Q = \sum_{j=1}^{N} q_{pj}. \tag{6}$$

It is necessary to calculate the temperature distributions in the hot and cold flows before calculating the overall energy transfer rate using equations (4) through (6). By applying the energy balance to the jth section, the temperature changes of the flows across each section can be written as

$$T_{c,j} - T_{c,j-1} = n_j(1/R_j)(\bar{T}_{hj} - \bar{T}_{cj})/M_c C_c \tag{7}$$

$$T_{h,j} - T_{h,j-1} = -n_j(1/R_j)(\bar{T}_{hj} - \bar{T}_{cj})/M_h C_h \tag{8}$$

where M_h and M_c are the mass flowrates of the hot and the cold flow, respectively, both are taken as positive for parallel-flow design and M_c is taken as negative for counter-flow design. To solve the temperature distributions, the iteration method with relaxation factor is employed [5] with the following equations which are derived from equations (7) and (8)

$$T^{n}_{h,j} = \frac{1}{1+K_{hj}} T^{o}_{h,j-1} - \frac{K_{hj}}{1+K_{hj}} \times [T^{o}_{h,j-1} - T^{o}_{c,j} - T^{o}_{c,j-1}] \tag{9}$$

$$T^{n}_{c,j} = \frac{1}{1+K_{cj}} T^{o}_{c,j-1} - \frac{K_{cj}}{1+K_{cj}} [T^{o}_{c,j-1} - T^{o}_{h,j} - T^{o}_{h,j-1}] \tag{10}$$

where the superscripts, n and o, represent the new and the old value, respectively, during the iteration and

$$K_{hj} = n_j/2M_h C_h R_j \quad \text{and} \quad K_{cj} = n_j/2M_c C_c R_j.$$

The iteration procedure is as shown in Fig. 6. To calculate the convective heat transfer coefficients in tube banks, h_e and h_c, the universal correlation obtained by Whitaker [6] is used

$$Nu = f(0.5Re^{1/2} + 0.2Re^{2/3})Pr^{1/3}(\mu/\mu_w) \tag{11}$$

where Nu and Re are defined as

$$Nu = (hD_p/k_f)\varepsilon_v/(1-\varepsilon_v) \tag{12}$$

and

$$Re = (\rho D_p u/\mu)\varepsilon_v/(1-\varepsilon_v). \tag{13}$$

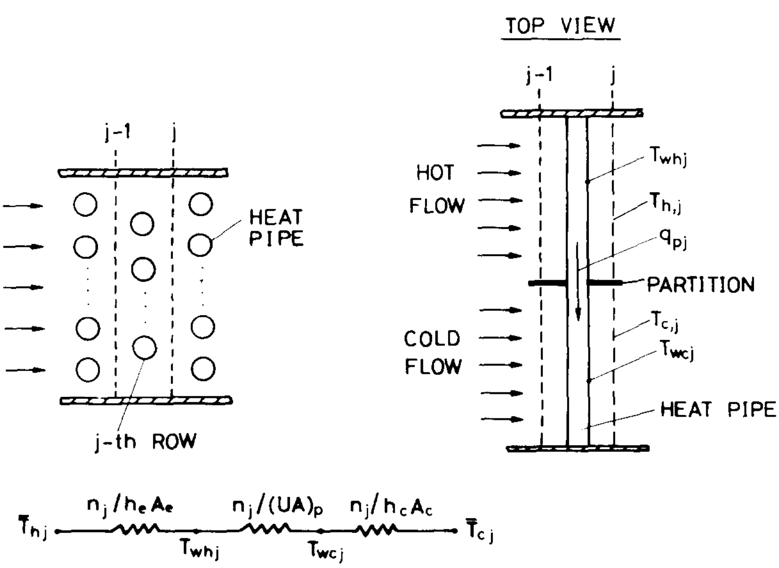

FIG. 5. Schematic diagram of heat pipe heat exchanger.

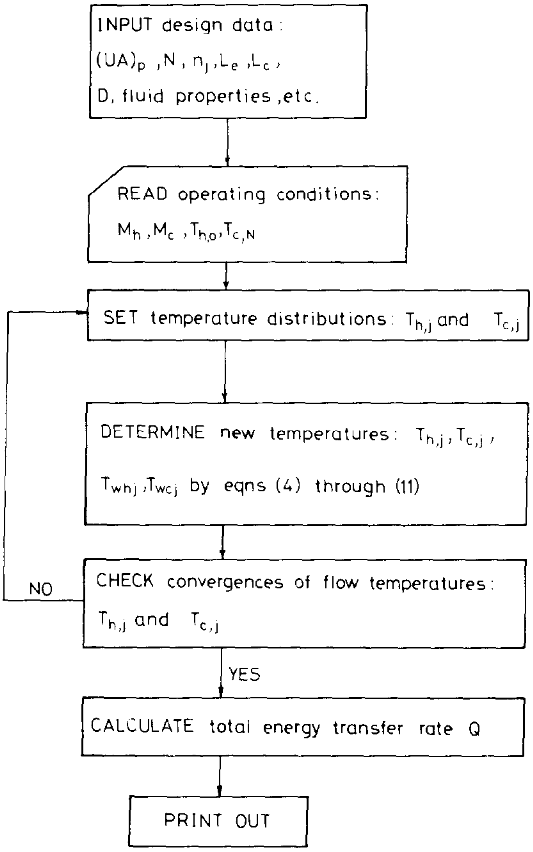

FIG. 6. Flow chart of the iteration process in thermal analysis.

D_p is the characteristic diameter defined as $1.5D$ (D is the pipe O.D.) for tube banks and ε_v is the void fraction of the heat pipe heat exchanger, u is the characteristic flow velocity which is usually taken as the maximum velocity in the tube bank and f is a parameter depending on the number of rows in the heat pipe heat exchanger [7].

A computer program based on the above equations and the iteration process as shown in Fig. 6 was developed in the present study to calculate the thermal performance of a heat pipe heat exchanger.

EXPERIMENTS

An experiment for a heat pipe heat exchanger was performed in the present investigation to verify the above analysis. The schematic diagram of the experimental setup is as shown in Fig. 7. The specifications of the heat pipe exchanger are listed in Table 2.

The heat pipe heat exchanger was designed to be operated in counter-flow conditions with air as the energy exchange fluids in the present study. To provide a hot air flow with temperatures up to 300°C for the experiment, a 30 kW electric heater was installed in the rectangular hot air duct. The air flowrates in both hot and cold air ducts were measured in two 100-mm circular portions of the air ducts in which Pitot tubes

Table 2. Specifications of heat pipe heat exchanger

Number of rows, N	4
Number of heat pipes in each row, n_j	8
Total number of heat pipes	32
Condenser length of heat pipes, L_c	305 mm
Evaporator length of heat pipes, L_e	305 mm
Pipe arrangement	staggered
Pipe pitch : longitudinal	44.7 mm
transverse	38.7 mm
Cross-section : for hot flow	$305 \times 380\ \mathrm{mm}^2$
for cold flow	$305 \times 380\ \mathrm{mm}^2$

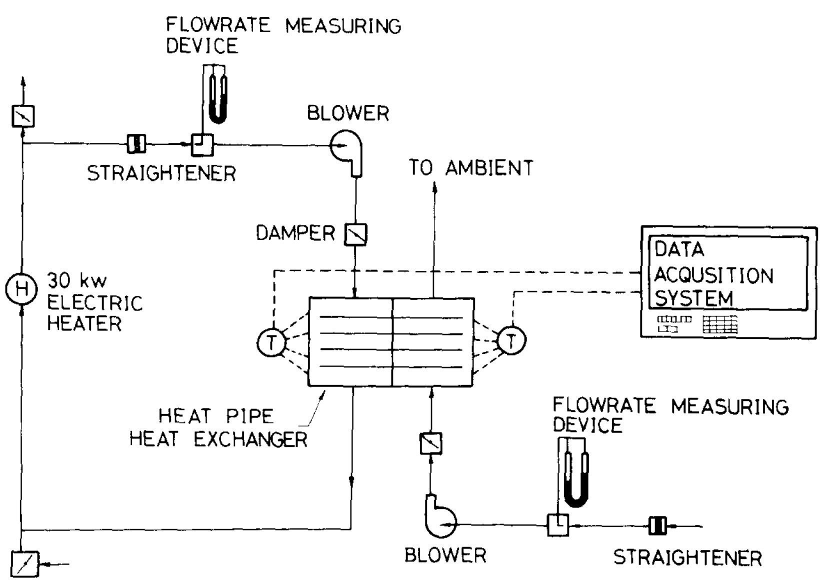

FIG. 7. Testing facilities of heat pipe heat exchanger.

Table 3. Comparison of experiment with theory for total energy transfer rate of the heat pipe heat exchanger

	Hot air		Cold air		Energy transfer rate, Q (W)	
Test run No.	Flowrate (SCMM)	Inlet temp., $T_{h,0}$ (°C)	Flowrate (SCMM)	Inlet temp., $T_{c,4}$ (°C)	Experiment	Theory
1	12.9	244.1	13.2	25.1	6290	5920
2	12.8	240.0	13.0	25.2	6050	5770
3	12.6	238.4	13.2	24.6	5870	5750
4	12.2	276.2	12.7	24.8	6750	6520
5	9.10	297.1	10.4	28.8	5710	6020
6	9.10	295.7	10.3	28.7	5640	5990
7	12.3	274.4	7.20	32.0	5660	5480
8	11.8	296.6	6.80	34.7	5720	5710
9	11.5	261.6	7.00	31.7	4720	5080
10	11.3	298.5	7.20	33.0	5700	5820
11	8.20	299.0	7.10	31.3	4880	5320
12	8.20	300.7	7.10	31.4	4940	5350

and precision pressure gauges were used to measure the velocity distribution at given cross sections. The velocity was measured at 10 points along the radial direction in the present experiments to obtain the velocity distribution. By integrating the velocity distribution then multiplying by the air density at the same cross-section, the air mass flowrate in each duct was determined. The experimental uncertainties were estimated to be within $\pm 5\%$.

Temperature measurements were made by T-type thermocouples at the inlet and the oulet of the hot and the cold air flows and all over the heat pipe heat exchanger. The mean outside wall temperatures of the condensing and the evaporating sections of the heat pipes at the jth row, T_{wcj} and T_{whj}, were measured by 16 thermocouples. To do this, two pipes were selected at each row and four thermocouples were separately fixed at the middle points of the condensing and the evaporating sections of the two heat pipes. The mean wall temperature of the condensing or evaporating section at the jth row, T_{wcj} or T_{whj}, then was obtained by averaging the two temperatures on the two heat pipes.

To measure the air temperatures between two adjacent rows of heat pipes in the hot or cold channel of the heat exchanger, $T_{h,j}$ or $T_{c,j}$, three thermocouples were installed at each measuring cross-section and the average readings gave the bulk air temperatures. There were five measuring cross-sections in each channel and 30 thermocouples were used in this measurement. Besides, two temperature measurements were also made at the velocity measuring ducts to obtain the air densities for determining the air mass flow rates.

To process the 38 thermocouple readings and the related calculations in the present experiment, a Kaye Digistrip II data acquisition and microcomputer system was used. The maximum uncertainties in measuring the total rates of energy transfer of the heat pipe heat exchanger were estimated to be within $\pm 5\%$.

RESULTS AND DISCUSSION

Some test runs using the above heat pipe heat exchanger equipment were carried out for various air flow rates and temperatures in the present experiment. Theoretical calculations based on the analytical model described previously were also carried out for comparisons. The results are listed in Table 3. It can be seen that the errors caused by the analysis in the total energy transfer rates are all within $\pm 10\%$ which is acceptable in engineering applications. It is also shown from Table 4 that the measured temperature distributions of the air flows in the heat exchanger are in fairly good agreement with the analytical results. Larger errors in the predictions of the pipe wall temperatures occur in the present experiment, as can be seen from Table 5. This probably resulted from the contact resistances between the pipe wall and the temperature probes which were introduced during the installation of the probes.

In fact, the convective heat transfer in tube banks increases along with the penetrating of the air flow into the tube banks due to the transition of the flow pattern from laminar to turbulent [8]. Therefore, the use of the mean convective heat transfer coefficients in tube banks in the present analysis could lead to the small errors in temperature predictions as shown in Tables 4 and 5.

From the experiment, we verified that the method of analysis for the thermal performance of heat pipe heat exchanger proposed in the present study is validated. Although the present experiment was performed using heat pipes without fins for the sake of simplicity, the present method of analysis still can be applied to heat pipes with fins as long as the convective heat transfer coefficients for finned-tube banks are used.

Finally, it should be noted that, to ensure good accuracy, the present method of analysis is not valid when the operating conditions of any heat pipe in the

Table 4. Air temperature distributions

Test run No.		Air temp. in hot flow, $T_{h,j}$ (°C) $j=0$	1	2	3	4	Air temp. in cold flow, $T_{c,j}$ (°C) $j=0$	1	2	3	4
1	Experiment	244.1	236.0	223.2	213.8	200.7	49.7	42.2	35.7	29.8	25.1
	Theory	244.1	233.5	223.1	213.0	203.2	48.3	42.2	36.4	30.7	25.1
2	Experiment	239.5	232.1	219.6	210.5	197.9	49.1	42.2	36.1	30.0	25.2
	Theory	239.5	229.1	219.1	209.3	199.7	48.1	42.1	36.3	30.7	25.2
3	Experiment	238.4	231.1	218.9	210.0	197.5	47.5	40.8	35.1	29.2	24.6
	Theory	238.4	228.0	217.8	207.9	198.2	47.1	41.2	35.5	30.0	24.6
4	Experiment	276.2	266.8	250.4	239.2	223.6	52.2	44.0	36.8	30.0	24.8
	Theory	276.2	262.8	249.8	237.3	225.2	51.4	44.4	37.7	31.1	24.8
5	Experiment	297.1	288.7	267.0	254.1	235.2	57.3	47.7	39.5	33.2	28.8
	Theory	297.1	279.6	262.9	246.9	231.6	59.0	50.9	43.2	35.9	28.8
6	Experiment	295.7	287.7	266.3	253.8	234.7	57.1	47.9	39.6	33.4	28.7
	Theory	295.7	278.4	261.8	245.9	230.8	59.0	50.9	43.2	35.8	28.7
7	Experiment	274.4	266.9	253.7	244.4	230.7	73.0	61.3	47.5	37.5	32.0
	Theory	274.4	263.6	253.0	242.5	232.0	71.8	61.7	51.7	41.8	32.0
8	Experiment	296.6	292.1	275.0	265.4	248.9	79.2	67.0	51.4	40.4	34.7
	Theory	296.6	284.5	272.5	260.6	248.9	79.2	67.9	56.7	45.6	34.7
9	Experiment	261.6	258.1	244.6	237.2	223.8	67.2	57.3	45.2	36.4	31.7
	Theory	261.6	251.2	241.0	230.9	220.8	70.0	60.3	50.7	41.2	31.7
10	Experiment	298.5	293.4	275.7	266.0	248.9	74.9	63.2	48.6	38.3	33.0
	Theory	298.5	285.5	272.8	260.2	247.8	75.8	64.9	54.1	43.5	33.0
11	Experiment	299.0	291.3	270.8	258.8	240.2	67.2	56.7	44.3	35.7	31.3
	Theory	299.0	282.1	265.8	250.0	234.8	70.5	60.1	50.2	40.6	31.3
12	Experiment	300.7	292.7	272.0	259.6	241.0	67.7	57.1	44.6	36.0	31.4
	Theory	300.7	283.6	267.2	251.3	235.9	70.7	60.4	50.4	40.7	31.4
Average error		0.0	6.2	2.5	4.8	1.2	−0.7	−1.6	−3.5	−3.1	0

heat exchanger are beyond the testing range in the performance test of a single heat pipe as described previously. Therefore, checks at any instant during the analysis are always required and it is important to perform the performance test for a single heat pipe over a wider range of operating conditions, as near the heat pipe operating limits as possible.

The present method of analysis can be applied to simulate the thermal performance of the heat pipe heat exchanger used in the experiment beyond the testing

Table 5. Pipe wall temperature distributions

Test run No.		Wall temp. in hot air side, T_{whj} (°C) $j=1$	2	3	4	Wall temp. in cold air side, T_{wcj} (°C) $j=1$	2	3	4
1	Experiment	169.2	160.3	144.6	140.5	100.2	92.6	83.2	80.0
	Theory	165.6	156.6	147.9	139.4	108.0	100.7	93.6	86.7
2	Experiment	167.2	159.3	143.8	140.0	97.8	90.8	83.0	80.0
	Theory	162.9	154.1	145.6	137.4	106.8	99.6	92.6	85.8
3	Experiment	173.8	160.6	144.4	140.8	95.2	88.0	79.6	78.9
	Theory	161.2	152.5	144.0	135.7	105.3	98.2	91.2	84.4
4	Experiment	194.0	180.4	160.6	156.8	110.4	99.7	88.1	87.0
	Theory	183.5	172.6	162.0	151.7	119.5	110.7	102.2	94.0
5	Experiment	208.3	201.6	176.3	170.3	113.8	101.8	88.0	83.0
	Theory	190.9	177.3	164.3	151.9	130.7	119.9	109.6	99.8
6	Experiment	207.8	201.4	176.0	170.2	113.2	101.0	87.3	82.4
	Theory	190.4	176.8	164.0	151.6	130.6	119.8	109.6	99.7
7	Experiment	208.1	198.8	179.4	176.6	139.5	130.9	117.0	116.7
	Theory	199.6	189.2	178.8	168.6	147.8	137.9	128.0	118.3
8	Experiment	223.5	216.9	193.4	190.8	150.4	141.8	126.0	125.8
	Theory	215.5	203.8	192.2	180.7	161.3	150.2	139.2	128.4
9	Experiment	202.0	196.2	176.2	174.4	126.6	118.4	108.3	108.4
	Theory	190.0	180.0	170.0	160.2	141.9	132.4	122.9	113.6
10	Experiment	223.0	216.7	192.8	189.8	145.2	134.9	120.3	120.2
	Theory	213.1	201.0	189.0	177.2	157.6	146.3	135.3	124.4
11	Experiment	214.7	208.4	183.2	178.8	132.6	121.5	107.0	105.3
	Theory	200.2	186.0	172.3	159.1	147.9	135.6	123.8	112.3
12	Experiment	215.8	209.2	183.8	179.4	133.6	122.4	107.6	105.8
	Theory	201.2	186.9	173.1	159.8	148.5	136.2	124.3	112.8
Average error		11.1	14.4	4.3	11.3	−12.3	−12.0	−14.7	−7.2

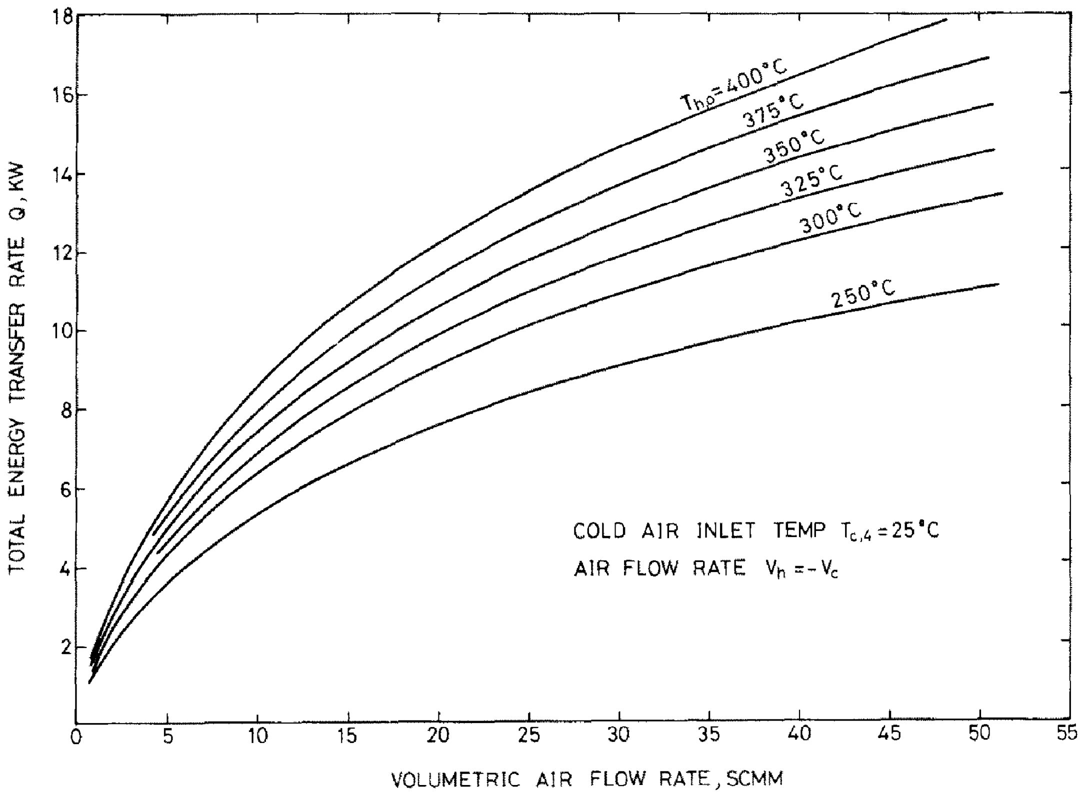

FIG. 8. Total energy transfer rate at various inlet temperatures of hot air.

conditions. Figure 8 shows the total energy transfer rates at various inlet temperatures of hot air, $T_{h,0}$, and equal volumetric air flow rates, $V_c = V_h$, in the hot and the cold flows. For different combinations of flowrates, the overall heat transfer coefficient of the heat pipe heat exchanger, U, can be determined in terms of the Reynolds number of the cold air flow, Re_c, and the ratio of the Reynolds numbers of the two air flows, r, as can be seen in Fig. 9. The Reynolds numbers, Re_c and Re_h, are defined as in equations (12) and (13) and the ratio, r, is defined as $r = Re_h/Re_c$. The overall heat transfer coefficient, U, is defined as

$$U = Q/A(\mathrm{LMTD}) \tag{14}$$

where (LMTD) is the log mean air temperature difference of the heat exchanger, A is the total heat transfer area of the heat exchanger. It can be seen from Figs 8 and 9 that the energy transfer rate as well as the

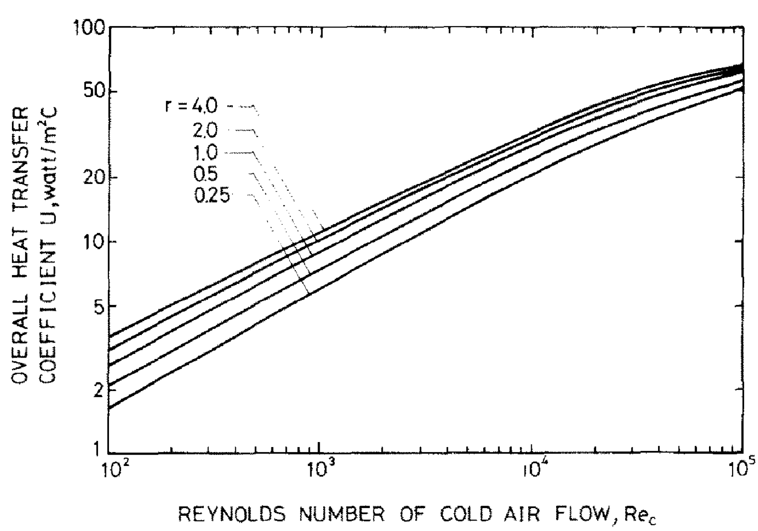

FIG. 9. Overall heat transfer coefficients of heat pipe heat exchanger at various operating conditions.

overall heat transfer coefficient increases with increasing flowrate but tends to reach a constant. This is due to the fact that the convective heat transfer resistances in tube banks decreases with increasing flow rate but the specific heat conductance of the heat pipes still remains nearly constant over a wide range of operating temperatures.

CONCLUSIONS

A method of analysis based on the conductance model was developed in the present study. In the analysis, the specific heat conductance of heat pipe was obtained from a thermal performance test for a single heat pipe and the universal correlations for the convective heat transfer coefficients in tube banks were used. A computer program using the finite difference equations of the model was also developed to calculate the thermal performance of heat pipe heat exchanger and the present method of analysis has been validated experimentally and shown to be applicable in engineering applications.

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UNE METHODE D'ANALYSE DES ECHANGEURS CALODUCS

Résumé—On développe une méthode d'analyse des performances thermiques des échangeurs–caloducs basée sur le modèle de conductance. Dans l'analyse, la conductance thermique spécifique du caloduc est obtenue à partir d'essais d'un caloduc unique décrit dans cet article et les formules universelles classiques sont utilisées pour calculer les coefficients de convection dans un arrangement de tubes. Un programme de calcul basé sur les équations aux différences finies du modèle est développé pour calculer la performance thermique de l'échangeur–caloduc. L'analyse est finalement validée par une expérimentation et elle est montré applicable aux problèmes de l'ingénierie.

EIN VERFAHREN ZUR UNTERSUCHUNG VON WÄRMEROHR-WÄRMETAUSCHERN

Zusammenfassung—Es wurde ein Verfahren zur Untersuchung des thermischen Verhaltens von Wärmerohr-Wärmetauschern entwickelt, das auf einem Leitungs-Modell basiert. Die spezifische Wärmeleitfähigkeit des Wärmerohres wurde mit Hilfe eines Versuchs am Einzelrohr ermittelt. Zur Berechnung der Wärmeübergangs-Koeffizienten bei der Konvektion in Rohrbündeln wurden die wohlbekannten universellen Korrelationen verwendet. Damit wurde nach dem Finite-Differenzen-Verfahren ein Computer-Programm zur Berechnung des thermischen Verhaltens von Wärmerohr-Wärmetauschern entwickelt. Diese theoretische Untersuchung wurde schließlich experimentell überprüft und erwies sich für ingenieurmäßige Anwendungen als geeignet.

МЕТОД АНАЛИЗА ТЕПЛООБМЕННИКА НА ТЕПЛОВЫХ ТРУБАХ

Аннотация—В работе развит основанный на модели проводимости метод анализа теплового режима теплообменника на тепловых трубах. Эффективная теплопроводность тепловой трубы получена на основании экспериментальных исследований отдельной тепловой трубы, описанной в работе, а для расчета коэффициента конвективного теплообмена в пучках труб использовались хорошо известные универсальные уравнения. Разработана программа счета, основанная на уравнениях в конечных разностях, для определения тепловой характеристики теплообменника на тепловых трубах. В результате анализ обосновывается данными эксперимента. Показаны возможности технического применения.