Experimental Investigation of Heat Pipe with Annular Fins under Natural Convection at Different Inclinations

Gangacharyulu Dasaroju, Sumeet Sharma, Sanjay Singh

Abstract**—**Heat pipe is characterised as superconductor of heat because of its excellent heat removal ability. The operation of several engineering system results in generation of heat. This may cause several overheating problems and lead to failure of the systems. To overcome this problem and to achieve desired rate of heat dissipation, there is need to study the performance of heat pipe with annular fins under free convection at different inclinations. This study demonstrates the effect of different mass flow rate of hot fluid into evaporator section on the condenser side heat transfer coefficient with annular fins under natural convection at different inclinations. In this study annular fins are used for the experimental work having dimensions of length of fin, thickness of fin and spacing of fin as 10 mm, 1 mm and 6 mm, respectively. The main aim of present study is to discover at what inclination angles the maximum heat transfer coefficient shall be achieved. The heat transfer coefficient on the external surface of heat pipe condenser section is determined by experimental method and then predicted by empirical correlations. The results obtained from experimental and Churchill and Chu relation for laminar are in fair agreement with not more than 22% deviation. It is elucidated the maximum heat transfer coefficient of 31.2 W/(m^2-K) at 25˚ tilt angle and minimal condenser heat transfer coefficient of 26.4 W/(m^2-K) is seen at 45˚ tilt angle and 200 ml/min mass flow rate. Inclination angle also affects the thermal performance of heat pipe. Beyond 25^o inclination, heat transport rate starts to decrease.

Keywords**—**Annular fins, condenser heat transfer coefficient, heat pipe, natural convection, tilt angle.

I. Introduction

THE heat pipe is a vacuum sealed hollow container. The container is made of a material that has high thermal conductivity, such as copper. Small amount of container is filed with liquid working fluid while the rest of it is occupied by vapour state of the same liquid. The wick structure develops necessary capillary forces on the working fluid that condenses in the condenser. The container should be chosen in such a way that it has good strength to weight ratio. The fin array is provided on the condenser section to enhance the heat rejection rate as a result of increase in the surface area [1]-[4]. The objective for the present investigation on heat pipe is to predict the condenser side heat transfer coefficient of copper water heat pipe at different inclinations with annular fins under natural convection and compare the experimental value of condenser side heat transfer coefficient with analytical method and find out at what inclination angle the maximum heat transfer coefficient is achieved.

Gangacharyulu Dasaroju is Professor, Department of Chemical Engineering, Thapar Institute of Engineering & Technology, PATIALA – 147004, Punjab, India (e-mail: gangacharyulu@gmail.com).

Sumeet Sharma is Associate Professor and Sanjay Singh is Research Scholar, Department of Mechanical Engineering, Thapar Institute of Engineering & Technology, PATIALA – 147004, Punjab, India.

II. Experimental

In this experimental work, heat pipe used is made of copper because of its high thermal conductivity. Distilled water is used as working fluid. The heat pipe is of 7.8 mm outer diameter, 6 mm inner diameter and 180 mm length. The lengths of evaporator, adiabatic and condenser sections of heat pipe are 60 mm, 55 mm and 65 mm, respectively. A specially designed heat pipe manufactured by M/s Golden Star, Pune, India is used. The screen mesh wick used is made of phosphor-bronze. To control the temperature of water inside the tank, proportional-integral-derivative (PID) controller is used. The pipe test rig has been fabricated in such way to carry out the experiment at different inclination angles under natural convection. Tilting mechanism is used to lift the heat pipe when it goes from horizontal to vertical position and is provided at the evaporator section of heat pipe, as shown in Fig. 2. A nut and bolt arrangement can be used to loosen and fix the heat pipe at required inclination.

The arrangement is made to change and measure the angle of inclination accurately with the help of angular protector. Resistance temperature detectors are used to measure the temperature of circulating water which are calibrated before start experiment. Six RTD sensors are placed at the surface of heat pipe at the distance of 5 mm, 55 mm, 65 mm, 110 mm, 120 mm and 175 mm, respectively.

The evaporator section of heat pipe is enclosed by cylindrical jacket and it works as heating chamber with hot fluid. The adiabatic section of copper water heat pipe is insulated with glass wool and polyurethane. The main function of adiabatic section of heat pipe is to prevent heat escape in in this section, so that all the heat energy transfer from evaporator section dissipates from only condenser surface. The condenser section of 65 mm axial length is cooled by fitted circumferential fins under free convection.

The rotameter and temperature sensor used in this test-rig are calibrated before actual experiment to know its accuracy. If calibration is not done properly it always gives wrong value. The schematic of heat pipes with annular fins experimental setup is shown in Fig. 1 and Table I gives the detail of various instruments used in the setup.

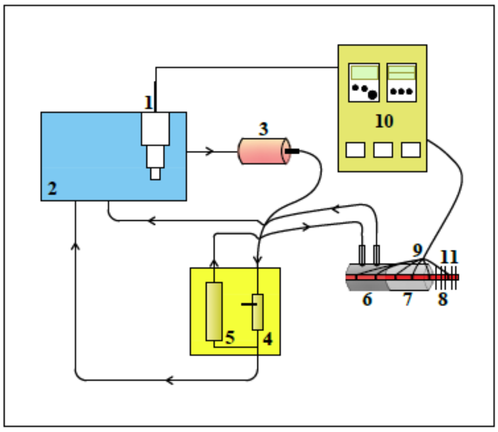

Fig. 1 Schematic heat pipe with annular fins test rig used in this study

TABLE I
PART DESCRIPTION OF EXPERIMENTAL SETUP

1	Heating element
2	Water tank
3	Pump
4	By-pass valve
5	Flow meter
6	Evaporator section of heat pipe
7	Adiabatic section of heat pipe
8	Condenser section of heat pipe
9	Pt-100 RTD sensors
10	Data logger
11	Annular fin

Water is heated in the tank with the help of water heating element. The temperature kept constant in this experiment by using PID control and mass flow rate is varied from 40 ml/min to 200 ml/min. A data logger has been used to record the temperature at various locations on the surface of condenser of heat pipe as well as hot water temperature at evaporator section inlet and outlet. Annular fins are made up of aluminum alloys 6063 because of its high thermal conductivity [5]-[9].

TABLE II
HEAT PIPE SPECIFICATIONS AND THEIR NOTATION

S. No	Parameter	Value
1.	Thermal conductivity of solid wick material	50 W m^{-1} k^{-1}
2.	Thermal conductivity of heat pipe material	400 W m^{-1} k^{-1}
3.	Vapor core region cross-section area	1.9625×10^{-5} m
4.	Wick cross-section area	8.635×10^{-6} m^2
5.	Heat pipe wall thickness	0.9×10^{-3} m
6.	Wick wire thickness	0.0406×10^{-3} m
7.	Effective thermal conductivity of wick	1 W m^{-1} k^{-1}
8.	Evaporator section length	60×10^{-3} m
9.	Adiabatic section length	55×10^{-3} m
10.	Condenser section length	65×10^{-3} m
11.	Vapor core diameter	5×10^{-3} m
12.	Hydraulic radius of heat pipe	2.5×10^{-3} m
13.	Inside radius of heat pipe	3×10^{-3} m
14.	Outside radius of heat pipe	3.9×10^{-3} m

The optimum fin spacing and fin length vary between 5 and 6 mm and 8 and 10 mm, respectively, for maximum heat transfer in free convection conditions. The thickness of fins is also an important parameter and it should be kept as small as possible to avoid conduction but in this study the fins are of uniform thickness 1 mm because beyond this level it is difficult to manufacture due to mechanical constraints. Keeping this consideration in view, the condenser section of heat pipe has been provided with 9 annular aluminium fins of 32 mm diameter at a pitch of 6 mm. The specifications of copper-water heat pipe and annular fins are given in Tables II and III, respectively.

TABLE III
ANNULAR FINS SPECIFICATION AND THEIR NOTATION

S. No	Parameter	Value
1	Fin base radius	6 mm
2	Fin tip radius	16 mm
3	Length of fin	10 mm
4	Spacing between two fins	6 mm
5	Height of condenser length	65 mm
6	No of fins	9

III. TEST PROCEDURE & ANALYTICAL METHOD

The following procedure is adopted to determine heat transfer coefficient:

1. Annular fin is assembled and placed over the condenser section of heat pipe with RTD sensors
2. The performance of heat pipe under free convection, number of test is conducted under steady state condition.
3. The inclination of heat pipe from the horizontal is varied and kept from 0° to 45° tilt angle.
4. The mass flow rate of hot fluid is varied from 40 to 200 ml/min.
5. The ambient temperature has varied between 32 and 34 °C.
6. Once the steady state is achieved, all the required temperatures at various sections are noted.
7. The heat transfer coefficient of copper heat pipe with aluminium annular fin over condenser section under natural convection is calculated.

The arrangement to change and measure the angle of inclination accurately with the help of angular protector is shown in Fig. 2.

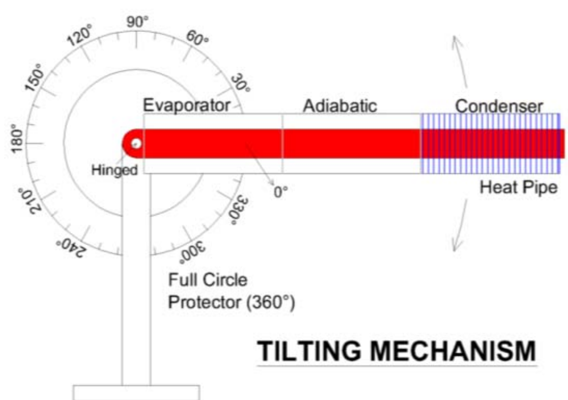

Fig. 2 Schematic diagram of tilting mechanism

The heat input of the heat pipe is equal to amount of heat supplied at evaporator section and is finally obtained as shown in [10]-[12]:

$$Q = \dot{m}c_p(T_1-T_2) \tag{1}$$

where $\dot{m}$ = mass flow rate of circulating fluid (kg/s); C_p = specific heat of water at particular temperature; T_1 = Water inlet temperature (°C); T_2 = Water outlet temperature (°C).

The amount of heat dissipated by the condenser with annular fin over condenser section is equal to the heat absorbed by the circulating water in the jacket. Heat loss can be calculated by:

$$Q = h_C A\,(T_C-T_a) \tag{2}$$

where h_C = Heat transfer coefficient of condenser (W/m^2-K); T_C = Average temperature of condenser surface (°C), T_a = Ambient temperature (°C).

Area of fin surface can be calculated by:

$$A = NA_f+A_b \tag{3}$$

where A = Total surface area of fin, N = Number of fins, A_f = Surface area of each fin, A_b = Area of prime surface.

The condenser side heat transfer coefficient is calculated by:

$$\text{Heat lost} = \text{Heat gained}$$
$$h_C A\,(T_C-T_a) = \dot{m}c_p(T_1-T_2) \tag{4}$$

To find out analytically the heat transfer coefficient of condenser section of heat pipe, the following steps are required.

(a). To find film temperature, T_f (°C):

$$T_f = \frac{(\text{Average wall temperature of condenser} + \text{Ambient temperature})}{2} \tag{5}$$

With the help of film temperature, we note down the thermo-physical properties of air at corresponding film temperature such as kinematic viscosity, thermal conductivity, Prandtl number and coefficient of volume expansion.

(b). To find Grashof Number (Gr)

$$Gr = \frac{g\beta(T_C - T_a)L^3}{\upsilon^2} \tag{6}$$

Characteristic length of fins surface is calculated by following basic definition;

$$L = L+t/2 \tag{7}$$

(c). To find Rayleigh Number (Ra):

$$Ra = Gr * Pr \tag{8}$$

(d). Flow condition:

1. If value Ra is more than 10^9, then flow is turbulent.
2. If value Ra is less than 10^9, then flow is laminar.

(e). Nusselt Number, Nu:

Nusselt number can be obtained from Churchill and Chu empirical relation.

1. If Ra lies between 10^{-1}<Ra<10^{12}, then Nusselt number for all entire range of Ra can be calculated by:

$$Nu_{plate} = \left\{0.825 + \frac{0.387\,Ra^{\frac{1}{6}}}{\left[1+\left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right\}^2 \tag{9}$$

2. If Ra lies between 0<Ra<10^9, then Nusselt number for laminar can be calculated by:

$$Nu_{plate} = 0.68 + \frac{0.670\,Ra^{\frac{1}{6}}}{\left[1+\left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}} \tag{10}$$

Since the heat pipe is in cylindrical shape, so the corrected Nusselt number for cylinder can be determined as:

$$Nu_{corrected} = Nu_{plate}\,(1+1.43£^{0.9}) \tag{11}$$

where,

$$£ = \left(\frac{L}{d_o}\right) Gr^{-1/4}$$

(f). To find heat transfer coefficient:

$$h = \frac{Nu\,k_{air}}{L} \tag{12}$$

The analytical heat transfer coefficient can be compared with experimental value to find out the percentage deviation between experimental heat transfer coefficient and analytical heat transfer coefficient.

IV. Results and Discussion

The results obtained after the experimentation of heat pipe are discussed here. Condenser heat transfer rate is calculated for each mass flow rate and for each angle of inclination. While carrying out the experiment, the range of mass flow rate and angle of inclination are kept at 40 to 200 ml/min and 0 to 45°, respectively.

Rotameter is used in this experiment to measure the volumetric flow rate of hot fluid is calibrated. The calibration is conducted in at five different flow rates, at 30 ml/min, 60 ml/min, 90 ml/min, 120 ml/min and 150 ml/min and the calibration is shown in Fig. 3.

RTD sensors used in this test facility to measure the temperatures are calibrated. The calibration is conducted at different temperatures; 20 °C, 30 °C, 40 °C, 50 °C, 60 °C, 70 °C and error found to be within ± 1 °C to 2 °C. The calibration graph is shown in Fig. 4.

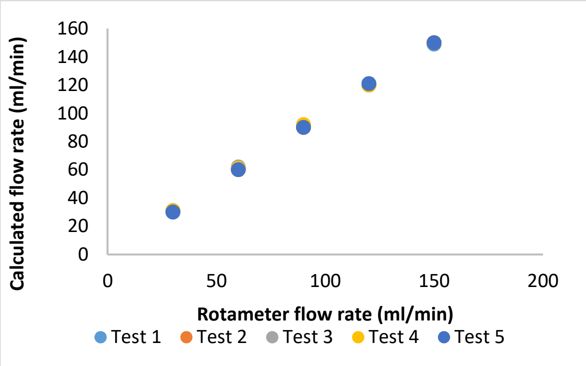

Fig. 3 Calibration of rotameter

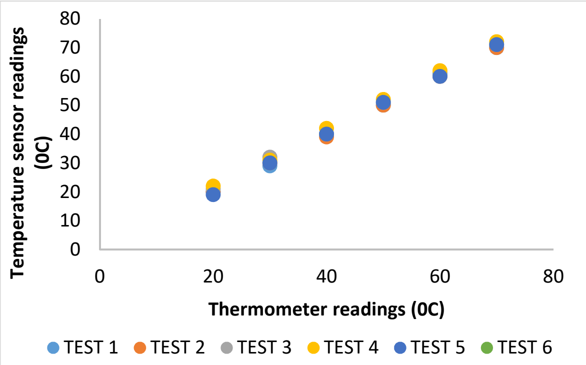

Fig. 4 Calibration of temperature sensor

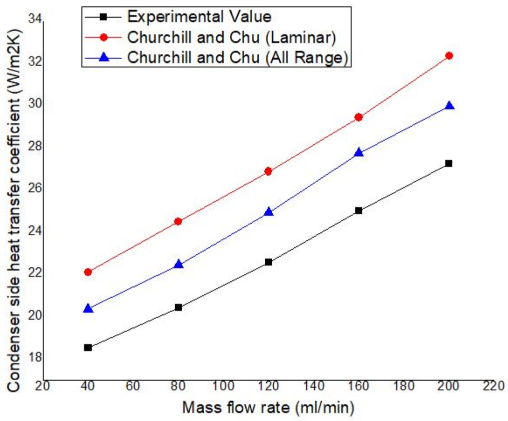

Fig. 5 Variation of condenser heat transfer coefficient with mass flow rate at 0° inclination

The empirical correlation of Churchill and Chu has been used to predict the heat transfer coefficient on the condenser side of copper water heat pipe at 0° inclination. The empirical results are compared with experimental results and these are shown in Fig. 5. The percentage deviation between Churchill & Chu (laminar range) correlation value and experimental value is of 19.25%, 20.4%, 19.20%, 18.2% and 19.60%, respectively. Similarly, the deviation between Churchill & Chu (all entire range of Ra) correlation value and experimental values is of 10.23%, 10.11%, 9.85%, 10.45% and 9.54%, respectively. It has also been observed that the heat transfer coefficient calculated by Churchill & Chu empirical correlations for laminar range are away from experimental value compared to Churchill & Chu empirical correlations for all entire range.

Similar experiments are conducted at different tilt angles for different flow rates of hot water and calculated the condenser side heat transfer coefficient. These values are presented in Table IV and presented in Fig. 6.

TABLE IV
CONDENSER HEAT TRANSFER COEFFICIENT AT DIFFERENT MASS FLOW RATE AND TILT ANGLES

Tilt angle, degree	40 ml/min	80 ml/min	120 ml/min	160 ml/min	200 ml/min
0	18.50	20.40	22.26	24.98	27.21
5	19.50	21.30	23.28	26.10	29.23
10	20.20	22.10	23.92	26.50	29.70
15	20.40	22.60	24.12	27.20	30.10
20	20.50	23.10	25.50	28.10	30.30
25	20.80	23.20	25.90	28.30	31.20
30	18.46	20.60	23.10	26.30	29.10
35	17.60	19.80	22.20	24.10	27.30
40	17.20	19.40	21.50	23.60	26.80
45	16.80	18.90	20.90	23.40	26.40

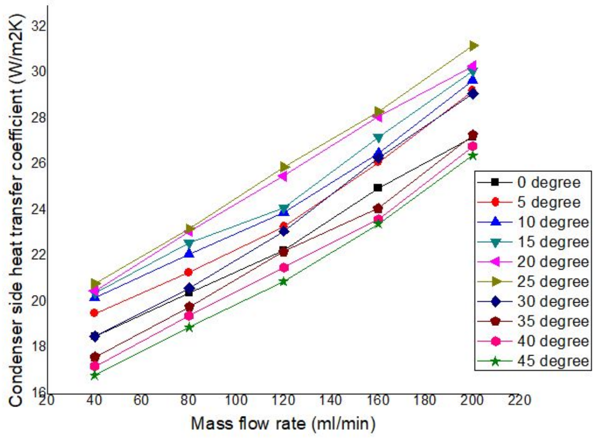

Fig. 6 Comparisons of condenser heat transfer coefficient with mass flow rate at different angle of inclination

It is observed that heat pipe beyond 30° inclination angle shows deterioration of performance.

V. CONCLUSIONS

Numerous experiments are conducted on a straight copper water heat pipe with annular fins under natural convection to examine its performance at different inclinations. The analytical model has been developed to predict the individual heat transfer coefficient at the surface of condenser and compared with experimental results. The noticeable observation can be drawn from present investigation are:

1. With the increase in mass flow rate, the rate of heat transfers of heat pipe also increases since mass flow rate is linearly proportional to the heat transfer coefficient.
2. The predicted heat transfer coefficient by Churchill and Chu (for laminar flow) and experimental heat transfer coefficient are in fair agreement with deviation no more

than 22%.

3. The predicted heat transfer coefficient by Churchill and Chu (for all range of Rayleigh number) and experimental heat transfer coefficient are in fair agreement with deviation not more than 11%
4. It is envisaged that inclination angle had the most significant effect on heat pipe performance. It is concluded that heat transfer rate of heat pipe increases when heat pipe operates between inclinations at 0 to 25°. It is observed that beyond 25° the heat transfer rate starts to decrease.
5. The maximum heat transfer coefficient under free convection is 31.2 W/m^2-K at 25° inclination and least is obtained at 45° inclination, which may be due to poor buoyancy driven flow in annular finned surface of condenser.
6. The condenser temperature is high at 45° tilt angle, but at this angle condenser heat transfer is low may be because of incomplete formation of boundary layers.

NOMENCLATURE

A	Area, m^2
Gr	Grashof number
h	Heat transfer coefficient, W/(m^2-K)
k	Thermal conductivity, W/(m-K)
PID	Proportional Integral Derivative
Pr	Prandtl number
Q_{in}	Heat Input, W
T	Temperature, °C
Subscripts	
a	ambient
c	condenser
f	film
o	outside

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