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

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***Abstract-* This paper presents the experimental investigations of heat pipe with Annular fins under natural convection were performed on 10 different tilt angles of heat pipe (0^0, 5^0, 10^0, 15^0, 20^0, 25^0, 30^0, 35^0, 40^0, and 45^0) five different mass flow rate were applied over the evaporator section (40ml/min, 80ml/min, 120ml/min, 160ml/min and 200 ml/min). So a set 50 experiments were performed to determine at what inclination angle and at what mass flow rate give maximum condenser heat transfer coefficient. Experimental results show that the mass flow rate is linearly proportional to the heat transport rate. The maximum condenser heat transfer coefficient of 31.2 W/m^2-K was observed at 200 ml/min for 25^0 tilt angle. On average, the minimal condenser heat transfer coefficient of 26.4 W/m^2-K was seen at vertical 45^0 heat pipe tilt.**

Keywords – Heat pipe, Natural convection, Annular Fins, Inclination angles

I. INTRODUCTION

In this scenario, cooling of engineering equipment's was major assignment in today engineering research. When we work on engineering equipment's as well as electronic components that outcome in creation of heat that could result in system failure due to excessive heating. Thus there is serious need of loftier heat transfer element with increasingly smaller weight, volume and cost. There are numerous systems like refrigerators and transformers that require urgent dissipation of heat for their optimum operation with advancement in machinery, the demand for heat carrying machinery has also spurred up. To overcome these type of limitations, Heat pipe is one of the novel solution to achieve the desired rate heat dissipation. In the conventional heat transport units of the 1880s like Perkin's boiler, in this boiler very simple mechanism is used to fetch a heat from one place to another place. Perkin's boiler was design in such a way in which liquid was heated with the help of hot gasses placed above the pipe in which liquid flow. In Perkins tube, heat transfer can be attain with the aid of change of phase [1]. The Perkins tube design was very close to the existence heat pipe appliance. The heat pipe was first introduced by Gaugler's and also recommended the use of a capillary structure wick [2]. After that, in early 1960s, Grover and his team partners was revel that the productivity of heat pipe was very high as compared to other traditional heat transport device and most economical heat transfer system today [3]. Heat pipe mainly consists of the three section first is evaporator section, second is adiabatic section, and third one is condenser section. Heat pipe also termed as superconductor of heat. The heat pipe has ability to lift up heat transfer with minimal heat loss.

In this plot, cooling of engineering equipment demand was increase day by day due to elevation in machinery. It observed that if we want any engineering equipment can perform smoothly that is must to avoid overheating otherwise device can be damaged in little time. So, it can be done by cooling. Cooling rise up the life of equipment and increase the efficiency of component consequential was envisages by [4-7]. To overcome this problem and to achieve desire rate of dissipation as when fins are utilizing. The heat fritter away from system to surrounding can be acquire with the help of free or forced convection.

Mostly to control the temperature between system and surrounding is not optimum and enhancing the heat transport rate with the help of external source such as pump, fan etc. lead to the cost. So demand is that type of device which increase the heat transfer rate and no more effect on cost and maintenance. Fins (extended surface) fulfill all types of requirements.

The use of fins is more suitable, concern free and cheap way to increase the heat transport rate. In today most of the engineering component used fins by fabricate fins above the surface in which fluid flow to obtain desirable amount of heat transfer. However, it also should be kept in mind if we add more and more number of fins to increase the area of contact surface that doing area of surface increase but heat transfer rate start decreasing because in between fins formation of boundary layer does not fully developed [8-11].

II. LITERATURE REVIEW

Extended surface or fins are used to increases the heat transfer due to increase surface area of cross section in which convection process occur. The fin material should have high thermal conductivity and minimize temperature variation from base to top. There are many type of fin including Pin fin, Straight fin, Annular fin etc. The previous literature suggests under natural convection the use of three or four fin per inches [12]. Length of fin is also one important parameter of fin. We know that the length of fin is directly proportional to the fin length. However, temperature drop along the fin follow exponentially path so that's why it reaches the surrounding temperature at some length. After beyond this length it does not contribute any heat transfer. Therefore, design extra-long fin is meaningless as a result wastage of material, increase size and excessive weight and also cost increases [13]. Nagarani et al. [14] presented the study of how fin with heat exchangers has been used over the last 20 years in the field of heat transfer. Due to the advancement of technology, most of the industries required effective heat transfer components with less weight, volume and cost. The author was investigated five major type of fins are as: annular fins, elliptical fins and elliptical tube, pin fins, longitudinal fins by experimental and analytical method. It was observed that coating on fins increase the heat transfer rate. It was also observed that elliptical fins will be better choice as compared to annular and eccentric fin. Senapati et al. [15] did an investigation of natural convection heat transfer from a vertical cylinder with annular fin have been studied numerically by varying Rayleigh number in both laminar and turbulent flow. His calculation was carried out by varying fin spacing to diameter ratio (S/d) and fin to tube diameter ratio (D/d) in the range of 0.126-5.84mm and 2-5 respectively. According to the author observation, with the addition of fins to the temperature constant cylindrical wall, the heat transfer goes on increasing for laminar flow and for turbulent flow firstly the heat transfer increases to a maximum value and after that decreases with further additions of fins. It also established that the maximum heat transfer takes place in the case of turbulent flow and also predicted that optimum fin spacing lies between 7 and 7.7mm

Senapati et al. [16] performed a numerical investigation of natural convection heat transfer with annular fins over horizontal cylinder. In the present study, author used numerical simulation of full naiver stoke equation along with energy equation has been conducted with annular fins of constant thickness for the laminar range $5 \leq Ra \leq 10^8$. After the result, the author observed that the ideal fin spacing for maximum heat transfer lies between 5 to 6 mm for Ra in the range $5 \leq Ra \leq 10^8$. It also established the correlation for optimum fin spacing as of function of Ra and D/d which can be very helpful to industrial purpose. Kumar et al. [17] investigate the heat transfer of heat pipe by comparing experimental data and analytical model. In this research, the evaporator portion of wire screen heat pipe is subjected to forced convection and condenser portion is under free convection air cooling. In this paper analytical model was establish on the basis of thermal network resistance approach. This model determines thermal resistance at the outer surface of the evaporator as well as condenser. The main goal of experiment to evaluate the thermal performance of heat pipe. It also studies the effect of many operating parameters such as heat pipe inclination angle and heating fluid inlet temperature on the evaporator where investigated by experimental. The experiment result compare with analytical model for the validation. It found that the heat transfer coefficient predicted by the model at outer surface a wire screen heat where observed to be in acceptable agreement with experimental result. It also concludes that maximum heat transport rate of heat pipe was found at inclination angle of 25^0 and at 70^0C heating fluid. Hong-Sen et al. [18] has been studied about a circumferential fins and divided into numerous circular section. In all circular region the rate of heat transfer along with thermal conductivity of every part was to be assumed. In this paper, a recursive formula was used to give the solution of temperature distributions and rate of heat transport coefficient on annular fin for both condition. Naphon et al. [19] investigated the heat transfer coefficient and fin efficiency of circular fins. Circular fin area observed in three condition dry, partly wet, and absolutely wet surface situation. Researcher developed analytical model to determine the temperature profile of the fin. The results obtained from mathematical model was fair agreement with other researchers.

Kundu et al. [20] has been presented a research paper on clipped fins with the aid of semi analytical technique. In this technique, describe a which type of fin are used as well explain how volume and rate of heat transfer effect the performance of any equipment which fins are placed. Elliptical fins give better dissipation rate as compared to circumferential fin along with area resist exist on either side of fins. Elliptical fins give better performance on the comparison of eccentric circular fin if the restriction was on one side. Chen-Nan et al. [21] has been analyzed a research paper of combined effect of rate of heat transfer and also mass transfer in clipped fins subjected to be all three different surface conditions. The author found that if the air humidity was spurred up as a result temperature distribution of fin was also increased. It also observed that elliptical fin efficiency is high nearly about 4% as compared to circumferential fin efficiency.

Abdel- Rehim Zeinab [22] will investigated the impact on overall thermal performance of extended surface such as square, circumferential, and clipped pin fins associated with different fin geometries. The authors used EGM to find the resultant impact of thermal resistance as well as pressure falls. The mathematical model was developed on the basis of dimensionless variable. The result exposed that fin profile was significantly depend on these parameters.

III. EXPERIMENTAL SETUP

A Fabrication of Heat pipe

In this experimental work, heat pipe used is made of copper because of its high thermal conductivity. Copper water heat pipe with 7.8 mm outer diameter, 6 mm inner diameter and length of heat pipe is 180 mm has been selected for procuration of heat pipe. 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. It provide at the one end of evaporator section of heat pipe. A nut and bolt arrangement can be used to loosen and fix the heat pipe at required inclination. The evaporator, adiabatic and condenser section of copper water heat pipe a 60 mm, 55 mm and 65 mm in length respectively as shown in Figure 1

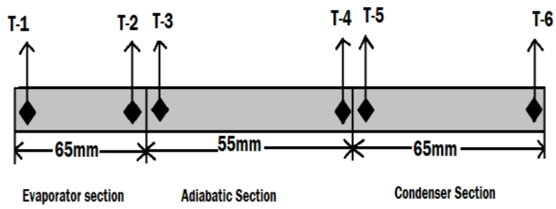

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Figure 1: Schematic diagram of heat pipe with thermocouple

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 as shown in Figure 1.

The rotameter and temperature sensor are used in this experimental work calibration must be done before to determine the accuracy of the instrument. Otherwise, if calibration is not done properly it always give wrong value. The schematic of heat pipes with annular fins experimental setup is shown in Figure 2 and Table 1 gives the detail of various instrument which used in the setup.

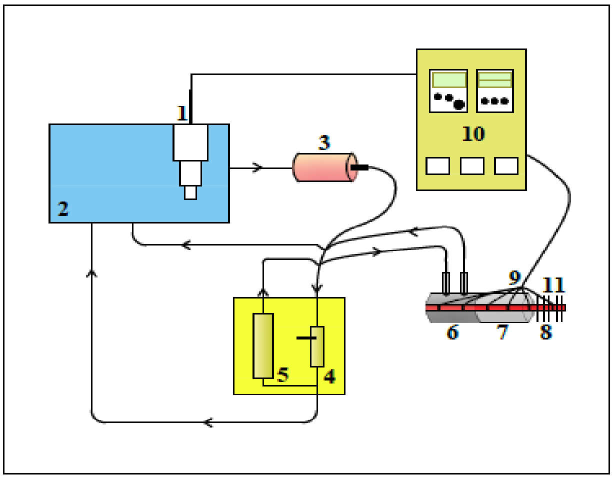

Figure 2 Schematic heat pipe with annular fins test rig Used in this study

Table 1: Part description of experimental test rig

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 power input given to the heater in the form of temperature. Heat input can be controlled in two ways: First is by changing the temperature of circulating water and secondly by changing the mass flow rate. However, for the sake calculation the temperature value in this experiment is kept constant and only mass flow rate is varied from 40 ml/min to 200ml/min. PID mechanism is used to control the process. Steady temperature reached after 30 to 35 min. After steady state, pump is on and hot water has been circulated from the tank towards cylindrical jacket. In this process, the water also passes through a bypass valve where the excess water is released back into the tank and remaining water is allowed to enter the rotameter to measure the flow rate. A data logger system has been used to displayed the temperature of heat pipe at various location on the surface of condenser as well as evaporator inlet and outlet. The front view of fabricated heat pipe with annular fin set up used in this study as shown in Figure 3. The fabrication of heat pipe experimental setup could be better understand from the top view of heat pipe with annular fins as shown in Figure 4. It depicts the layout of pipe from tank evaporator section. The evaporator cylindrical jacket is black in color, adiabatic section is painted with red color and condenser section is enclosed with annular fins under natural convection.

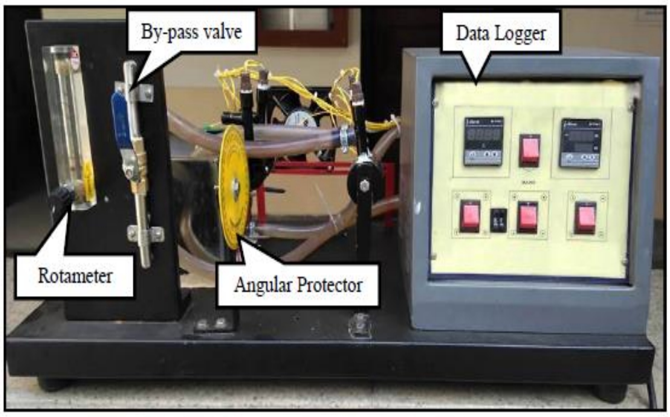

Figure 3: Front view of heat pipe test rig

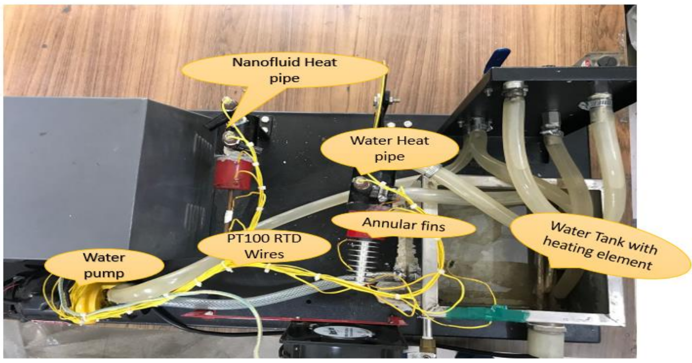

Fig.4: Top view of heat pipe experimental setup

B Fabrication of Fins

Annular fins are made up of aluminum alloys 6063. It is one of the most widely used materials for manufacture of fins. Because it provides good resistance to corrosion and has good extrudibility and high quality surface. A6063 has high thermal conductivity and easily available. A6063 was alloy which is made up of two element one is magnesium and other one is silicon. The standard controlling composition is maintain by aluminum association. It also give good mechanical properties and high heat transfer property. Chemical composition of A 6063 is given in the Table 2. The 3D view of cylider with annular fin as shown in Figure 5.

Table 2: Chemical composition of a A6063

Material	Si	Fe	Cu	Mn	Mg	Cr	Ti	Other
Max %	0.6	0.35	0.10	0.10	0.9	0.10	0.10	0.15
Min %	0.2	0.35	0.10	0.10	0.45	0.10	0.10	0.15

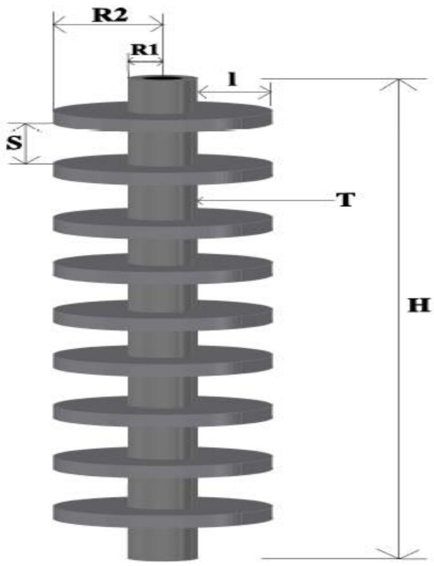

Figure 5. 3D view of cylinder with annular fins

C Specification of Annular fins

Various parameters on which heat transportation depends are length, spacing, no of fins and fin diameter. Specification of annular fin used in experimented are listed below as shown in Table 3.

Table 3: Annular fin specification and their notation

S.no	Notation	Parameter	Value
1	R1	Fin base radius	6 mm
2	R2	Fin tip radius	16 mm
3	l	Length of fin	10 mm
4	S	Spacing between two fins	6 mm
5	H	Height of condenser length	65 mm
6	N	No of fins	9

D (I) Experimental procedure to find heat transfer coefficient

1) Annular fin is assembled and placed over the condenser section of heat pipe with thermocouples (RTD sensors)
2) To study the performance of heat pipe under free convection number of test were conducted under steady state at a constant temperature.
3) The inclination of heat pipe from the horizontal was varied and kept at 0^0 to 45^0 tilt angle.
4) The mass flow rate of hot fluid was also varied from 40 to 200 ml/min.
5) The ambient temperature varied between the range 32^0 C to 34^0 C.
6) Once the steady state is achieved.
7) Note down all the required temperatures at respectively sections.
8) The heat input of the heat pipe is equal to amount of heat supplied at evaporator section and is finally obtained as shown in following equation [13].

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

9) 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 the following equation [13].

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

10) Area of fin surface can be calculated by using following equation [10].

$$A = NA_f+A_b \qquad (3)$$

11) The condenser side heat transfer coefficient was calculated by using following equation [23].

$$\text{Heat lost} = \text{Heat gained} \qquad (4)$$

with the help of equation (4) determine the value of experimental heat transfer coefficient of heat pipe under free convection.

(II) Analytical method to find heat transfer coefficient for natural convection.

a) To find film temperature, T_f (^{0}C) **[24]**

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

With the help of film temperature, 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) [25]:

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

Characteristic length of fins surface is calculated by following basic definition [10];-

$$L = L+t/2 \qquad (7)$$

c) To find Rayleigh Number (Ra) [13]:

$$Ra = Gr*Pr \qquad (8)$$

d) Flow condition [26]**:**

(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.

(i) If Ra lie between $10^{-1}<Ra<10^{12}$, then nusselt number for all entire range of Ra can be calculated by using following equation [27].

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ISSN: 2278-0181
Vol. 7 Issue 07, July-2018

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

(ii) If Ra lie between $0<Ra<10^9$, then nusselt number for laminar can be calculated by following equation [28].

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

If the heat pipe in the form of cylindrical shape. So the corrected nusselt number for cylinder can be determine by using equation [29].

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

Where,

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

f) To find heat transfer coefficient [30]:

$$h = \frac{Nu\; k_{air}}{L} \qquad (12)$$

The analytical heat transfer coefficient can be compared with experimental value and to find out the percentage deviation between experimental heat transfer coefficient and analytical heat transfer coefficient [31].

IV. RESULTS AND DISCUSSIONS

The empirical correlations established by Churchill and Chu for flat plate has been used to predict the heat transfer coefficient on the condenser side of copper water heat pipe. Outside condenser heat transfer coefficient calculated by the analytical model and obtained from experiments. The graphical comparisons among natural convection heat transfer coefficients for experimental results and that of different empirical correlations can be seen is shown in Figure 6. The percentage deviation between Churchill & Chu (laminar range) and experimental of 19.25%, 20.4%, 19.20%, 18.2% and 19.60% was seen. Similarly, the deviation between Churchill & Chu (all entire range of Ra) and experimental of 10.23%, 10.11%, 9.85%, 10.45% and 9.54% was seen in figure 6. It has been observed that the heat transfer coefficient calculated by Churchill & Chu empirical correlations for laminar range is spurred as compared to experimental value. It also noticed that heat transfer coefficient predicted by Churchill & Chu empirical correlations for all entire range is close as compared to experimental value.

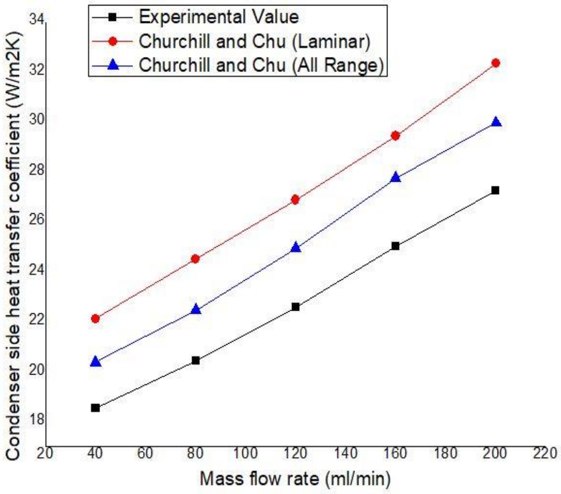

Figure 6. Variation of condenser heat transfer coefficient with flow rate at 0^0 mass inclination

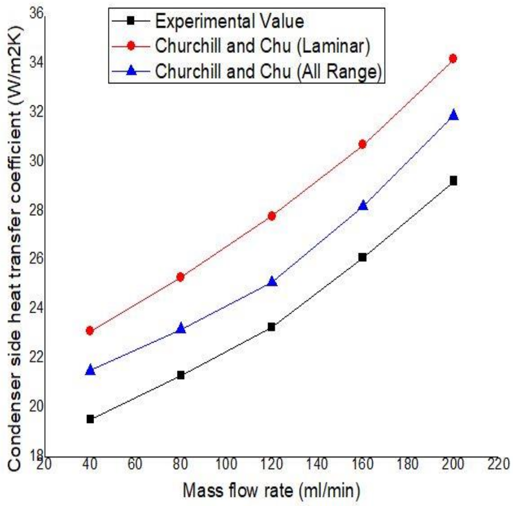

Figure 7. Variation of condenser heat transfer coefficient with mass flow rate at 5^0 inclination

In this graph, the x-axis represent mass flow rate in ml/min an y-axis represents condenser heat transfer coefficient of heat pipe in W/m^2-K. The following points was observed from the Figure 7. With the increase in mass flow rate, the rate of condenser heat transfer coefficient also increases because due to mass flow rate is linearly proportional to heat transport rate. The maximum heat transfer coefficient was found by experimental is 29.23 W/m^2-K at 5^0 inclination it is little more as compared to 0^0 inclination is due to better buoyancy driven flow in annular finned surface of condenser.

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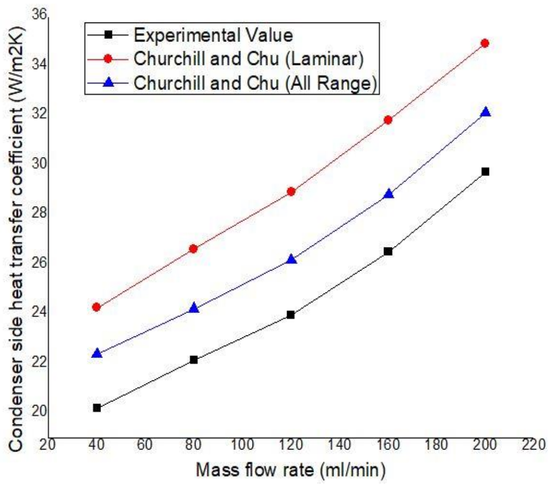

Figure 8. Variation of condenser heat transfer coefficient with mass flow rate at 10^0 inclination

The Figure 8 represent the deviations between experimental value and Churchill & Chu empirical correlations at 10^0 inclination. At this inclinations, noticed that Inclination angle also effect the condenser heat transfer coefficient because it would possible with the increase of inclination angle heat transport rate increases and optimum rate of evaporation and condensate return to the evaporator, which result in lower thermal resistance. With the increase in inclination angle heat transfer coefficient also increase. The maximum heat transfer coefficient was found by experimental 29.8 W/m^2-K is more as compared to condenser heat transfer coefficient at 5^0 inclination i.e. with increase the angle of inclination heat transfer coefficient increase due to lower thermal resistance [17]. At this inclination the percentage deviation between Churchill & Chu for laminar and results obtained from experimental is 19.36%, 20.2%, 21.2%, 20.1%, and 20.23%.
From figure 9 it is clear that at this inclination the percentage deviation between Churchill & Chu for laminar and experimental within the range 6 to 21%. At this inclination the percentage deviation between Churchill & Chu for entire range and experimental within the range 2 to 11%.The maximum heat transfer was noticed at this inclination is 30.2 W/m^2-K by experimental at 200 ml/min.

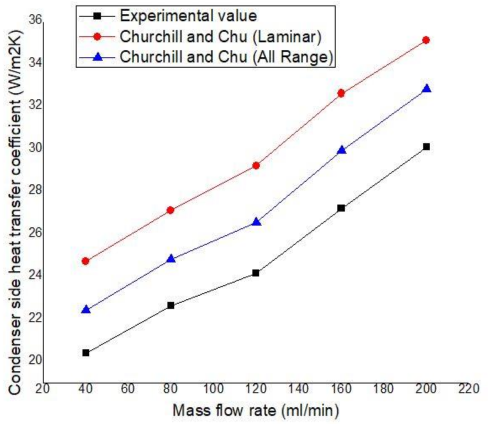

Figure 9. Variation of condenser heat transfer coefficient with flow rate at 15^0 mass inclination.

From figure 10 represent the variation of condenser heat transfer coefficient with mass flow rate at 20^0 inclination. In terms of percentage deviations of 20.25%, 18.91%, 19.26%, 19.67% and 20.67% corresponding to 5 different mass flow rate starting from 40 ml/min to 200 ml/min were observed between experimental results and empirical prediction from Churchill and Chu for laminar. As seen from Figure 10, the variations between empirical results from Churchill and Chu for all range and experiments were significantly less compared to previous case with changes of 10.34%, 9.89%, 10.34%, 8.45% and 10.19%. All these deviation were within permissible limit [32].

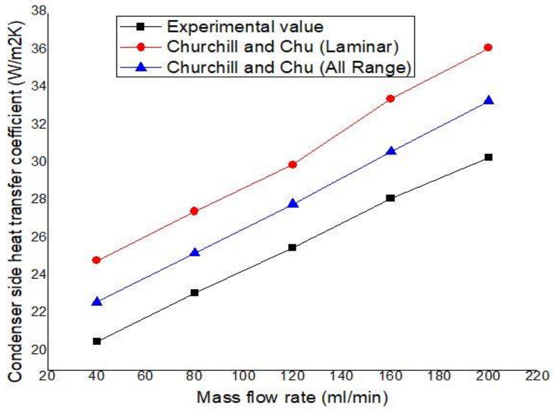

Figure 10. variation of condenser heat transfer coefficient with flow rate at 20^0 inclination

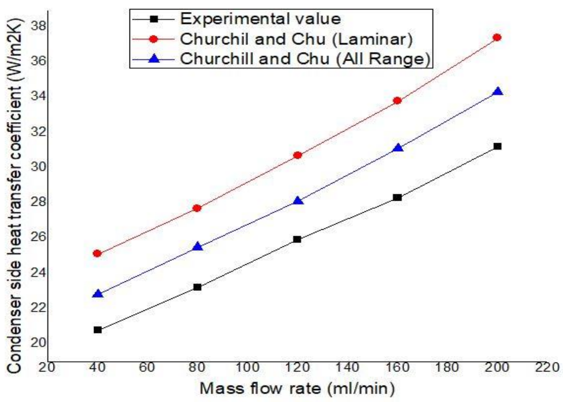

Figure 11. Variation of condenser heat transfer coefficient with flow rate at 25^0 mass inclination.

A graphical comparisons between experimental results and Churchill & Chu empirical correlations is shown in Figure 11. At this inclination the maximum condenser heat transfer coefficient was found 31.2 W/m^2-K at 200 ml/min mass flow rate due to better exposure of condenser annular finned surface to ambient environment and the optimum rate of evaporation and condensate return to the evaporator, which results in its lower internal and external thermal resistance. The condenser heat transfer coefficient, predicted by Churchill and Chu for all range of Rayleigh number is closer (2 to 11%) to the experimental value as compared to Churchill and Chu correlation for laminar range, which varied in the range of (5 to 22%).

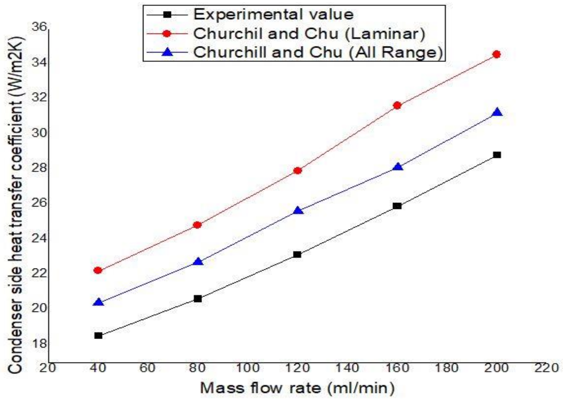

Figure 12. variation of condenser heat transfer coefficient with flow rate at 30^0 inclination

From figure 12 presented the percentage deviation between results obtained from the experimental and result calculated by analytical model. At this angle of inclination following point was observed. At this inclination it was found that condenser heat transfer coefficient was started decreased as compared to condenser heat transfer coefficient at 25^0 inclination because temperature of condenser part start increases , as the boundary layer may not be fully developed at the bottom surface of annular fins.

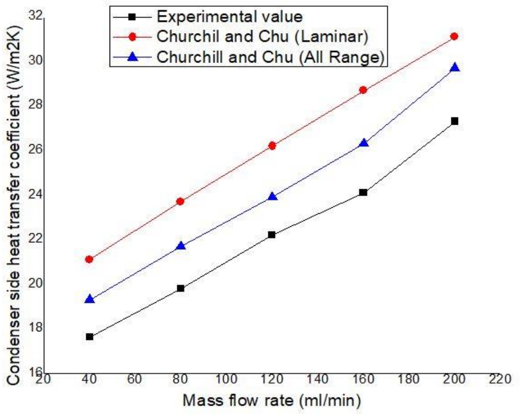

Figure 13. Variation of condenser heat transfer coefficient with flow rate at 35^0 inclination

From figure 13 represent the variation of condenser heat transfer coefficient with mass flow rate at 35^0 inclination. At this angle of inclination, it found that beyond the 25^0 inclination condenser heat transfer coefficient start decreases. It also noticed that at this angle maximum condenser heat transfer coefficient is 12.5% less as compared to 25^0 inclination angle due to less amount of heat transferred at this angle. The best thermal performance of heat pipe has been reported in the range of 15-30^0 inclination by various authors [27-29].

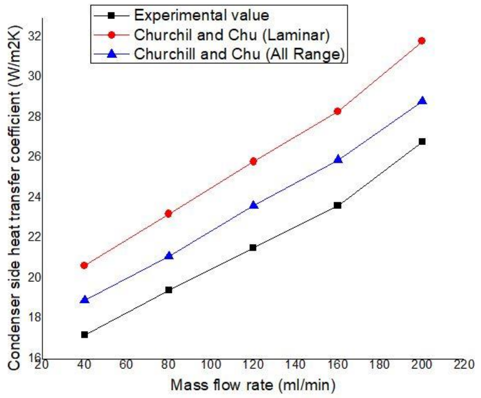

Figure 14. Variation of condenser heat transfer coefficient with flow rate at 40^0 inclination

At this angle of inclination ,In terms of percentage deviations of 21.25%, 19.91%, 20.26%, 18.67% and 21.67% corresponding to 5 different mass flow rate starting from 40 ml/min to 200 ml/min were observed between experimental results and empirical prediction from Churchill and Chu for laminar. As seen from Figure 14, the variations between empirical results from Churchill and Chu for all range and experiments were significantly less compared to previous case with changes of 9.34%, 10.89%, 8.34%, 9.45% and 10.19%.It also found that maximum condenser heat transfer coefficient is less as compared to 25^0 inclination angle due to condenser dry-out.

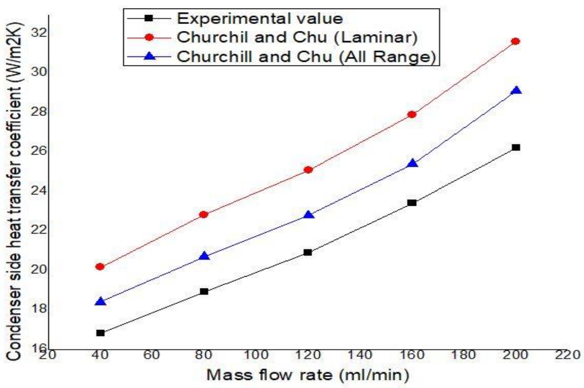

Figure 15. Variation of condenser heat transfer coefficient with flow rate at 45^0 inclination

The graphical comparisons among free convection heat transfer coefficients for experimental results and that of empirical correlations can be seen in Figure 15. At this tilt angle the percentage deviation between Churchill & Chu for laminar and experimental within the range 6 to 21%.At this inclination the percentage deviation between Churchill & Chu for entire range and experimental within the range 2 to 11%. The maximum condenser heat transfer coefficient was found 26.41 W/m^2-K is 18.2 % less as compared to 25^0 inclination angle due to condenser dry-out.

The condenser heat transfer coefficient, which showed a linear increase with mass flow rate. Corresponding to 10 different tilt angles of heat pipe (0^0, 5^0, 10^0, 15^0, 20^0, 25^0, 30^0, 35^0, 40^0, and 45^0) five different mass flow rate were applied over the evaporator section (40ml/min, 80ml/min, 120ml/min, 160ml/min and 200 ml/min). So a set 50 experiments were performed to determine at what inclination angle and at what mass flow rate give maximum condenser heat transfer coefficient. The maximum condenser heat transfer coefficient of 31.2 W/m^2-K was observed at 200 ml/min for 25^0 tilt angle. On average, the minimal condenser heat transfer coefficient of 26.4 W/m^2-K was seen at vertical 45^0 heat pipe tilt. Graphical results for the same are demonstrated in Figure 16.

Table 4: Condenser heat transfer coefficient (W/m^2-K) of copper water heat pipe at different mass flow rate and tilt angles

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

From the Figure 16 , it is clear that condenser heat transfer coefficient was maximum at 25^0 inclinations as compared to other inclination angle. At this angle heat transfer is maximum because it would be possible better buoyancy driven flow in condenser section. The best thermal performance of the heat pipe under this operating condition may be attributed due to better exposure of condenser annular finned surface to ambient environment and the optimum rate of evaporation and condensate return to the evaporator, which results in its lower internal and external thermal resistance. After 25^0 inclination heat transfer rate goes on decreasing. This is primarily due to the fact that beyond 25^0 inclination may be restriction of boundary layer formation at higher angle of elevation that lead to reduction of heat transfer coefficient at higher orientation. It is reported that variation in the experimental heat transfer and Churchill and Chu (for laminar range) was around 22%. For all range of Rayleigh was observed to be in the range of 11%. Literature reports that the calculated heat transfer coefficient may vary ± 25 % from the actual value in natural convection [19].

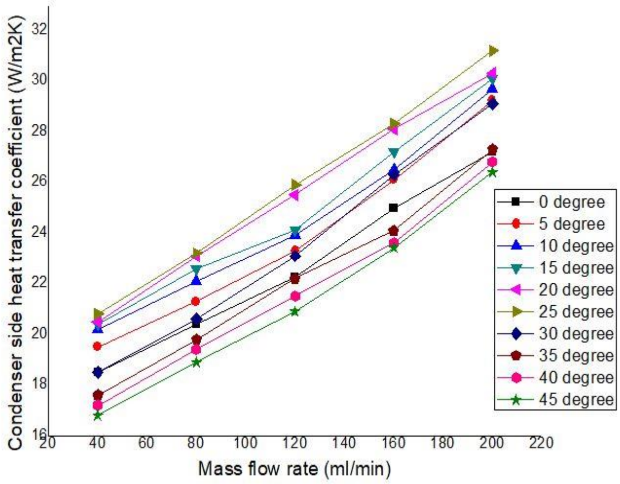

Figure 16 comparisons of condenser heat transfer coefficient with mass flow rate at different angle of inclination.

V. CONCLUSIONS

There was numerous experiments were conducted on a straight copper water heat pipe with annular fins under natural convection to examine it performance at different inclination. The analytical model has been developed to predict the individual heat transfer coefficient at the surface of condenser. The noticeable observation can be drawn from present investigation are as follow:

1. With the increase in mass flow rate, the rate of heat transfers of heat pipe also increases due mass flow rate is linearly proportional to the heat transfer coefficient.
2. The heat transfer coefficient over condenser section with annular fins, under free convection increases with the increase in heat input on the evaporator section.
3. The predicted heat transfer coefficient by Churchill and Chu (for laminar flow) and experimental heat transfer coefficient were in fair agreement with deviation no more than 22%.
4. The predicted heat transfer coefficient by Churchill and Chu (for all range of Rayleigh number) and experimental heat transfer coefficient were in fair agreement with deviation no more than 11%
5. It was envisaged, that inclination angle had the most significant effect on heat pipe performance. It concluded that heat transfer rate of heat pipe increase when heat pipe operates between inclinations at 0^0 to 25^0 because due to better exposure of condenser annular finned surface to ambient environment. Beyond, 25^0 heat transfer rate start decrease.
6. Noticeable, enhancement was found in heat transfer coefficient with annular fins over condenser section under natural convection. The maximum heat transfer coefficient under free convection was 31.2 W/m^2-K at 25^0 inclination and least was obtained at 45^0 inclinations because rate of heat transport is less at this angle, which may be due to poor buoyancy driven flow in annular finned surface of condenser.

7. The condenser temperature is high at 45^0 tilt angle, but at this angle condenser heat transfer is low because may be of incomplete formation of boundary layer.

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