

# MODELLING AND THERMAL SIMULATION OF COMPOSITE WICK HEAT PIPES USING ETHANOL

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#### **ABSTRACT:**

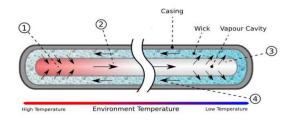
Heat pipes operate with the highest thermal conductivity compared to any other heat transfer and are available in a wide range of parameters. In the present study the flow of water in the phenomenon simultaneous ofevaporation, adiabatic and condensation in plain thermo-siphon, sintered copper wick and helical grooved heat pipes is studied using heat pipe testing equipment. This device uses a water jacket with a controlled flow of water to dissipate heat energy to the end of the heat pipe and condenser subject to a predetermined heat dissipation by the resistance heater at the end of its evaporator. All temperatures are measured and the calculations required to obtain percentage efficiency at different flow rates and heat inputs are performed. A comparison is made between the performance of the heat pipes and their capacities. Container thickness in working fluid fluid ethanol wick is 0.5 mm as 50 to 800 watts evaporator 30, condenser 72, adiabatic 110, flat heat pipe width 7.6 mm, thickness 3.4 mm heat pipe has high capillary property. Variations of the evaporator and condenser surface temperatures are plotted to change the heat inputs and flow rate changes at the condenser water jacket. ANSYS software is used for computational analysis and the experimental results are in good agreement with the analysis.

### 1. Introduction

The technology of heating and cooling of systems is one of the most basic areas of Mechanical engineering. Wherever steam is used, or wherever hot or cold fluids are required we will find a heat exchanger. They are used to heat and cool homes, offices, markets, shopping malls, cars, trucks, trailers, aero-planes, and other

transportation systems. They are used to process foods, paper, petroleum, and in many other industrial processes. They are found in superconductors, fusion power labs, spacecrafts, and advanced computer systems. The list of applications, in both low and high tech industries, is practically endless. In our basic study thermodynamics and heat transfer, studied the form of the control volume energy balance and its application too many engineering problems, including to a basic heat exchanger problem. In this module, we will extend heat exchanger analysis to include the convection rate equation, and demonstrate methodology for predicting heat exchanger performance that include both design and performance rating problems. If the true design flows, heat load, and temperatures cannot be achieved in testing, the test data must be extrapolated (taking into account tube plugging) to compute the overall heat transfer coefficient and compare it to the design value. But any calculation must also consider uncertainty and error in the experimental measurements of temperature and flow. Furthermore, even before an overall coefficient is computed, the test data must be assessed to have sufficient accuracy. This may be done as a pretest where experimental parameters estimated, and it is judged whether the overall heat transfer coefficient can be meaningfully found. For power plant

personnel such analysis is challenging due to the fact that information must be drawn from instrument design and operation, heat transfer applications, and statistical theory. Certainly a great amount of published work already deals with these topics. Although statistical methods are numerous, many techniques are cumber some and difficult to apply to actual power plant heat measurements exchangers as operating units are often limited in sample size and quality; plant personnel must have a relatively simple and practical, but effective, method to judge the data. simplified However, a methodology specific to power plant coolers is not readily available in the common literature. Knowledge of instrument sensor accuracy, impact of number of sensors, data sampling intervals, and a reasonable statistical confidence level are some of the factors to be considered, especially in the case of temperature measurement. To address this, the present work, drawing from industry standards, presents a centralized, practical statistical approach and quantifies these factors so that heat exchanger temperature data may be easily and quickly assessed. sample calculations Furthermore, presented to facilitate the use of the proposed methodology. Although concentration is on temperature measurement, the procedure can be easily extended to flow measurement or any other parameter



## 1.2 PERFORMANCE PARAMETERS OF HEAT PIPE

Heat pipe has three components.

- 1. Casing
- 2. Working fluid
- 3. Wick structure

#### 1.3 COMPOSITE WICK HEAT PIPE

Composite wick structures use the benefits of having little pores for making high capillary pumping and having huge pores for extending the permeability of the liquid return way. Afresh, the most direct sort of composite wick is the screen wick, beside that two screens with different pore sizes are used. A couple of wraps of a screen with a significant pore measure is used against the internal pipe divider for the liquid return way, and a solitary wrap of a screen with an extensively tinier pore appraise is put touching the vapor space to develop high capillary pressure. Basically, axial groves anchored by a single wrap of a little pore screen can deal with gigantic quantities of the issues related with the homogeneous hub groove wick. Since the screen effectively segregates the liquid and vapor streams, the entrainment of the liquid into the vapor stream by the interfacial shear is nearly wiped out. Similarly, this composite wick can be used in hostile gravity fields because the screen makes the required narrow weight.

## 2. LITERATURE REVIEW

Hong Seok ,Seongpilet al., 2018, "
modifying capillary pressure and boiling regime of micro-porous wicks textured with grapheme oxide". He introduced highly porous wicking surface to promote capillary driven flow in heatpipe.Micro-porous copper coated with graphite oxide (GO) increased the critical heat flux, capillary pressure and wicking effect. Small pores of GO increases the capillary pressure due to hydrophilic nature of GO coating, the contact angle between the



liquid and surface as decreased. Excessive GO coating decreases permeability which reduces pool-boiling performance.

- Yong tanget al., 2017 investigated some tests to increase the thermal performance of an ultra-thin heat pipe. The tests are chemical deposition, sintering, composition of surface after deposition and capillary rise experiments. Test results indicate that the capillary force of the deposited wick structures was larger than that of a normal wick.
- Yanping Huanget al., 2017 "Experimental study of heat transfer and start-up of loop heat pipe with multi scale porous wicks". He studied the effect of heat transfer and start-up of loop heat pipe with multi scale porous wicks and compared with conventional monoporous wicks, the composite multiscale porous shortened the start-up time, decreased the wall temperature, and suppressed the temperature instability of the LHP. A synergy between thermal conductivity and insulation was achieved, which ensured a high thermal conductivity for the primary layer wick and a good thermal insulation for the entire wick. This greatly reduced the heat leakage from the evaporator to the compensation chamber.
- Kumaresanet al., 2014 studied and compared the heat transfer performance characteristics of sintered wick and mesh wick heat pipes using CuO/DI water Nano fluids at various heat input and inclination angles. It is found that the heat transport capacity of sintered wick heat pipe is 14.3% more compared with mesh wick heat pipe under the same operating conditions. Similarly, a higher reduction in the surface temperature of 27.08% is observed for the sintered wick heat pipe with 1.0wt. % of CuO/DI water Nano fluids compared with

- mesh wick heat pipe. The inclination angle and weight percentage of CuO nanoparticles significantly influence the thermal performance of both the heat pipes. Optimum tilt angles of  $45^{\circ}$  and  $60^{\circ}$ respectively are observed for sintered wick and mesh wick heat pipes, whereas the optimum weight percentage is the same (1 wt.%) for both the cases. At these optimum reduction in conditions. resistance of 49.64% and 35.44% and an enhancement in the thermal conductivity of and 29.84% are respectively observed for both sintered wick and mesh wick heat pipes. Finally it is concluded that the thermal performance of sintered wick heat pipe is better than that of the mesh wick heat pipe.
- M.K Russellet al., 2011 studied the effect of orientation on the performance of U shape heat pipe parameters i.e. heat input and orientations. Thermal resistance of sintered wick heat pipe was found to be constant over its operating range for all orientations. In grooved wick heat pipe the thermal resistance in the inverted U orientation had a significantly higher thermal resistance.

#### 3. SCOPE AND OBJECTIVES

## 3.1 MOTIVATION TO THE PRESENT WORK

As electronic devices get smaller, architects and engineers are looked with the developing test of staying aware of the need to improve processing speed inside a shrinking form factor. Speedier processors require expanded power utilization, which creates heat; and smaller form factors require more noteworthy scaling down of the implementation used to scatter that heat. These contemplations are driving shrewd architects and engineers to think as



far as systemic solutions in which each thought in a devices power condition is inspected for most noteworthy improvement. As a rule, prerequisites of the power condition request that heat dissemination must be relative to the power dispersal of a given device. Power dissemination is the measure of power squandered by a device (i.e., power dissipation is reliant on the capacitance of the logic components, the operating voltage swing, and the operating frequency). Despite the fact that the processors in mobile phones, for instance, regularly utilize only a couple of hundred milliwatts of power, quite a bit of this is essentially lost to heat.

Customary strategies for cooling of electronic hardware incorporate enhancing the outline of printed circuit board, Using of thermal interface materials to fill the tiny air holes and utilizing of fans .These techniques are supplanted by coordination of heat pipes channels which gives productive cooling

### 3.2 METHODOLOGY:

- > Mass flow rate of fluid in condenser section water was used and now also used 0.01 kg/sec but in future will continue with 0.02, 0.03 kg/sec.
- > Heat input given in evaporator section and eight levels were used 50,100,150,200,250,300,350,400 watts.
- > Tilt angle of the heat pipe now were used 15° but in future will continue with four levels more 30°,45°,60°,90°

#### **EXPERIMENTAL INVESTIGATIONS**



Fig 4.1 Experimental setup of a heat pipe.

#### 4.1 EXPERIMENTAL SETUP:

The equipment is very versatile and can be used for checking performances of different heat pipes for by varying different parameters like heat input and flow rate with reference to different angle of inclination. Experimental setup consists of the following components

1. Evaporator 2. Condenser 3. Control unit 4. Thermocouple 5. Water sink/tank 6. Measuring jars and stop watch:

## **Experimental results:**

S.NO	Qin(W)	<u>Te</u> (°c)	<u>Tc(</u> °C)	Tout(°C)	Tin(°C)	Rth(°C/W	h(W/sq.m °C)	<b>N</b> =(Qout/ Qin)*100
1	50	44.4	30.7	28.8	28.9	0.274	366	30
2	100	55.5	32.1	29.6	29.5	0.234	429	49
3	150	69.9	32.4	30.3	29.9	0.25	400	46
4	200	86.7	32.8	30.6	30.4	0.2715	369	39
5	250	52.6	30.9	28.8	28.9	0.086	1154	57
6	300	62.9	32.3	30.2	29.7	0.102	982	78
7	350	62.5	35.8	35.8	35.7	0.0762	1313	80
8	400	75.9	36.1	36.0	35.9	0.0995	1007	95

Table-5 observation for At 150 angle and m-0.02 kg/sec

S.NO	Qin(W)	Te(°C)	Tc(°C)	Tout(°C)	Tin(°C)	Rth(°C/W	h(W/sq.m °C)	N=(Qout/ Qin)*100
1	50	31.1	30.8	26.5	26.1	0.045	4837	33.48
2	100	32.0	30.6	26.9	26.5	0.041	5823	25.02
3	150	33.2	31.2	27	26.7	0.040	6540	27.86
4	200	34.9	31	27.5	27.2	0.0396	6116	25.08
5	250	36	31.7	27.6	27.3	0.0392	5199	23.40
6	300	38.6	32.9	28.5	27.8	0.0398	4512	19.50
7	350	42.9	32.5	29.5	28.7	0.0399	3711	28.66
8	400	45.2	32.9	30.5	29.8	0.040	3000	20.9



#### Table-7 observation for At 600 angle and m-0.02 kg/sec

S.NO	Qin(W)	Te <b>(</b> °C)	Tc(°C)	Tout(°C)	Tin(°C)	Rth(°C/W	h(W/sq.m °C)	<b>П</b> =(Qout/ Qin)*100
1	50	32.1	27.9	28.4	28	0.044	2200	50.12
2	100	34.6	33.8	29.5	28.9	0.039	2700	41.5
3	150	35.5	34.9	30.3	29.4	0.030	3342	33.44
4	200	37.7	35.5	31.5	30.4	0.025	3828	3.44
5	250	39.5	37.8	31.6	30.8	0.024	4042	30
6	300	42.4	39.2	32	31.2	0.035	3500	27.86
7	350	46.5	42.2	32.5	31.4	0.045	2800	33.44
8	400	50.1	46.7	34.7	32.9	0.043	2050	37.62

#### Table-8 observation for At 900 angle and m-0.02 kg/sec

S.NO	Qin(W)	Te(°C)	Tc(°C)	Tout(°C)	Tin(°C)	Rth(°C/W	h(W/sq.m °C)	<b>n</b> =(Qout/ Qin)*100
1	50	31.8	29.5	28	26.9	0.045	2000	33.42
2	100	33.6	29.5	28.9	27.8	0.039	2400	24.09
3	150	36	30	29.3	28.9	0.040	2790	27.86
4	200	39.8	31.7	29.7	29.1	0.0382	3100	25.09
5	250	41.9	31.9	31.0	29.5	0.0393	3260	55.09
6	300	44.3	32	31.8	31.2	0.0398	2600	22.29
7	350	45.9	32.1	32.8	31.9	0.0399	2148	22.49
8	400	48.6	32.2	32.3	32.1	0.040	2000	20.09

#### Table-9 observation for At 150 angle and m-0.03 kg/sec

S.NO	Qin(W)	Te(°C)	Tc(°C)	Tout(°C)	Tin(°C)	Rth(°C/W	h(W/sq.m °C)	n=(Qout/ Qin)*100
1	50	32	31.2	29.4	29	0.026		50.18
2	100	33.5	31.9	29.6	29.3	0.016		50.17
3	150	34.3	32.8	30.2	29.7	0.009		50.17
4	200	35.4	32.7	30.6	29.9	0.015		43.90
5	250	36.9	32.7	30.9	30	0.018		40.15
6	300	39.4	33.6	31.7	30.8	0.019		37.20
7	350	42.3	33.7	31.9	30.9	0.023		46.60
8	400	46.8	35.4	32	31.6	0.029		18.80
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#### Table-10 observation for At 300 angle and m-0.03 kg/sec

S.NO	Qin(W)	Te(°C)	Tc(°C)	Tout(°C)	Tin(°C)	Rth(°C/W	h(W/sq.m °C)	N=(Qout/ Qin)*100
1	50	33.8	31.7	29.2	29	0.038	3000	50.15
2	100	34.8	31.8	29.6	29.5	0.031	3500	38.60
3	150	36	32.3	29.9	29.3	0.025	4200	41.9
4	200	39.3	33.2	30.6	30	0.023	4600	43.82
5	250	42.8	33.7	30.8	30	0.021	4830	40.12
6	300	45.5	34	31.9	30.9	0.030	3900	41.10
7	350	47.2	34.1	32.3	31.1	0.031	3200	39.40
8	400	49.6	34.5	32.4	31.2	0.036	2800	34.49
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Table-11 observation for At 600 angle and m-0.03 kg/sec

S.NO	Qin(W)	Te <b>(</b> °C)	Tc(°C)	Tout(°C)	Tin(°C)	Rth(°C/W	h(W/sq.m °C)	N=(Qout/ Qin)*100
1	50	32.6	30.4	29.8	29.5	0.040	2700	75.25
2	100	35	31	30.3	29.7	0.037	3000	62.7
3	150	35.8	31.5	30.8	30.5	0.029	3418	75.25
4	200	37.4	32.2	31	30.3	0.025	3973	43.90
5	250	38	32.9	31.4	30.1	0.024	4108	40.15
6	300	41.7	32.9	32.2	31.4	0.033	3200	38.60
7	350	46.5	32.3	32.5	31.6	0.039	2800	35.80
8	400	47.8	32.3	32.6	31.5	0.038	2500	35.5

Table-12 observation for At 900 angle and m-0.03 kg/sec

S.NO	Qin(W)	Te(°C)	Tc(°C)	Tout(°C)	Tin(°C)	Rth(°C/W	h(W/sq.m °C)	n=(Qout/ Qin)*100
1	50	32	29.9	29.7	29.5	0.041	2200	75.25
2	100	34.7	30.5	29.9	29.6	0.040	2500	50.14
3	150	36.4	30.7	30.4	29.9	0.038	2900	59.50
4	200	38.9	30.8	30.8	29.8	0.036	3200	58.52
5	250	40.2	31.5	31.7	30.4	0.034	3349	62.6
6	300	44	31.9	32.5	30.8	0.0396	2850	60.18
7	350	46.1	31.6	32.6	30.7	0.0398	2350	75.25
8	400	47.9	31.6	32.7	30.6	0.040	2050	71.65

### 4. CFD ANALYSIS

methods consist ofnumerical solutions of mass, Momentum and energy conservation with other equations like species transport. Two main stages comprise the solution of CFD problems. First, whole of fluid field divides to small elements that their names are mesh, and then partial differential equations that explain transport phenomena in fluid flow are used for these elements. Consequently, many non-linear equations appear which have to solve simultaneously. The solution of these equations accomplish with numerical algorithm methods. and Conservation equations for compressible flow

#### **4.1 MATERIALS OF HEAT PIPES:**

➤ Working fluid : Ethanol

> Container material : Copper

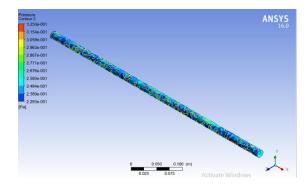
- > Type of wick : Composite wick (sintered powder and screen mesh of 250 mesh number)
- ➤ Properties of Ethanol : Melting point at (-112°C) Boiling point at Atm pressure(78°C)

Useful range (0 to 130°C)

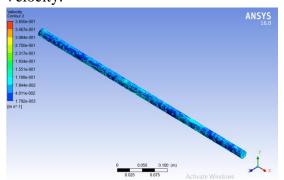
#### Temperature:



#### Pressure:



## Velocity:



#### **CONCLUSION:**

Under this research effect of heat pipe processing parameters, the mass flow rate (m), angle of inclination ( $\theta$ ) of the coolant and the thermal input force (Q) on the thermal performance of the helical grooved heat pipe were studied experimentally.

- 1. The heat transfer coefficient increases with the increase of heat input, increase in mass flow rate and inclination of the angle. The increased heat input heat transfer coefficient increases to a certain point because more water evaporates, thereby efficiently removing heat through phase changes, further decreasing rapidly due to further increase of heat input produces more interfacial resistance between vapor and condensate flow and will result in insufficient evaporation section from condensate.
- 3. With the use of tilt angle, heat pipe performance decreases
- 4. As the mass flow of water increases, the performance of the heat pipe also increases.

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35