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Design and Development of Phase Change Material Cascade Heat Sink

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Abstract: This research article deals with experimental study of three-stage cascade heat pipe for industrial applications. The design and experimental analysis is used for selection of suitable materials, phase change material as working fluid for cascade heat pipe. Phase change material cascade heat pipe was designed and experiments were conducted with three different heat inputs like 40, 50 and 60W. The temperature of first, second and third stage fluid temperatures, the evaporator, condenser temperature are measured by suing Agilent Data Logger. Experiments results showed that steady state temperature is attained at each stage after 35 seconds. The results are used to predict the heat transfer of PCM heat pipe. Temperature Data logger was used to acquire the data of temperature and pressure of three stages by using Piezo electric pressure transducer. This three stage PCM heat pipe was suitable for the range of 100 - 350 °C and Merit number of naphthalene is 241.1 x 10^6. The latent heat of fusion of naphthalene is compared to commonly used fluids. The vapour pressure at high temperature is relatively low as compared to water which allows the operating temperature to go beyond the operating temperature of the water. Based on this experimental research, this cascade heat pipe working range was suitable for industrial applications.
Keywords: Heat Transfer Analysis, Three fluid Heat Pipe, cascade system, Design of heat pipe

I. INTRODUCTION

Heat pipe is broadly used for different energy storage device and many computer processers like micro processes, electronics devices, electrical drives, solar energy storage device, super capacitor and ultra-capacitors. I. A. Mhaisne et al. [1] carried out experimental Investigation of Copper Sintered Heat Pipe Flow For Power Electronic Cooling by Vortex and suggested that suitable design procedure are required for advanced heat pipe. Hence, the design and analysis of heat pipes are required to some new advanced applications namely capacitor cooling. Increasing or decreasing the temperature of any electronic device depends on heat generation due to energy storage and energy consumption. K. N. Shukla et al. [2] investigated the thermal performance of heat pipe with suspended nano-particles and the authors suggested that prediction of suitable nano - particle is required for heat pipes. Hence, cooling devices are highly required for maintaining the system temperature and satisfy the heat transfer performance of heat pipes. R. Hari and C. Muraleedharan, [3] carried out the analysis of effect of heat pipe parameters in minimising the entropy generation rate. In addition to that, the heat load varied with respect to energy consumption or energy storage. However, previous heat pipe was not suitable for the variation of heat load with respect to given design values. S. Lips et al. [4] carried out overview of heat pipe studies and suggested that design procedure are required to improve the performance of heat pipe. Hence, this research focused to design the suitable cascade heat pipe for variable heat load input, energy consumption of energy storage device. R. Andrzejczyk investigated [5] the experimental investigation of the thermal performance of a wickless heat pipe operating with different fluids namely water, ethanol, and SES36. Analysis of influences of instability processes at working operation parameters. Based on the requirements, the following four processes are mandatory for selection of suitable materials for heat pipe, fluid property, and geometry parameters. They are : 1) Selection of heat pipe materials like the container, 2) selection of Tube materials, 3) selection of wick materials for working fluid, 4) selection of suitable materials, volume of condenser, evaporator and predict the performance limits. M. Narcy et al. [6] carried out experimental investigation of a confined flat two-phase thermosyphon for electronics cooling and found that suitable fluid and design parameters are required to improve the performance. P. K. Jain [7] investigated the influence of different parameters on heat pipe performance. In the case of heat, pipe compatibility may limit the choice of working fluids or in the case of materials are first specified. M. Ramezanizadeh et al. [8] carried out experimental and numerical analysis of a nanofluidic thermosyphon heat exchanger. Two more aspects are to be considered in the case of heat pipes in the amount of fluid charge and the start-up of the heat pipe [8]. M. Sandeep et al. [9] carried out analysis and fabrication of heat pipe and thermosyphon and author found that design procedure are required for improving the performance of heat transfer. B. M. Jibhakate and M. Basavaraj [10] investigated the study of parameters affecting the thermal performance of heat pipe. One way of overcoming this problem is to provide an excess fluid reservoir, which will absorb the excess fluid that is not required by the primary wick structure.

W. W. Wits and G. J. te Riele [11] discussed that modelling and performance of heat pipes with long evaporator sections. In the case of arterial, wicks are necessary to ensure that in the case of an artery becomes depleted of working fluid, it should be able to refill automatically which is called priming[6]. J. Jose and R. Baby [13] carried out Recent advances in loop heat pipes. Based on the literature and requirement of design of heat pipe is satisfied for developing the design methodology for cascade heat pipe.

II. DESIGN METHODOLOGY

The concept of design of cascade heat pipe is important for selection of suitable devices and procedures as shown in figure 1. The design procedure was followed systematically and required materials, fluids property are shown in tables 1 and 2. Present researches have many intensive investigations and resulted in rapid development and commercial production and use of heat pipes for different applications. The next step is the selection of a set of working fluid and predicts the suitable for the operating temperature. With reference to the fluid chosen the operating materials and wick, materials and type are chosen. The best of the working fluid compatible with these materials is decided for given applications P. Z. Shi et al .carried out design and performance optimization of miniature heat pipes. The type of wick capable of providing of the required capability head is to be selected.

If the requirement is not met, the cycle is repeated until the requirement is satisfied. Next, the temperature drop between the condenser and evaporator is predicted for the maximum allowable value. A.Chaudhari et al. investigated that the effect of wick microstructures on heat pipe performance. J. V. Suresh et al. investigated the effect of working fluid on thermal performance of closed loop pulsating heat pipe and the detail design is taken up once these conditioned are satisfied. C. E. Andraka et al. [18] carried out the high performance felt-metal-wick heat pipe for solar receivers. Based on the literature any limitation on weight, temperature drop between evaporator and condenser will also specified.

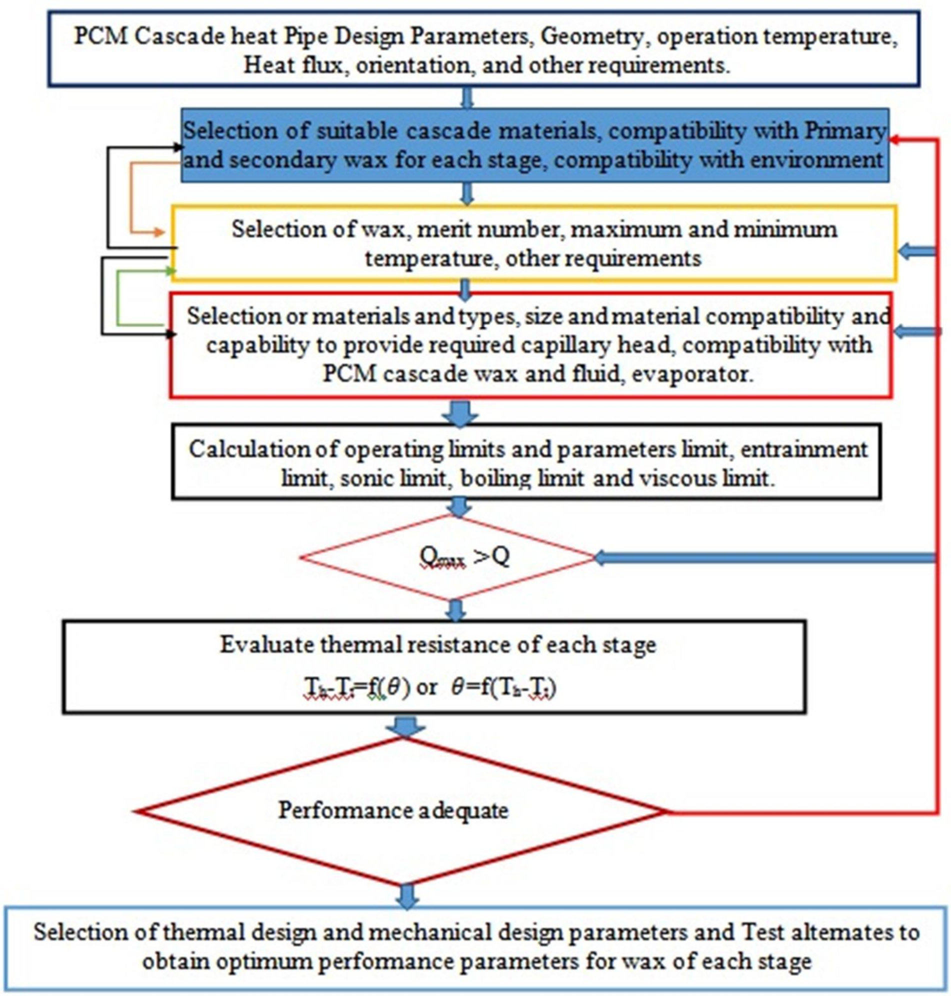

Fig. 1 Design methodology for cascade heat pipe

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Design of cascade heat pipe is based on the some important steps are involved they are: The maximum diameter of the heat Pipe is calculated by the below procedure and all the parameters are predicted. Where d_a is the maximum diameter of the artery, h- is the vertical height to the base of the artery, θ- is the contact angle of the fluid, σ_l - is the surface tension of the fluid, ρ_l- and ρ_v- are the densities of the liquid and vapour. Heat Pipe design procedure can be easily understood by flow chart shown in Figure 2.

The following parameters and detail of the specification of heat pipe, which include the geometry, operating temperature, heat load, orientation and other details like where it is to be used as in space are determined. The temperature range of working fluid is specified and the sonic limit for the fluids is determined by using the equation for maximum heat flux (Q_{max}). In case the heat flux is larger than the required heat flux, the fluid can be chosen. The maximum heat flux (Q_{max}) without entrainment is checked [17].

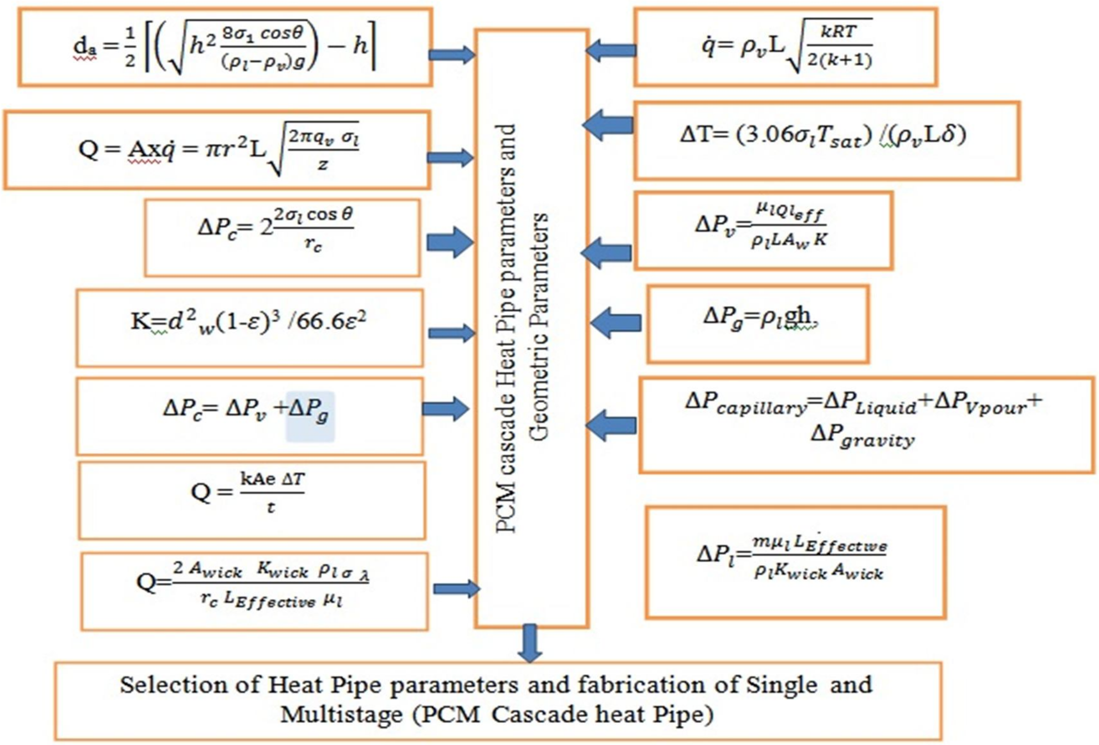

Fig. 2 Heat pipe design procedure

Z is the characteristic dimension of the liquid and vapour interface having value of about 0.16; δ is thermal layer, having a value of $15\mu m$. It is required to predict the Q and equal to compare with Q_{max} or higher than the heat load of the pipe to be designed, than the fluids pass this limit[6]. In addition to that working limit is checked by using the Merit Number shown in Table 1. To avoid nucleation in the wick, the super heat may be obtained and ΔT is calculated. The fluid having the highest value for ΔT is the most suitable one for the application. Priming Factor for the fluid is determined from plots of Temperature verses, priming factor. The fluid having higher value of priming factor should be considered if it satisfies the other requirements.

The weight and vapour pressure of the fluid dictate the container thickness and weight. A final choice of working fluid selection is based on the above parameters and other specification requirements. The type of wick is selected based on the some parameters, and develop the homogeneous wick and arterial wick.

The capillary pressure (ΔP_c) is calculated using design procedure. r_c is the wick radius and θ is the contact angle equal to zero for wetting fluids. The pipe will work, if this pressure equals the sum of pressure drop for vapour flow and gravitational pressure. Predicting the vapour pressure is required for vapour flow which is calculated by the equation shown Figure 2. A_w is the wick area and K is calculated, l_{eff} is the length of pipe = ½ (Length of evaporator + length of condenser).

$$K = d^2{}_w(1-\varepsilon)^3 / 66.6\varepsilon^2 \quad (1)$$

d_w is the wick diameter, ε is the volume fraction of solid phase in the wick. The values are substituted in the equation (1). The pressure required for movement against gravity is given as $\Delta P_g = \rho_l gh$, where h is the height of movement against gravity. The P_c is calculated by using the equation (2)

$$\Delta P_c = \Delta P_v + \Delta P_g \quad (2)$$

In case, A_w is larger than the pipe diameter. The equation for heat flow across the wick is used and determines the thickness of the wick. Heat transfer (Q) to the evaporator is predicted by the equation. t (m) is the thickness Q is the heat load, A_e is the evaporator area ΔT is the allowable temperature drop. The conductivity k is the combined conductivities of the sold material in the wick and the working fluid in the wick. The maximum value of pressure drop is calculated from the basic equation ΔP_c = sum of other pressure drops. The number of arteries is also decided using these equations.

The design of the heat pipe requires a large number of data about properties of the liquid and the wick. These are not readily available and one has to search through the literature and handbooks for the same. The amount of power that a heat pipe can carry is governed by the lowest heat pipe limit at a given temperature.

For a given heat pipe, the Merit number ranks the maximum heat pipe power when the heat pipe is capillary limited[6]. (The capillary limit generally controls the power in the mid-range, while other limits control at higher and lower temperatures).The capillary limit is reached when the sum of the liquid, vapour, and gravitational pressure drops is equal to the capillary pumping capability and predicted by using equation .The merit number neglects the vapour and gravitational pressure drops and assume that the capillary pumping capability is equal to liquid pressure drop.

The liquid pressure drop in a heat pipe was calculated by using given equation shown in Figure 2. ΔP_l is the Liquid pressure drop assumed equal to the wick pumping capability, $E_{effective}$, is the effective length, K_{wick} is the wick permeability, A_{wick} is the wick area, the mass flow rate is the heat transfer rate divided by the $\dot{m}=\frac{Q}{\lambda}$, The wick pumping capability is $\Delta P_l=\frac{2.\sigma}{r_c}$, Where r_c is the pore radius.

Combining the above the three equations and solving for Q, the maximum heat transfer when only the liquid pressure drop is considered by using this equation. Based on the design procedure, method of calculations and experiments comments, required materials are chosen.

TABLE I

Operating Temperature, Operating Maximum Temperature Working Fluid and Envelop Materials for Heat Pipe

Operating Minimum Temperature (°C)	Operating Maximum Temperature (°C)	Working Fluid	Envelope Materials	Comments
20	280, short term to 300	Water	Copper, Monel, Nickel, Titanium	Aluminium, steels, stainless steels and nickel are not compatible
100	350	Naphthalene	Al, Steel, Stainless Steel, Titanium, Cu-Ni	380°C for short term. Freezes at 80°C
200	300, short term to 350	Dowtherm A/Therminol VP	Al, Steel, Stainless Steel, Titanium	Incompatible with Copper and Cu-Ni
200	400	$AlBr_3$	Hastelloys	Aluminium is not compatible. Freezes at 100°C

The design and materials selection are done by using design methodology. Some conflicts mainly depend on material deterioration of one of component and other component parameter selection procedure. Compatibility with wick and container materials, good thermal stability, wet ability of wick and wall surface, suitable vapour pressure not too high or too low, large latent heat, good thermal conductivity, high surface tension, low viscosity of liquid as well as vapour and acceptable freezing and pour points are predicted and list of working fluid as shown in Table 2.

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TABLE III
List of Working Fluids Boiling Point, Useful Range and Merit Number for Heat Pipes

S.No	Description fo fluids	Boiling point (^{0}C)	Useful range (^{0}C)	Merit number (W/m^2)
1	Helium	-261	-271 to -296	170.5x10^3
2	Nitrogen	-196	-271 to -160	74.8x10^6
3	Ammonia	-33	-60 to 100	449.8x10^6
4	Pentane	28	-20 to 120	134.7x10^6
5	Acetone	57	0 to 120	364.6x10^6
6	Naphthalene	218	100 to 350	241.1x10^6
7	Ethanol	78	0 to 120	166.3x10^6
8	Heptane	98	0 to 150	123.8x10^6
9	Methanol	64	10 to 130	377.9x10^6
10	Flutec PP2	76	10 to 160	17.7x10^6
11	Water	100	30 to 200	4.26x10^9

Source: https://www.1-act.com/merit-number-and-fluid-selection

Heat pipe require to considering a single property, fluids are evaluated by a combination of properties influencing the working of heat pipes called Merit number, it is defined as $N_l = \frac{\sigma_l L \rho_l}{\mu_l}$ W/m^2, where σ, ρ, μ and L are the surface tension, density, viscosity and latent heat of the fluid. The dimension of merit number is in W/m$^{2.}$ Effectively, merit number indicates the heat transport capacity of the fluid[18]. The sectional area required for transporting a given thermal load is directly proportional to the merit number. The merit number as function of temperature is shown in Figure 2 for a number of typical heat pipe working fluids[18][13]. Its merit number is approximately equal 10 times higher than everything else except the liquid metal is and it is meaning that, it will carry ten times more power than other working fluids.

III.EXPERIMENTAL LAYOUT

The experimental layout is shown in Figure 3. They are 1) Working fluid, 2) Wick and capillary structure and container. As cascade thermo siphon heat pipes are required to work at various temperatures from 4 K to 2300 K, different fluids are required for different temperature bands. For example, helium was used in the low temperature range of 4 K.

Liquid oxygen and nitrogen are used in the next band. Water, Ammonia and alcohols were used in the range of 300 to 500K. Special materials are required in the range of 600 to 600k. Above this temperature sodium, lithium and similar materials are used. More than two dozen fluids are in use as working fluid. Similarly, the arrangement and material for wick also vary as per the applications. In order to be compatible with the various fluids more than a dozen materials are use. The above difficulty always arises in the selection of suitable combination of material and fluid.

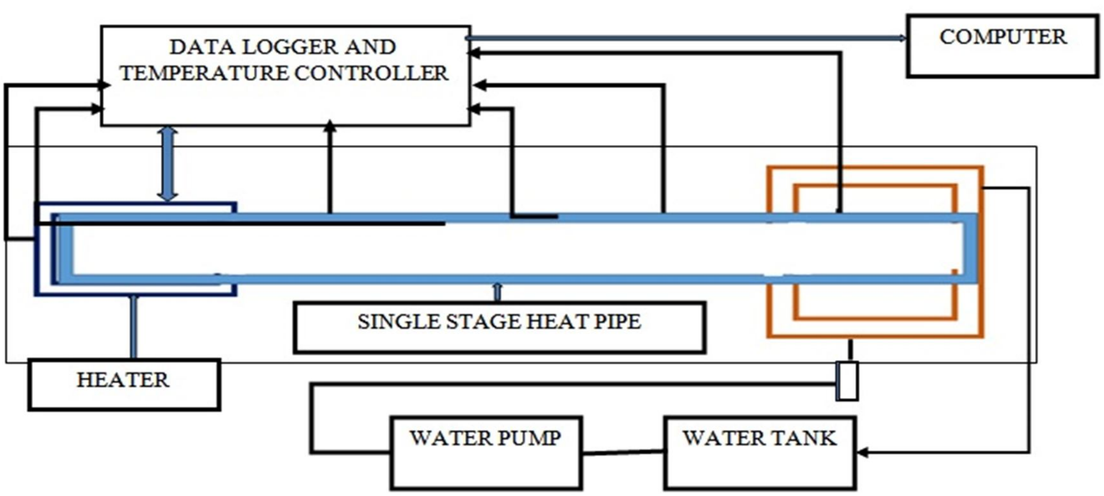

Fig. 3 Layout of cascade heat pipe setup

The schematic diagram of the single stage heat pipe experiment setup is shown in Figure 3. The components in the experimental setup are evaporator and condenser. The first stage heat pipe condenser is used to heat input of the second heat pipe and second heat pipe condensed is used to third stage heat input. The components in the experimental setup consist of a primary heat pipe stage 1 and the second stage heat pipe is inserted in to the first stage. In the setup a heater for the experimental purpose, in which a stable watt is supplied to the system with a regulator by which amps, replaces the electrical component and volt input is controlled. This heater is placed at the bottom of the primary reservoir.

TABLE IIIII
Specification of Heat Pipe

SL.No	Description	Values(mm)	Area mm2	Exp 1 – Fluid 1
1	3 stage cascade heat pipe length	1000	28260	Water
SL.No	Description	Stage 1	Stage 2	Stage 3
1	Length of Heat Pipe	400	400	400
2	Evaporator length	100	100	100
3	Adiabatic section	200	200	200
4	Condenser	100	100	100
5	OD of heat Pipe	30	20	14
6	ID of the Heat Pipe	26	16	10
7	Wick Type Screen Mesh(4 layer)	50/cm	50/cm	50/cm
8	Merit Number (W/m2)	241.1x10^6	241.1x10^6	241.1x10^6
9	Boiling Point deg C	218	218	218
9	Mesh wire diameter	0.08 mm	0.08 mm	0.08 mm

The primary reservoir consist of distilled water as a working fluid and the secondary reservoir consist of secondary working fluid The heater of 1000W is attached to bottom of the primary reservoir of area 65mm*80 mm 192.308*10^3 w/m^2 is the maximum amount of heat flux can be provided to the system. The heater is attached to the system by clamping with the primary reservoir.

Heat pipe was experiment was focused to two phase, first one was single stage with variable heat load and second one was multistage heat pipe (Cascade heat pipe) with variable heat load conditions. A 1000 mm long, 30 mm diameter copper tube was used to fabricate single stage heat pipe, and both ends were sealed with end caps. One end cap was provided by filling tube for charging the working fluids.

Wick section was made by A 4-layer 50/cm mesh copper screen was fixed on the inner tube which was held in a fixed positions by a guide setup. The heat pipe was evacuated by using the vacuum pump and 10^{-4} bar pressure was maintained at 120 C for about 4 hours to remove the non –condensable present in the tube. Then the heat pipe was cooled by applying ice and the working fluid of desired quantity (210 ml) was injected through a capillary tube by adjusting the vacuum valve which is present (7.38 kPa).

The capillary tube was then crimped and sealed. The evaporator, adiabatic and condenser sections are of length 100, 600, and 300 mm respectively. The heat pipe adiabatic section was maintained by isothermal boundary conditions and ensured by using glass wool insulation and increased heat transfer capacity of the heat pipe with augmented by 40 numbers (70x70) of flat fins each of 0.6 mm thickness mounted on the condenser section by bracing. The same procedure was followed and fabricated cascade (Multi stage) heat pipe and three stages was fabricated, tested. The specification of single and multi-stage the heat pipe is presented in table 1.

The experimental setup consists of resistance heater of 1500W power output; wattmeter and an auto transfer were providing the necessary power supply to the heaters. The national instruments (NI) based system was used to recorded the thermocouple readings at different positions of the heat pipe. Thermo-couples of K-type (10 numbers) were used to measure the temperature response at the different position of the axial distance. The layout of thermocouples is presented in Fig 3.

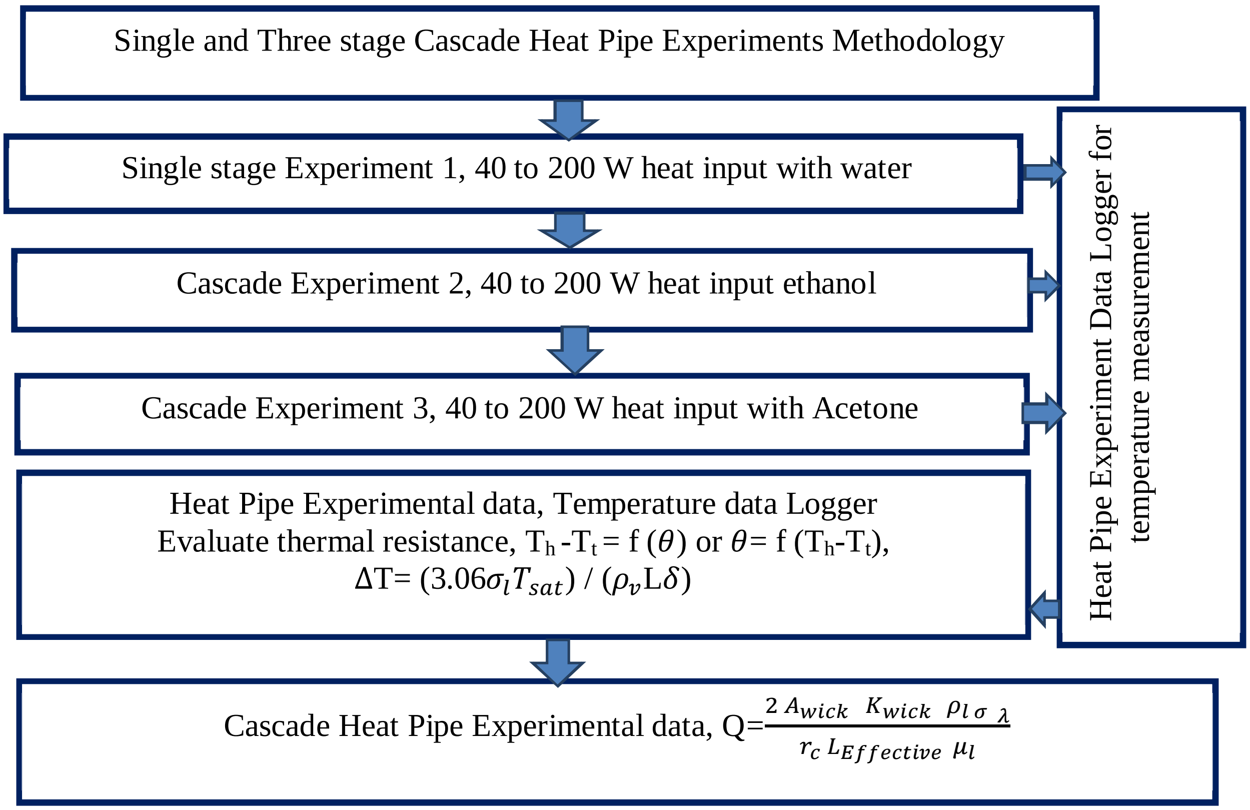

Fig. 4 Single and three stage cascade heat pipe methodology

IV. EXPERIMENTAL PROCEDURE

The evaporator, adiabatic and condenser sections are of length 100, 600, and 300 mm respectively. The heat pipe adiabatic section was maintained by isothermal boundary conditions and ensured by using glass wool insulation and increased heat transfer capacity of the heat pipe with augmented by 40 numbers (70x70) of flat fins each of 0.6 mm thickness mounted on the condenser section by bracing. The same procedure was followed and fabricated cascade (Multi stage) heat pipe and three stages was fabricated, tested, and shown in Figure 2.

The specification of single and multi-stage the heat pipe is presented in table 1. The experimental setup consists of resistance heater of 1500W power output; wattmeter and an auto transfer were providing the necessary power supply to the heaters. The national instruments (NI) based system was used to recorded the thermocouple readings at different positions of the heat pipe. Thermo - couples of K - type (10 numbers) were used to measure the temperature response at the different position of the axial distance.

The inlet and outlet temperatures of the cooling water were measured by using two-T type thermocouple and integrated with data logger. The mass flow rating of cooling water at steady state was measured by using control valve. The heater and the adiabatic sections were insulated with 1.0 cm thick glass wool and then the power supply was controlled by autotransformer. The heat input is varied by using the variable transformer from 60 to 300 W.

The inlet and outlet temperature of the cooling water and the temperature of heat pipe were monitored by using the data acquisition system. The mass flow rate of water was measured when the heat pipe operates under steady state. The same procedure was used to conduct the experiment three different fluids and results are used to analysis. The cascade heat pipe experiment was conducted with stage 1, 2 and 3 fluids like water, ethanol and acetone.

V. RESULTS AND DISCUSSIONS

Thermal resistance and overall heat transfer coefficient the effect of nano particles on heat pipe performance can be explained by calculating the variation of thermal resistance. Thermal resistance of the heat pipe can be calculated by R=$\Delta T/Q$, Where ΔT is the temperature difference between the evaporator wall and condenser wall.

Expressions for calculating thermal resistance for different sections of the heat pipe are presented in table 3. The variation of thermal resistance of the heat pipe filled with and the copper nano-particles suspension are presenting in Fig. 5.

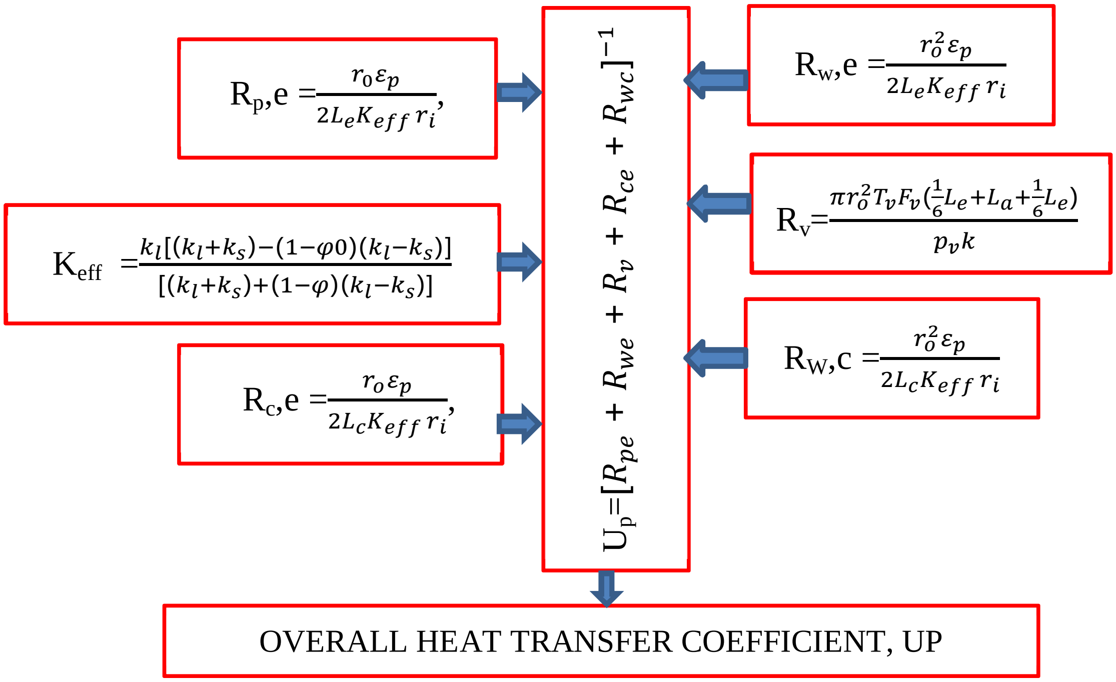

Fig. 5 Overall heat Transfer Coefficient calculation procedure for single and multistage (cascade system) heat Pipe

The trend of thermal resistance shows that the thermal resistance of heat pipe decreases with the increases of heat the heat input. For 0.01 volume % of copper nano particles, the thermal resistance was reduced by 32.10%. The experimental results showed that lower thermal resistance for higher heat input by using The Nano fluid in the heat pipe. The reduction in thermal resistance is due to the activation of large number of nucleation sites in the evaporator section, which extends the regime of nucleate boiling to vary high heat fluxes.

The Error estimates are very important analysis for the experimental part and influence of results and analysis. The following main sources of experimental uncertainty were the temperature measurement, namely coolant flow rate measurement and the wattmeter. This error analysis was done and tabulated for the accuracy of flow measurement is around ±2.5 % and the accuracy of the thermocouple is around ± 5% ^{0}C.

The maximum uncertainty of the wattmeter is around ±1.5., and the error analysis is followed by the method was described by Holman. In addition to uncertainties of the heat flux and the heat transfer co efficient was predicted and tabulated as shown in Table 4 and the uncertainties calculated by the equations 3, 4:

$$q_c = \sqrt{\left(\frac{\partial q}{\partial Q} w_q\right)^2 + \left(\frac{\partial q}{\partial D} w_D\right)^2 + \left(\frac{\partial q}{\partial L_c} W_{l_c}\right)^2} - \quad (3)$$

$$h_c = \sqrt{\left(\frac{\partial h_c}{\partial q_c} w_{q_c}\right)^2 + \left(\frac{\partial h_c}{\partial \Delta T} w_{\Delta T}\right)^2} \quad (4)$$

The equations shows that uncertainties within a reasonable limit of 5 - 6%.

The symbols $w_q, w_D, W_{l_c}, w_{q_c}, w_{\Delta T}$ are the uncertainties in the heat flow rate, diameter, condenser-length condenser-heat flux and temperature drops respectively.

TABLE IVV
Uncertainty of Heat Flux and Heat Transfer Coefficient of Single Stage and Cascade Heat Pipe

Heat input	Heat pipe with water		Heat pipe with Naphthalene Cascade heat pipe	
	Uncertainty of Heat Flux single stage (%)	Uncertainty of heat transfer coefficient single stage (%)	Uncertainty of Heat Flux cascade system (%)	Uncertainty of heat transfer coefficient cascade system (%)
100	6.01	6.34	6.22	6.51
150	6.0	6.11	6.34	6.39
200	5.3	5.14	6.03	6.24
250	5.89	5.96	6.07	6.35
275	5.92	6.01	6.24	6.39

The above the procedure was used to calculate the overall heat transfer coefficient shown in Figure 5.

$$U_p = [R_{pe} + R_{we} + R_v + R_{ce} + R_{wc}]^{-1} \quad (5)$$

Heat Transfer in the condenser section is an additional evaluation of the heat transfer performance of the heat pipe, the heat transfer coefficient at the condenser section was calculated with base fluid for single stage and multistage . The experimental heat transfer coefficient at the condenser section of single and multi-stage can be calculated as given below $q_c = \frac{Q_{out}}{\pi d l_c}$, $Q_{out} = mC_p\Delta T$, Where m is the mass of Water flow , C_p- is the specific heat of the water ΔT is the temperature difference of water flow. The same procedure was used to find cascade system. Single Stage heat output was calculated by using the equation

$$Q_c = (Q_{out1} = mC_p\Delta T) \quad (6)$$

$$\text{Cascade system} - Q_{cascade} = (Q_{out1} = mC_p\Delta T) + (Q_{out2} = mC_p\Delta T) + (Q_{out3} = mC_p\Delta T) \quad (7)$$

Q_{out1}- is the heat transfer at stage 1, Q_{out2}- is the heat transfer at stage 2, Q_{out3}- is the heat transfer stage 3.

Total Heat Transfer is $Q_{out} = (Q_{out1} + Q_{out2} + Q_{out3})$.

Figure 5 shows the condensing heat transfer coefficient enhancement due to the addition with two stages (multistage) with two different fluids like ethanol and acetone .From the results it was observed that, the heat transfer co efficient of the heat pipe changed with multistage and compared to single stage heat pipe. It is believed that the heat transfer enhancement in the evaporator and condenser section is mainly dependent on the nature of surface created by the multi stage as convection area was increased. Heat transfer coefficient in the condenser section depends on the thickness of the liquid layer and the hydrodynamic properties of the working fluid. A correlation can be established between the Nusselt number, Reynolds number and Prandtl number

$$N_u = \frac{(hC_l)}{K} = 1.12 R_e^{0.8} p_r^{0.7} \quad (8)$$

The correlation may be compared with the existing correlation for condensing water vapour in thermo siphon as a function of Reynolds number, sec Figure

$$N_u = 5.03 Re1^{/3} P_r^{1/3} \quad (9)$$

Figure Shows the Nusselt number variation with respect to the Reynolds number for constant Prandtl number. The trend shows an enhancement in the Nusselt number with the increase in the Prandtl number and Reynolds number, which means that the increase in Prandtl number leads to enhancement of Nusselt number.

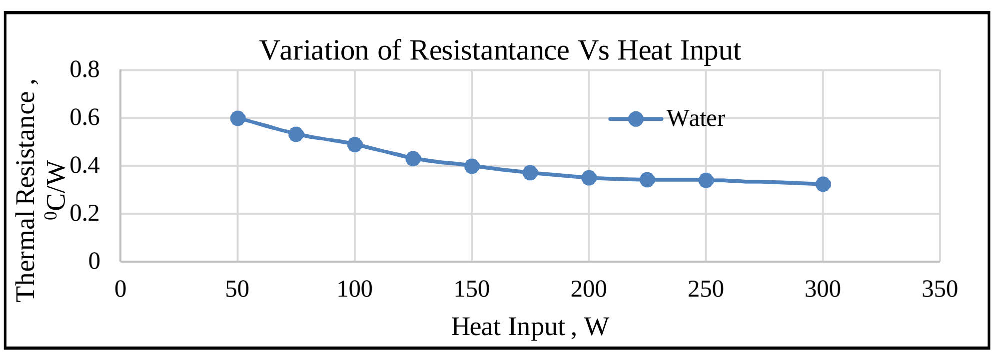

Fig. 6 Variation of Resistance with respect to Heat Input

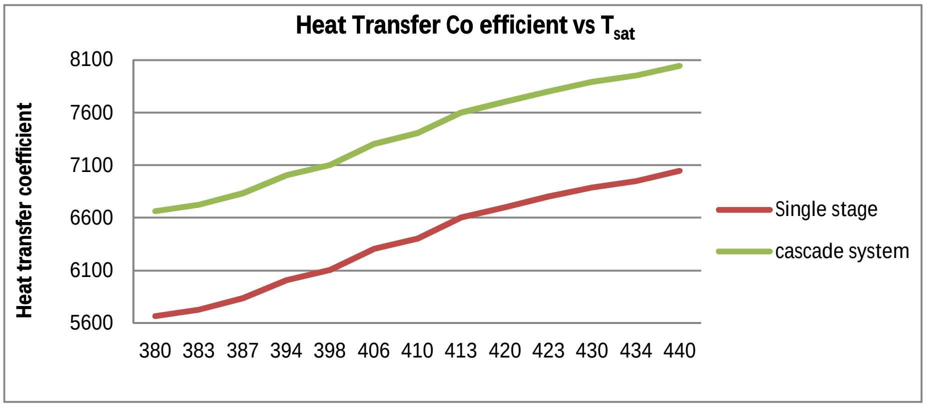

Fig. 7 Variation of Heat Transfer coefficient vs T_{sat}

It can be seen from the Figure, Variation of heat transfer coefficient with T_{sat} (K) shows the figure 7. In can be clearly indicated that, heat transfer coefficient increase when temperature increases. Cascade heat pipe heat transfer coefficient is 18 to 19.5 % higher compared to single stage heat pipe. Experiments were performed to study the heat transfer characteristics in the evaporator and condenser section of the heat pipe. The heat transfer in the condenser section depends on the Reynolds number and the Prandtl number. The both single and cascade heat pipe in side fluid velocity and condensate liquid and the hydrodynamic properties are very important role in condensing heat transfer.

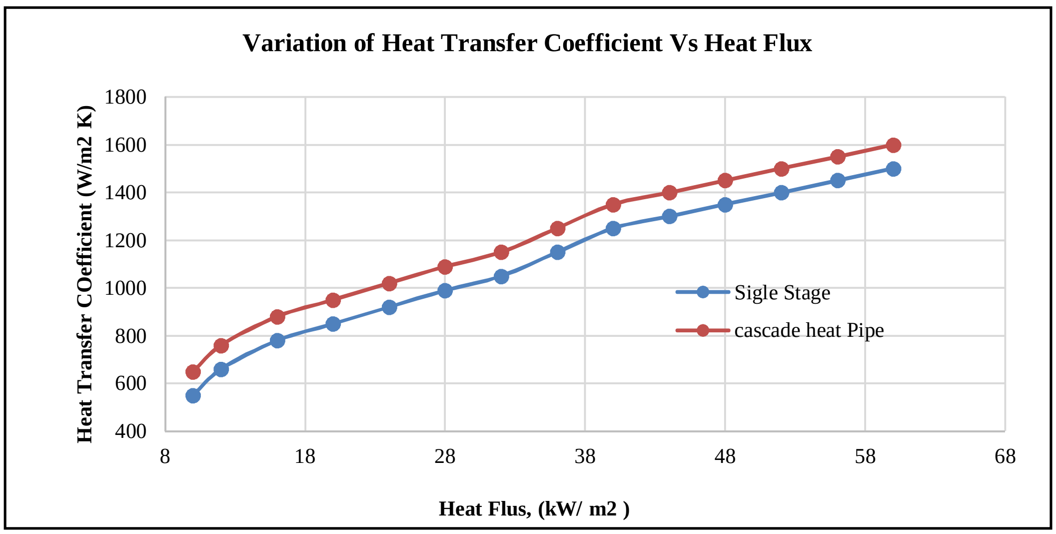

Fig. 8 Variation of Heat Transfer Coefficient Vs Heat Flux

The variation of heat Transfer coefficient with Heat flux is shown in Figure 8. Single and multistage heat pipe experiments were performed to study the heat transfer characteristics in the evaporator and condenser section of the heat pipe shown in Figure 4.4. Heat transfer coefficient increases when heat flux increases from 8 - 68 kW/m^2. The overall heat transfer coefficient of the system increases gradually with increase in heat flux or the saturation temperature maximum of 23.10 % and heat transfer coefficient increase with 16% compared to single stage due to decrease the thermal resistance of the heat pipe.

VI. CONCLUSION

Experiments were performed to study the heat transfer characteristics of evaporator and condenser section of both single and cascade heat pipe. The heat transfer in the condenser section of single stage with water fluid and ethanol fluid heat pipe depends on the Reynolds number and the Prandtl number. Single stage heat pipe heat transfer is 18% low compared to cascade heat pipe due to two low boiling point fluid used in second and third stages. Based on the experimental investigation, convection area of cascade is increased as well as the heat transfer rate was increased.

VII. NOMENCLATURE

TABLE V
Symbols, Abbreviations and Nomenclature

SYMBOLS, ABBREVIATIONS AND NOMENCLATURE	CONTENT
G	Acceleration due to gravity
∞	Ambient
Avg	Average
H	Heat Transfer Co Efficient
B	Bottom wall
T_m	Bulk mean temperature
B	Co-efficient of volumetric expansion
Q_{sides}	Convective heat loss
Q_{top}	Convective heat loss
T_{max}	Dimensionless temperature
exp	Experimental
FN	Figure of Merit
f	Fluid
Gr	Grashof number
q_{pri}	Grashof number
q_{sec}	Heat
q”	Heat flux (W/m^2)
Q_{in}	Heat input (W)
T_{heater}	Heater temperature (^{o} C)
T_{water}	Water temperature (^{o} C)
v	Kinematic viscosity (m^2/s)
N	Number of secondary tubes arranged
Nu	Nusselt number (hL/k / hd/k)
Pr	Prandtl number (ν/α)
pri	Primary coolant
Ra	Rayleigh number
sec	Secondary coolant
A_h	Surface area of the heater (m^2)
A_1	Surface area of the primary tank (m^2)
A_2	Surface area of the secondary tubes
T_s	Surface temperature
ΔT	Temperature difference surface (^{o} C)
T	Temperature excess (^{o} C)
Kf	Thermal conductivity of the fluid (W/m-K)
α	Thermal diffusivity of the fluid (m^2/s)
R_{th}	Thermal resistance (^{o} C/W)
V	Voltage applied to the heater (V)

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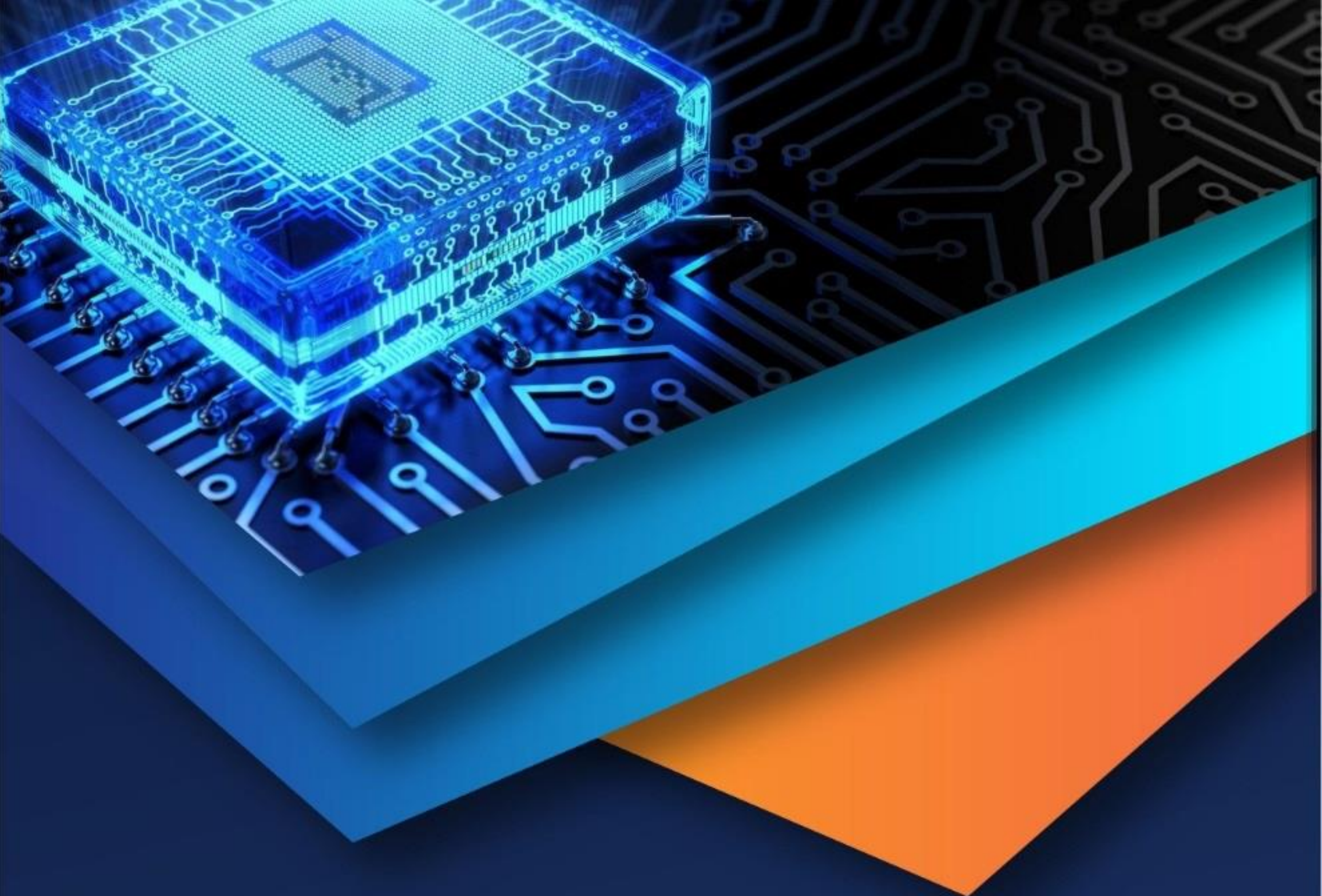

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