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Inline Array Jet Impingement Cooling Using Al_2O_3 / Water Nanofluid In A Plate Finned Electronic Heat Sink

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Abstract: - Jet impingement cooling is a technique used for cooling the electronic systems. In this work, heat transfer and pressure drop characteristics of deionized water and Al_2O_3/water nanofluid in an electronic heat sink having aluminium plate fins and provision for jet impingement cooling have been studied. A novel heat sink contains two rows of plate fins of size 29mm x 24mm x 0.56mm. A thin plate having 110 holes of diameter 2.5 mm is used to produce number of jets. The plate is kept inside the heat sink in such a way that H/d_n is 5.2 mm and adjacent jet spacing is 2mm. The overall dimension of the heat sink is 60x60x 65 mm. For this work we prepared a Al_2O_3/water nanofluid by dispersing specified quantity of nanoparticles in to deionized water by using a ultrasonic bath. Experiments were conducted under constant heat flux condition and the volume flow rate of the fluid was in the range of 1.315 to 2.778. It is found from the results that the nanofluid removes heat better than water in the jet impingement cooling with very low rise in pressure drop.

Keywords**: -** *Jet impingement, Nanofluid, Heat transfer, Pressure drop, Heat sink*

I. INTRODUCTION

The electronic systems of today are embedded with large number of minute circuits which makes it more compact and faster in processing speed. Thus it consumes a large amount of power, as a result the heat flux generated is high. These heat fluxes should be dissipated in a proper manner completely as fast as possible. Otherwise the heat thus generated will harm and affect the performance of the system and even may damage it.

Conventionally we use forced air and water cooling technique, but this cannot satisfy the increasing cooling requirement in the future. So in order to increase the cooling requirement and miniature the heat exchangers research on nanofluid is still going on from the past decade. The reason behind to keep an eye on this investigation is that it is the next generation heat transfer fluid. The heat transfer capabilities of these nanofluids are higher than that of their base fluid.

Nanofluid is a fluid containing nano sized particles known as nano particle. These fluids are engineered colloidal suspensions of nano particles in a base fluid. The nano particle is defined as a smallest object that behaves as a whole unit in terms of transport and properties. Nano fluids have novel properties that make them potentially useful in many application in heat transfer, including micro electronics fuel cells, pharmaceutical processes, hybrid powered engines, engine cooling /vehicle thermal management, domestic refrigerator chiller, heat exchanger, nuclear reactor coolant in grinding, machining, in space technology , defense and ships and in boiler flue gas temperature requisition they exhibit enhanced thermal conductivity and convective heat transfer coefficient compared to the base fluid.In analysis such as computational fluid dynamics (CFD), nanofluids can be assumed to be a single phase fluid, classical theory of single fluids can be applied, where physical properties of nanofluid is taken as a function of properties of both constituent and their concentrations. There are many different types of nanofluids that can be made by using different nano particles and base fluid combinations. Some of the most common nanoparticles used are Alumina Oxide (Al_2O_3), Copper Oxide (CuO), Zinc Oxide (ZrO_2), and SilicaOxide (SiO_2). The most common base fluids used for nanofluids are de-ionized water, oil and ethylene glycol.

All nanofluids follow a basic preparation technique. Once the desired weight or volume fraction has been determined, the nanoparticles are added into the base fluid and mixed. Mixing is usually done by ultrasonication to avoid settling of the particles. The amount of time spent mixing the nanofluids depends on the many factors such as the ratio of base to nanoparticles, how long the experiment will last, and the weight or volume fraction used. It's been shown that nanofluids in general have better heat transfer properties than the base fluid alone, specifically better thermal conductivity and heat transfer coefficient. These heat transfer properties theoretically should make nanofluids ideal for phase change heat transfer processes. These enhancements have been researched using experiments such as the transient hot wire method, pool boiling, spray cooling and impinging jet.

The transient hot wire method is a transient dynamic technique where the temperature rise of a sample is measured at a defined distance from a heat source. The hot wire is assumed to have a uniform heat output along its length and the thermal conductivity of the sample can be calculated from the temperature change of the sample over a known time interval. Pool boiling is the process in which vapor is created at the liquid-surface interface by a surface heated above the saturation temperature of the bulk fluid. The motion of the vapor and the surrounding fluid near the heated surface is due to buoyancy forces. As vapor escapes the surface, liquid comes in to fill the void and this process removes heat from the heated surface. Another method that utilizes the impingement of a working fluid onto a heated surface is spray cooling. During spray cooling the pressure difference between the nozzle and the environment is sufficient to create droplets of the working fluid and those droplets impinge the surface to remove heat.

Impinging jet research is another way to study the effects that nanoparticles have on the heat transfer coefficients of the base fluids. A nozzle is used to spray a jet of fluid onto a heated surface to enhance the heat transfer coefficients for convective heating, cooling or drying.

Jet impingement technique is one of the passive methods of convective heat transfer. The jet impingement cooling can be classified into two types. The first type is the submerged jet and the second type is the free surface jet.

In submerged jets, the jet from the nozzle gets injected into the same fluid at the same state. In free surface jet, the jet from the nozzle passes through gaseous atmosphere before impinging the target zone. This investigation work deals with free surface jet impingement cooling. Here arrays of numerous jets are made to impinge a heat sink which has large number of plate fins. The convective heat transfer in this type of cooling is based on the jet impingement velocity, adjacent jet distance to jet diameter (S/d_n), number of jets (n), distance between two rows of jet, jet diameter(d_n), target distance to jet diameter (H/d_n), Reynolds number (R_e),Prandle number (P_r) and physical geometry of heat sink. It is to be noted that the heat extracted by the working fluid should be uniform because temperature non uniformities in the heat sink make it to suffer mechanical stress which leads to permanent or temporary damages. The damage is bending of heat transferring surface makes air to fill which reduces the heat transferring capability very much. Thus the performance of the electrical system is affected due to less heat dissipation. So the temperature uniformity is a vital factor which should be considered while designing a jet impingement cooling with heat sink.

II. LITERATURE REVIEW

The term nanofluid was proposed by Choi in 1995 of Argonne National Laboratory, U.S.A [1]. He stated that "Fluid with nanoparticles suspended in them are called nanofluids". Later many researches are conducted on the properties of nanofluid and to enhance the heat transfer rate using these fluids and it is still going on. Xiang-Qi Wang et al [2] have prepared a paper which gives on overview of the recent developments in the study on heat transfer using nanofluids. This paper obviously shows the various investigations on thermal conductivity, viscosity, convective heat transfer and boiling heat transfer. This also provides clear information about the theoretical investigation of mechanisms of nanofluids, thermal conductivity and numerical investigations. Womac et al [3] have correlated equations for confined submerged and free surface jet impingement cooling with jet diameter 0.513mm, arrays of 2×2 and 3×3 with various jet to jet spacing. He found that by decreasing the distance of jet to jet spacing the heat transfer rate can be enhanced for free surface jets he also explains that the heat transfer can be enhanced by increasing the velocity of jet by raising the volumetric flow rate keeping the d_n and N as constant. He pointed out that the heat transfer doesn't depend on jet to target spacing ranges on $5 \leq H/d_n \leq 10$. In case of confined submerged liquid jet arrays. There is no serious effect in heat transfer coefficient for jet to target spacing lies on $2 \leq H/d_n \leq 4$. D.Y Lee and K.Vafoi [4] performed comparative studies of jet impingement and micro channel cooling. Obviously, the result showed jet impingement cooling is for longer target and micro channel for smaller target. Fabbri and Dhir [5] made an investigation on free surface micro jet arrays using water and FC40 as working fluid. They selected the jet diameter ranges $65\mu_m \leq d_n \leq 250\mu_m$and the Reynolds number ranges $73<R_e d_n<3813$. The investigation shows heat transfer coefficient increases with increase in R_e and P_r as well as in S/d_n. The result was that the heat transfer can be enhanced using micro jet arrays than using a convectional jet arrays. It was found that a maximum heat transfer can be obtained by using an optimum of $75\mu_m$ jet diameter with 5mm as jet interspacing distance.

B.Sagot et al [6] investigated experimentally the jet impingement heat transfer using air on a flat plate maintained at a constant wall temperature. The result reveals that the average Nusselt number (N_u) at constant wall temperature condition is a function of jets Reynolds number (R_{ej}), geometrical parameters (R/D, H/D) and

dimensionless viscosity ratio (μ_j/μ_ω). Brain P.Whelon et al [7] conducted on experiment for determining the nozzle geometry effects in liquid array impingement heat transfer. He used a square array of 45 jets of fixed 1mm diameter, fixed jet interspacing of 5mm and six different nozzle geometries were investigated. Here the jet is made to impinge a circular copper surface with a nominal heat flux of 25.66W/cm^2 and its diameter is 31.5mm. The result shows that by chamfering and contouring the nozzle inlet and outlet increases the heat transfer coefficient, at the same time reduces the pressure drop across the nozzle. AkhileshP. Rallabandi et al [8] have conducted an experiment on heat transfer and pressure loss using jet impingement and channel flow method. He used copper plate as test section with inline and staggered rib condition. The result shows the use of inline rib causes 50% to 90% increase in heat transfer in jet impingement as well as in channel flow method.
Terri B.Hoberg et al [9] have proved heat transfer co-efficient of about 900W/m^{2}k can be achieved using a compact staggered array pattern over an area of 8.5cm^2 with an inter jet spacing of 2.34 jet diameters. Jerome Barrae et al [10] made on experimental study of a new hybrid jet impingement / micro channel cooling scheme for improving the temperature uniformity of the heat sink. In this micro channels of depth 0.5mm and width varies from 3.5mm to 0.5mm is used. The result showed that this geometry enhances the heat transfer coefficient as well as uniform distribution of temperature over the surface of heat sink. Paisarn Naphonand Somachai Wongwises [11] have done on experimental investigation on jet impingement heat transfer for central processing unit of PC. They used deionized water as their working fluid and fabricated a rectangular fin heat sink in copper. The experiment was tested with channels of three different widths and the result showed reduction in processor temperature than other cooling methods. Mangosh Chaudhari et al [12] made on experiment on multiple orifice synthetic jet for improving the impingement heat transfer. He conducted experiments for different configuration with a center orifice surrounded by multiple satellite orifices. The result showed that upto 30% more heat transfer coefficient is obtained in multiple orifice synthetic jet compared to single orifice jet. S.Suresh, et al [13] have presented the effect of $AL_2 0_3$-Cu /water hybrid nanofluid in heat transfer which is carried out in a straight horizontal copper tube of 1000 mm long, 10mm ID and 12mm OD test section with constant heat flux. The result shows that the heat transfer performance is amplified with an average increase in Nusselt number of 10.94% also the convective heat transfer coefficient increases with increasing Reynolds number, but there is an average increase in friction factor of about 16.97% compared to water.

The current study is to determine the effectiveness of alumina nanofluids for dissipating heat from a plate finned heat sink using a inline array jet impingement cooling scheme. The data collected is compared to de-ionized water at the same volume flow rate and distance from the surface. A mass concentrations of 0.1% alumina nanofluids will be compared with de-ionized water to establish a relationship for heat transfer coefficients, interface temperature and pressure drop with volume flow rate. Other parameters can have an effect on the effectiveness of jet impingement cooling, such as the flow temperature at the inlet of the flow regime. Therefore, for the effective investigation of heat transfer study in alumina-water nanofluids the inlet flow temperature is recorded initially before heating takes place and it should be maintained constant till the end of the experiment.

III. SPECIFICATIONS

3.1 Specification of alumina particles

- Size of the particle - 50nm
- Shape of the particle - near spherical
- Density - 3800 kg/m^3
- Thermal conductivity - 40 W/mK
- Specific heat - 773 J/kgK

3.2 Specification of Al_2O_3/water nanofluid

- Density - 1002.8 kg/m^3
- Thermal conductivity - 0.6148 W/mK
- Specific heat - 4174.06 J/kgK

IV. EXPERIMENTAL SETUP AND PROCEDURE

4. 1. Preparation of nanofluid

The preparation of nanofluid includes the production of nano sized particles and dispersing it into the base fluid. The two techniques used to produce nanofluids are single-step method and two-step method. For the preparation of aluminium oxide particles two-step method is more suitable. In the current study aluminium oxide of 0.1% mass concentration is used and the reason for choosing alumina is that it's widely known thermal properties and easy dispersion. The aluminium oxide nanoparticles are purchased from a commercial trader. The

properties of the nanofluid are average particle size = 50nm, density = 3800kg/m^3, thermal conductivity = 40 W/mK, specific heat 773 J/kgK. The required volume fraction of 0.1% was prepared by dispersing the specified quantity in de-ionized water using an ultrasonic bath and sonication was done for 6 hours. This ultrasonic vibrator generates ultrasonic pulses in the power 180 W at 40 KHz. The stability of dispersion is determined by measuring its pH value and it is found around 5.5, which is far from iso-electrical point. Thus the Al_2O_3 nanoparticle in water is more stable. The visual inspection after 10 days showed Al_2O_3 nanoparticles maintained good dispersion with water.
The density of the nanofluid was determined from Pak and Cho equation[14].

$$\rho_{nf} = \rho[1+k_p Ø] \quad (1)$$

The specific heat of the nanofluid was determined from Xuan and Roetzel's equation [15].

$$C_{pnf} = Cp\left[\frac{1+kcØ}{1+kpØ}\right] \quad (2)$$

The thermal conductivity of the nanofluid was determined from Maxwell equation [16].

$$k_{nf} = k[1+k_k Ø] \quad (3)$$

The thermo physical properties of the nanofluid are density = 1002.8 kg/m^3, specific heat = 4801.68 J/kgK and thermal conductivity = 0.6148 W/mK.

4.2. Fabrication of test block

Fabrication of test block starts with shearing the sheet metal and bending it into the required cubical shape. Here a sheet metal of aluminium with a thickness of 1mm is used to fabricate the cover of the test block. The dimension of the heat sink is 60mm×60mm×42mm. It is also made up of aluminium. The heat sink has plate fins with a central flow channel.

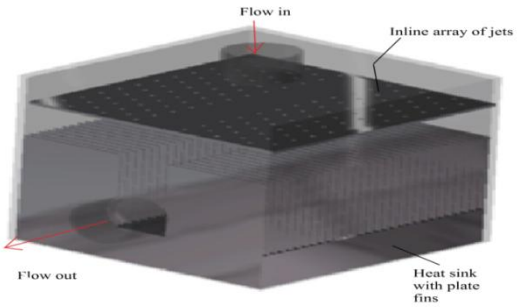

Figure.4.1. Test block

The dimension of the plate fin is 29mm x 12mm x0.56mm. Its thickness is assumed to be negligible. The total number of fins used is 56 and they are arranged in the form of 28 rows and 2 columns with a central flow channel. The fin to fin spacing is about 1.64mm and the channel width is 12mm. The central flow channel helps for the quick excitation of working fluid in order to prevent pool formation. A perforated plate with inline array pattern of jets are kept above the heat sink at a distance of 13mm, the inlet is provided at the top of the test block and two outlets are provided of either side of the central flow channel. The diameter of the inlet and outlet pipe is 13.5mm. The test block is sealed with epoxy compound to prevent the leakage losses.

4.3. Fabrication of heating block

The heating block is a solid aluminiun block. The dimensions of the heating block are 60mm x 60mm x 80mm. A central hole of 5mm diameter is drilled at the center of the heating block. Inside this a heating coil of 150 W is inserted as a tight press fit without any clearance. Another two holes of 2mm diameter on either side of the central hole is drilled to insert the thermocouple. Teflon is coated around the thermocouple to prevent the heat losses around the sides. The heating block is well insulated using polyurethane foam and styrofoam. As these insulating materials have very low thermal conductivity and the heat losses through the side walls can be neglected. Thus one dimensional heat transfer can be obtained in the heating block.

4.4. Acquisition system

The acquisition system is to get data for the heat transfer studies from the heat sensing devices. The acquisition system includes a data logger with a computer system and a temperature display devices. The computer system as a display element is connected to a data logger using a USB connector. This data logger gets data about the fluid flow temperature at the inlet and outlet of the regime. The flow temperature is sensed by a RTDs and the leads are connected to the data logger. Based on the difference in temperature its resistance value changes, the corresponding change is converted into electrical signal in the data logger.

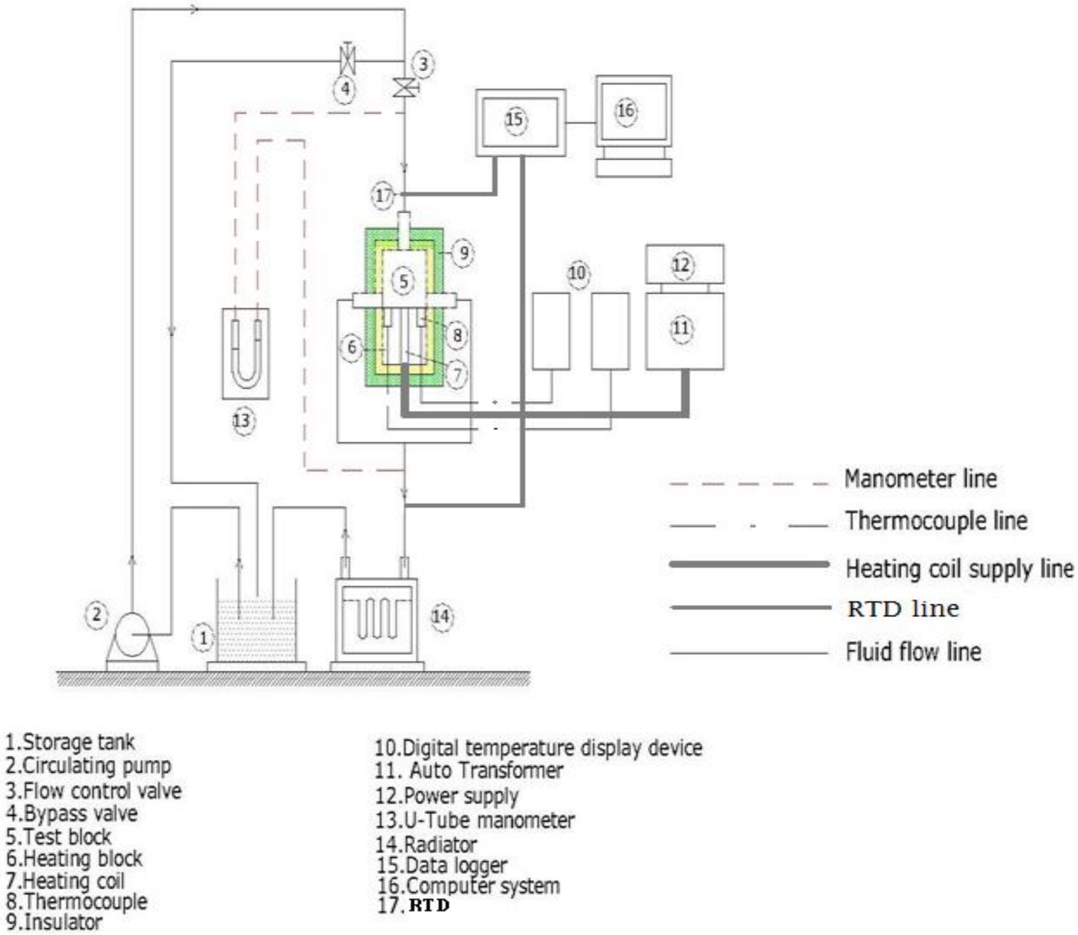

Figure.4.2. Experimental setup

The thermocouple is used to determine the interface temperature. The thermocouple used here is of k-type. The material combination in k-type thermocouple is alumal and chrome. The end leads of the thermocouple are connected to the digital temperature display devices. When there is a change in temperature a small current in the range of milli amps flows through this thermocouple and its value is displayed in the display devices.

4.5.Experimental procedure

Both the test block and heating block are well insulated using polyethylene and neoprene foam mixture and also with styrofoam. A thermal compound is applied at the interface of the heating block and the test block. The inlet and outlet temperatures of the working fluid are measured using RTDs. Data acquisition system such as data logger and digital temperature display devices are used to visualize the temperature obtained. The inlet line is connected to a peristaltic pump and exit to the storage tank through a radiator. The pressure drop between the inlet and exit is measured using U-tube manometer. Power of about 120 watts is supplied to the heating coil through an auto transformer.

After inspecting all the connection switch on the pump. Initially a blow down is done to remove all the dust and impurities then measure the temperature of the fluid. This temperature is taken as inlet temperature of the working fluid forever. Switch on the auto-transformer to energize the heating coil. After stability prevails for a certain volume flow rate, the interface temperature and fluid inlet and outlet flow temperatures are measured and noted from the data acquisition systems. The corresponding pressure drop value is also measured for every change in mass flow rate by noting the difference in mercury level in the manometer limb. The experiment is repeated by changing the mass flow rate for about five times. Calculate the heat transfer coefficient, Nusselt number and pressure drop using the formulae listed in the data reduction part to find out the effectiveness of the jet impingement cooling using alumina nanofluid.

V. THEORETICAL CONSIDERATION

Total heat $Q = Q_{in} - Q_{loss}$ (4)

Where

Q_{in} = Input heat supplied by cartridge heater

Q_{loss} = heat loss from the heated and water blocks assembly

Heat flux,

$$q'' = \frac{Q}{A} \quad (5)$$

A = Exposed heat transfer area of the water block.

The average of the measured interface temperatures (T_i), bulk mean temperature of the inlet and outlet fluids and heat flux are used to calculate the heat transfer coefficient of the water block as given in Eqn (6). The interface temperature is calculated as the average of the two temperatures measured at the interface of two different locations of the heater and water block.

$$h_w = \frac{q''}{(Ti - Tfm)} \quad (6)$$

Where T_{fm} is the bulk mean temperature of the fluid.

Reynolds number is calculated using the following equation.

$$Re = \frac{4m}{\pi \mu Di} \quad (7)$$

The Nusselt number is then calculated as,

$$N_u = \frac{hDi}{k} \quad (8)$$

Pumping power $= \dot{V} \Delta p$ (9)

VI. TABULATION

Table .6.1 Tabulation for de-ionized water

Sl. No	VFR (lpm)	$T_{i\,1}$ ^{0}C	$T_{i\,2}$ ^{0}C	$T_{i\,avg}$ ^{0}C	$T_{f\,in}$ ^{0}C	$T_{f\,out}$ ^{0}C	T_{fm} ^{0}C	h (w/m^{2}k)	Nu	Δp (Pascal)
1	1.315	44.6	42	43.3	32.6	33.1	32.85	57.3854277	1.26379	1200.74
2	1.614	43.9	41.4	42.65	32.6	33.4	33	122.036111	2.687582	1467.576
3	1.944	43.8	41	42.4	32.6	33.6	33.1	190.649411	4.198641	2001.24
4	2.454	43	40.1	41.55	32.7	33.8	33.25	296.627429	6.532578	2801.736
5	2.778	42.7	39.6	41.15	32.7	33.9	33.3	387.316489	8.529808	3335.4

Table.6.2 Tabulation for nanofluid

Sl. No	VFR (lpm)	$T_{i\,1}$ ^{0}C	$T_{i\,2}$ ^{0}C	$T_{i\,avg}$ ^{0}C	$T_{f\,in}$ ^{0}C	$T_{f\,out}$ ^{0}C	T_{fm} ^{0}C	h (w/m^{2}k)	Nu	Δp (Pascal)
1	1.315	43.4	40.5	41.95	32	32.6	32.3	74.3408586	1.632403	1280
2	1.614	42.7	39.9	41.3	32	33	32.5	166.762632	3.661834	1520
3	1.944	42.5	39.6	41.05	32	33.4	32.7	296.357351	6.507522	2115
4	2.454	42.3	39.4	40.85	32.1	33.6	32.85	418.363428	9.186575	2890
5	2.778	42.2	39.3	40.75	32.1	33.7	32.9	514.825866	11.30473	3389.5

VII. RESULTS AND DISCUSSION

Experiments are conducted with deionized water and Al_2O_3/water nanofluid and their results are presented for heat transfer coefficients, interface temperature and pressure drop with volume flow rate. The volume flow rate was given in the range from 1.315 LPM to 2.78 LPM. An array of inline jets were tested with d_n = 2.5 mm, H/d_n = 13 mm and S/d_n = 5 mm.

7.1 Convective heat transfer coefficient

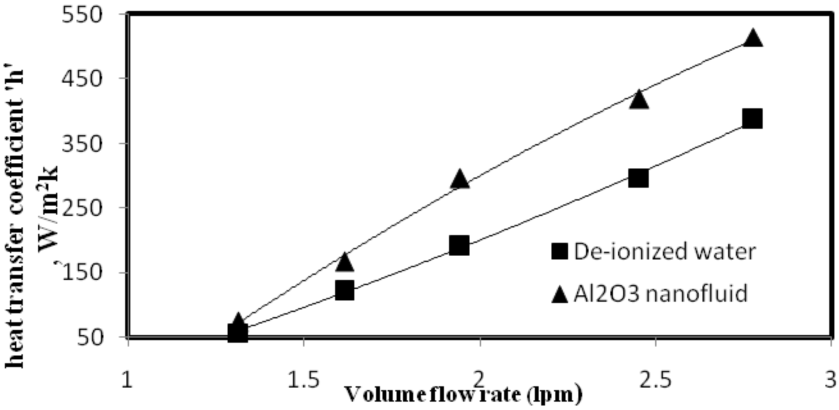

Figure.7.1. Volume flow rate V_s Heat transfer coefficient

Fig. 6.1.Shows the effect of convective heat transfer coefficient on volume flow rate. The heat transfer rate between the solid and the fluid depends upon the convective heat transfer coefficient. It is very important to study the heat transfer coefficient for determining the heat transfer rate. The convective heat transfer coefficient 'h' gradually increases with increasing volume flow rate and it is high for Al_2O_3 /water nanofluid compared to deionized water. Here upto32.92% increase in convective heat transfer coefficient is obtained frorn a nanofluid compared to water at a heat input of 120W. This is because the thermal conductivity of the nanofluid is higher than that of water. Thus the heat transfer rate is high in Al_2O_3/water nanofluid.

7.2 Interface temperature

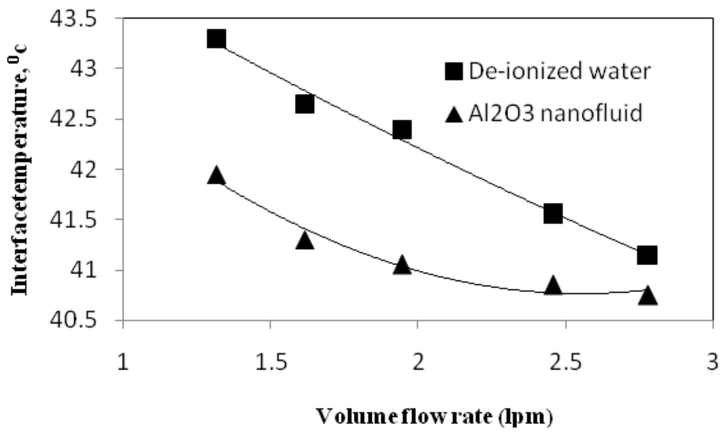

Fig. 7.2. Volume flow rate V_s Interface temperature

Fig.7.2. shows the change in interface temperature with respect to volume flow rate. Interface temperature is the temperature maintained between the heat sink and the heater block at its contacting surface. The interface temperature is reduced as far as possible for better performance of the electronic application. The interface temperature gradually decreases with increase in volume flow rate. The interface temperature maintained between the heated block and the heat sink is less for Al_2O_3/water nanofluid compared to water. It was found that about 0.4^0C decrease in interface temperature in nanofluid to water. This is because the heat transfer rate is high in Al_2O_3 /water nanofluid than water at constant volume flow rate. Consequently the surface temperature T_s gets reduced due to high heat transferring capability of nanofluid and corresponding increase in volume flow rate.

7.3 Pressure drop characteristics

Fig. 7.3.shows change in pressure drop with respect to volume flow rate. Pressure drop is the difference in pressure occurring between the entry and the exit of the test section. The pressure drop is caused due to the

frictional forces in liquid and these frictional forces increases the pump work. So the pressure drop plays a vital role in the fluid flow studies in the test regime. The pressure drop increases with increase in volume flow rate. It is obvious that pressure drop in Al_2O_3/water nanofluid is greater compared to water. This is because the density of the nanofluid is high compared to the density of water. It was found that about 6.6% pressure rise in nanofluid compared to water. The nozzle act as a restriction for the flow and thus a pressure difference is created at the inlet and exit of the nozzle in the test section. At high volume flow rate, there is no significant difference in pressure for deionized water and dilute Al_2O_3/water nanofluid. The figure shows a non linear rise in pressure drop with corresponding increase in volume flow rate.

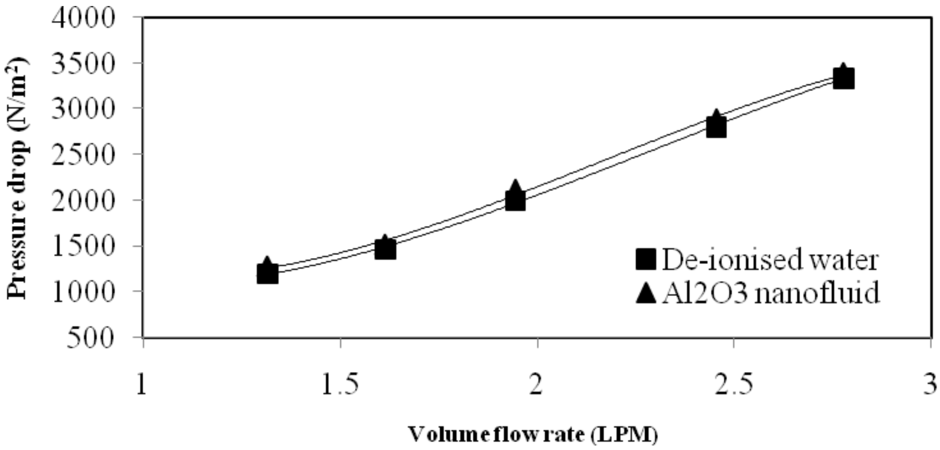

Figure. 7.3. Volume flow rate V_s Pressure drop

VIII. CONCLUSION

In jet impingement cooling the heat transfer coefficient, interface temperature, pressure drop characteristics in a rectangular plate heat sink with inline array jets are experimentally investigated. De-ionized water and Al_2O_3/water nanofluid are used as working fluid and it was found that Al2O3/water nanofluid is the best medium for the heat transfer enhancement compared to water. Jet impingement cooling scheme is compact in nature and hence used for high power density applications. The geometry of the heat sink, rectangular plate fins with a central channel along with uniformly distributed nozzles for jets makes the working fluid to spread uniformly over the entire flow regime to reduce the temperature non uniformity and pool formation.

Result of the study reveals that adding nano particles to the de-ionized water increases the heat transfer rate. About 32.92% increase in heat transfer coefficient is achieved when 0.1% volume concentration of Al_2O_3 nano particles is dispersed in de-ionized water. The interface temperature between the test block and the heater block is reduced by 0.4^0C inAl_2O_3/water nanofluid. Thus the nanofluid gives a better performance when compared to water. But in case of pressure drop it is higher in nanofluid due to its high density. Around 6.6% hike in pressure is found in Al_2O_3/water nanofluid. In low volume concentration there is no significant difference. This reveals that the pumping power required for nanofluid increases with corresponding increase in volume fraction of nano particle.

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LIST OF SYMBOLS

Symbol	Description	Unit
Nomenclature		
A	Exposed heat transfer area	m^2
C_p	Specific heat	J/Kg k
d_n	Jet diameter, m	m
H	Jet to target spacing	m
D	Diameter	m
n	Number of jets	
k	Thermal conductivity	W/m K
h	Heat transfer coefficient	w/m^2 k
Q	Heat transfer	W
Re	Reynolds number	
Nu	Nusselt number	
Pr	Prandtl number	
q”	Actual heat flux	W/m2
m	Mass flow rate	Kg/s
T	Temperature	° C
v	Velocity	m/s
T_i	Average interface temperature	^0c
T_{fm}	Bulk mean temperature	^0c
Δp	Pressure difference	N/m^2
Greek symbols		
Ø	Volume concentration	
μ	Dynamic viscosity	Kg/ms
ρ	Density	kg/m^3