Effect of Nano-Fluid Jet Impingement on Heat Transfer of Surfaces with Different Shapes

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Abstract

This study experimentally investigates heat transfer enhancement of an impinging free liquid jet on a hot surface of copper plate with different shapes, by replacing the base fluid, distilled water with Al_2O_3 nanofluid (10 nm), different shapes of heated surface of copper plate (Flat, Concave, Convex Waved and Corrugated), different impingement Reynolds numbers Re_i ranging from (7,565 to nearly 18,460), five nanoparticles mass concentrations (0%, 0.2%, 0.5%, 1% and 2%) were studied using distilled water and Al_2O_3 nanofluid jet impingement. The results of the current experiments show that increasing the volume flow rate of the cooling fluid improves the heat transfer between the free liquid jet and the hot copper plate. Moreover, using nanofluid jet enhances the heat transfer. Additionally, increasing the surface area enhances the heat transfer between the cooling fluid and the hot plate and decreases the cooling time. Finally, it was noted that surface shape, impingement Reynolds number and nanofluid concentration affect the heat transfer enhancement process significantly. The accumulative effect of using nanofluid jet 2% with corrugated surface with higher flow rate on Nusselt number can reach 185%. On the other hand, using a dispersant for the nanofluid as polyethylene glycol keeps the nanofluid suspension and prevents agglomeration, but reduces its heat transfer enhancement effect.

Keywords: Nanofluid, Heat Transfer, Surface Shapes, Impingement Jet.

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INTRODUCTION

Heat transfer augmentation represents an active and important field of engineering research due to its contribution towards the evolution of highly efficient cooling technologies to meet continuous increase in demand in order to develop an efficacious designs of thermal equipment [1]. Heat transfer enhancement techniques are very important to save energy. In the past decades, heat transfer enhancement technology has been developed and widely applied [2]. As cooling is one of the essential technical challenges that faces many diverse industries, including transportation, microelectronics and manufacturing [3], this raises the need for new and advanced coolants with improved performance. Heat transfer can be enhanced by employing various techniques such as increasing either the heat transfer surface or the heat transfer coefficient between the fluid and the surface that allow high heat transfer rates in a small volume. Rough surfaces or extended surfaces are used for the purpose of

increasing the effective surface area that can achieve a heat transfer enhancement of up to 65% [4].

Further theoretical and experimental research investigations were needed to understand the heat transfer characteristics of nanofluids. It has been found that significant enhancement of heat transfer performance was observed in comparison with pure water. This enhancement was intensified with the increase in the Reynolds number and the nanoparticle concentration [5]. nanofluid shows great promise in enhancing heat transfer characteristics in applications such as electronics cooling, nuclear reactors, and engine cooling applications, although nanofluids cause erosion and clogging of the heat transfer channels [6,7]. Several investigations have revealed that the thermal conductivity of the fluid containing nanoparticles could be increased by more than 20% for the case of very low nanoparticles concentrations. [4]. Another experiment conveyed that the use of nanofluids in a varied range of applications appears promising, but the development of this field may face numerous challenges [8, 9].

Impinging jets have received considerable attention during the last decade. Impinging liquid jet is an established technique to provide high local heat transfer coefficient between the impinging liquid and a surface. The use of jet impingement as a high performance technique for local heating or cooling a surface has become a well-established method because of its high heat transfer rates and its simplicity in application to a variety of industries [10]. The Combining the liquid jet impingement and the nanofluid technologies is thought to capture the advantages of both and consequently enhances the heat transfer significantly [11].

There are various parameters that influence the heat transfer in the jet impingement process including the heat flux, flow rate, inlet pressure, nozzle size, working medium properties and heating surface shape [12]. Using nanofluids (Specially Al_2O_3 –water nanofluids) as coolant was investigated experimentally, numerically, and also mathematically and it was found that the amount of heat transfer increase with the use of nanofluids compared with water as base fluid especially with the increase of Reynolds number and nanoparticle volume fraction [13]. Another investigation revealed that water-based nanofluids of 29 nm copper oxide (CuO) and 36 nm Al_2O_3 nanoparticles lead to a range of effective thermal conductivity improvements from 30% to 52% for nanofluid CuO/Distilled water, and 8% to 30% for nanofluid Al_2O_3 /Distilled water, in case of volume fractions of 6% and 10%, respectively [14]. Nguyen and his colleagues had experimentally studied the heat transfer performance of water– Al_2O_3 nanofluid and reported that using 5% particle volume fraction of the nanofluid provides a heat transfer enhancement of up to 72% compared with water [15].

Increasing the heat transfer efficiency of electronic devices is desirable. There are numerous methods to increase the heat transfer efficiency. Some of these methods include using extended surfaces, application of vibration to the heat transfer surface and the usage of micro-channels. Heat transfer efficiency can be improved by changing the thermal conductivity of the cooling fluid. Commonly used heat transfer base fluids such as water and ethylene glycol have relatively low thermal conductivities, when compared to the thermal conductivity of different solids. High thermal conductivity of these solids can be used to enhance the thermal conductivity of the base fluid by adding small solid particles to base fluid [16, 17].

Nguyen et al., performed an experiment investigation on the heat transfer characteristic of Al_2O_3 –water nanofluids with different volume fractions in a confined and submerged impinging jet on a flat, horizontal and circular heated surface. They found that the highest surface heat transfer coefficients were obtained with an intermediate nozzle-to-surface distance of 5 mm and a 2.8% particle volume fraction nanofluid [18]. Tie et al., experimentally investigated the heat transfer characteristics of CuO–water nanofluids of volume fraction varying from 0.17% to 0.64% flowing through the multiple jet arrays impingement cooling system. The addition of nanoparticles can remarkably improve the heat

transfer behaviors of the base fluid and the heat transfer coefficient increases with the increasing of volume fraction of nanoparticles. It is clear that compared with the case of using nanofluids without any addition of dispersant, the nanofluid with dispersant led to a great deterioration on impingement heat transfer coefficient and even the heat transfer coefficients were smaller than that of the base liquid [19].

Jet impingement cooling is a tool of heat transfer via means of collision of fluid molecules on to a surface. The impinging jet is defined as a high-velocity jet of cooling fluid enforced through a hole or slot which impinges on the surface for cooling, which leads to high heat transfer rates between the wall and the cooling fluid. Heat transfer occurs due to the collision of high velocity fluid molecules on the surface [20]. Zeitoun and Alicarried out an experimental investigation on the heat transfer of alumina-water nanofluid jet on a horizontal circular round surface. The results indicated that using the nanofluid enhanced the Nu up to 100% for some higher concentrations [21]. Jaber, et al., performed an experimental study to investigate the convective heat transfer coefficient of Al_2O_3 /water nanofluid using impingement of a fluid jet on a flat circular disk. The maximum enhancement of the average nanofluid heat transfer coefficient in comparison with water is about 50% for $Re = 4200$ [22]. Nayak et al., experimentally assessed Al_2O_3 and water- TiO_2 jet on hot steel surface. Experimental data were generated for nanofluids with Al_2O_3 and TiO_2 nanoparticle weight concentrations of 0.01%, 0.03%, 0.05% and 0.07%. They found that the nanofluid prepared with Al_2O_3 nanoparticles provided better heat transfer characteristics as compared to that of TiO_2 and DI water [23]. Jaware and Bhagat experimentally investigated the heat transfer and fluid flow due to the impingement of vertical circular single jet on a horizontal heated surface. The experimental results indicated that the local heat transfer coefficient at the stagnation point was more by 50% as compared to the heat transfer coefficient at the outermost point [24].

Teamah, et al., investigated numerically and experimentally the flow structure and heat transfer of a nanofluid jet normally impinging on a flat plate. Al_2O_3 -water nanofluid was used as the working fluid. The results showed that a good agreement was found between the numerical and experimental results. At a concentration of 10.0% and $Re = 24,000$ the heat transfer coefficient increased by 62% compared with the pure water. The Cu nanofluid increased the heat transfer by 8.9% and 12% compared to aluminium and titanium nanofluid, respectively [25]. Huang and Jang numerically investigated the impingement heat transfer features of nanofluids. They observed that the average Nusselt number increased with increasing the nanoparticle concentrations and Reynolds number [26].

Roy, et al., numerically studied the heat transfer performance of a liquid impingement jet using $\gamma-Al_2O_3$ nanoparticles suspended in water. They indicated a considerable improvement in heat transfer rates (up to 100% in local heat transfer coefficient) even with small particle volume fractions [27]. Gherasim, et al., investigated numerically the heat transfer enhancement and pumping power in an impingement jet using suspended nanoparticles of Al_2O_3 -water. They reported that the mean Nusselt number increased with particle volume fraction and Reynolds number and decreased with an increase in nozzle-to-disc spacing [28].

This paper investigates experimentally heat transfer enhancement of an impinging free liquid jet with different volume flow rate (0.6-1.4 l/min) on a copper plate surface with different shapes (Flat, Corrugated, Waved, Convex, Concave) by replacing the base fluid with different mass concentration of nanofluids Al_2O_3 (0% - 2%).

MATERIALS AND METHODS

Experimental setup

The experimental test rig consists of three parts; fluid cycle, heating system and data acquisition system as shown in figure 1 and photo 1

Figure 1 : Schematic diagram of the test rig

Photo 1: A photographic view for the experiment test rig

Fluid Cycle: which includes a cylindrical tank (1) that is connected to suction line of Diaphragm pump (2) maximum outlet pressure of the pump is 862 KPa (125 psi), discharge line is connected to control valve (3) to recycling to fluid tank, the ball valve (4) control the measuring range of flow to Hall effect water flow meter/sensor (5), The cooling fluid

(distilled water or nanofluids) flows through the jet nozzle (6) The nozzle is 25 mm long and 2 mm in diameter. The vertical distance between that nozzle and the heated surface, the nozzle-to-plate spacing, is carefully set to 50 mm from the surface of the copper plate and verified rigorously before and also after any test using prefabricated precision gauges, glass funnel (12) to collect the impinged liquid in the fluid sump (13).

Heating system: as shown in figure 2, which consists of heated copper plates with different geometries as hot surfaces to be cooled with the cooling fluid. The studied geometries are flat, concave, convex, waved and corrugated. The plates are 50 mm in diameter and 5-6 mm thick (7), mica laminated sheets placed on the heater are used to ensure that heat is distributed equally on the hot plate (8). The plates are heated by a 220 V and 150 W. Mica round plate heater (9) is connected to a DC Power supply to provide electric current, the heating system is thermally insulated by using thermal Teflon (10).

Figure 2 : heating system

Data acquisition system: which consists of six Type-K thermocouples, Four Type-K thermocouples probes (11), are used to measure the temperature of the heated surface at the center and at a distance 25, 16, 8 and 1 mm, from the outer edge that represent T1 stagnation point, T2 near impingement region, T3 boundary region and T4 periphery region respectively as shown in figure 3.

Figure3 : thermocouple locations

The thermocouples are connected to a temperature sensor and have a measurement range - 270°C-1260°C, uncertainty $\pm 2.2^{\circ}\text{C}$ and with error level $\pm 1.1^{\circ}\text{C}$. Also, two Type-K temperature probes are placed before the nozzle and exit port of test rig (fluid sump) to measure the temperature of the fluid before and after the impingement to estimate the temperature difference of the cooling fluid, two 4-port Type K differential temperature measurement (14), Xplorer GLX data logger (15) connected to a computer (16).

Nanofluid characterization and preparation**Characterization of the nanoparticle Al_2O_3**

The characterization of aluminum oxide nanoparticles is carried out in Egypt nanotechnology center using transmission electron microscopy (TEM), X-ray powder diffraction (XRD), scanning electron microscopy (SEM), atomic force microscopy (AFM), and Raman spectroscopy. The most important characterization parameters: agglomeration, concentration, size distribution, size, shape, surface charge, surface area, pore size, pore volume, surface roughness, surface waviness and composition.

Transmission electron microscopy (TEM)

Alumina nanoparticles of 10 nm particle size were used the image used by transmission electron microscopy is shown in figure 4.

Figure 4 : TEM image of the synthesized Al_2O_3 nanoparticles

X-ray powder diffraction (XRD)

The average crystallite size of the alumina samples is calculated, utilizing the XRD instrument, using the Sherrer's formula, as shown in Equation 1.

$$D = \frac{K\lambda}{\beta \cos \theta} \quad (1)$$

Where D is the crystallite size, K is the Sherrer's constant ($K = 0.9$), λ is the wavelength, β is the peak width at half height, and θ is the diffraction angle. Figure 5, shows the X-ray powder diffractogram for the tested sample. Peak positions occur where the crystal lattice diffracts the X-ray beam. The X-ray powder diffractogram shows that the diffraction peaks appear at 33° , 37° , 46° and 67° of alumina nanoparticle. The average sizes of the nano-alumina are estimated from the half-height width of the strong peaks to be 1.826 nm.

Figure 5: X-ray powder diffractogram of the synthesized nanoparticles

Scanning electron microscopy (SEM)

SEM image provides information on the distribution, pattern and size of the nanoparticles. Figure6, shows the morphology and particle size of the synthesized nanoparticles that are characterized by the SEM.

Figure6: SEM image micrographs of the synthesized nanoparticles

Atomic force microscopy (AFM)

The AFM is a three-dimensional topographic technique with a high atomic resolution to measure surface roughness. Figure7, shows the AFM topography of the synthesized nanoparticles. The AFM examination, as presented in table 1, reveals that the mean height of the roughness profile (R_c) is 0.0136 μm . The root-mean-square (RMS) surface roughness is 0.00483 μm which indicates that the present Al_2O_3 samples are relatively smooth on an atomic-scale.

Figure7: AFM topography of the synthesized nanoparticles

- (a) 3D AFM image for Al_2O_3 nanoparticles (45×45 μm) (c) 3D AFM image for Al_2O_3 nanoparticles (50×50 μm)
(b) AFM top view for Al_2O_3 nanoparticles (45×45 μm) (d) AFM top view for Al_2O_3 nanoparticles (50×50 μm)

Table 1: Amplitude parameter and roughness profile

Amplitude parameter and roughness profile		
Maximum peak high of the roughness profile	R_p	0.0183 μm
Maximum valley depth of the roughness profile	R_v	0.0224 μm
Maximum height of roughness profile	R_z	0.0407 μm
Mean height of the roughness profile elements	R_c	0.0136 μm
Total height of roughness profile	R_t	0.0407 μm
Arithmetic mean deviation of the roughness profile	R_a	0.00318 μm
Root-Mean-Square (RMS) deviation of the roughness profile	R_q	0.00483 μm
Skewness of the roughness profile	R_{sk}	-0.49 μm
Kurtosis of the roughness profile	R_{ku}	6.70 μm

Raman spectroscopy

A Raman spectrometer is used to obtain detailed information about the molecular structure of the metal oxide. Raman spectra show the crystalline nature of the nanoparticles. In an amorphous state, the Raman bands are quite broad but often can be derived from one of the crystalline forms of the same material. The results of the Raman spectroscopy are summarized in table2.

Table 2: Results of the Raman spectra of the synthesized nanoparticles

Position	Raman shift cm^{-1}
1	59.74
2	364.09
3	473.49
4	1455.4
5	1518.5
6	1716.62
7	1780.68
8	1817.53
9	2328.84
10	2895.34

Preparation of the nanofluid

The two-step method was used for preparation as commercial nanoparticles were purchased in dry powder and dispersed in distilled water. In this case, initially, the Al_2O_3 nanoparticles were mixed in the distilled water then submitted to a sonication treatment for 30 minute to disperse all the material and to break the agglomerates.

Thermo physical properties of alumina nanoparticles in table 3

Photo 2 : AL_2O_3 suspension in distilled water

Table3: Thermo-physical properties of alumina nanoparticles

Property	Value
Particle size	10 nm
Particle shape	Spherical
Appearance	white powder
pH value	6.6
Density (ρ_{np})	3.97 gm/cm ³
Specific surface area	15-20 m ² /gm
High purity	99%
Thermal conductivity (k)	36 W/m-K
Specific heat coefficient (C_{np})	880 J/kg-K

Experiment procedure and investigated parameters

The experimental investigation passes through 6 steps to assess different parameters.

Step 1: to assess the effect of different volume flow rates, $\dot{V}$ from (0.6 to 1.4 l/min) on the heat transfer enhancement of the flat copper plate using distilled water (mass concentration 0%) at target temperature 175°C.

Step 2: to assess the effect of different initial average plate temperatures, sometimes referred to as the target temperature, (175°C, 150°C, 125°C, 100°C and 75°C) on the heat transfer enhancement of the flat copper plate using water (mass concentration 0%) and 1.4 l/min flow rate.

Step 3: to assess the effect of different copper plate surfaces (flat, waved, corrugated, convex and concave) on the heat transfer enhancement of the copper plate using distilled water at best flow 1.4 l/min with target temperature 175°C.

Step 4: to assess the effect of different copper plate surfaces (flat, waved, corrugated convex and concave) on the heat transfer enhancement of the copper plate using nanofluid AL_2O_3 with mass concentration 2% at 1.4 l/min flow rate and with target temperature 175°C.

Step 5: to assess the effect of different mass concentrations of AL_2O_3 nanofluid (0.2%, 0.5%, 1% & 2%) on the heat transfer enhancement of best copper plate shape.

Step 6: to assess the effect of adding polyethylene glycol 600 (PEG 600) with different mass concentrations of AL_2O_3 nanofluid (0.2%, 0.5%, 1% & 2%) on the heat transfer enhancement of best copper plate shape. PEG 600 is added as surfactant which plays an important role in dispersing the nanoparticles into the base fluid and improves the stability of AL_2O_3 /de-ionized water nanofluids.

Theory and problem formulation

A significant parameter for evaluating the heat transfer coefficients between the nanofluid jet and the target is Nu . The magnitude and distribution of the heat transfer coefficient are found to be dependent on several parameters, including $Pr, Re, H/d, d$ and the physical geometry of the jets and the heated surface. Also, Nu depends on various thermo-physical properties of the nanofluid such as thermal conductivity, heat capacity, viscosity, the volume fraction of the suspended particles, and the dimension and shape of the particles.

Thermo-physical properties of the nanofluid

The mixture properties of nanofluids are usually expressed in terms of the mass concentration, X , as expressed by Equation2.

$$X = \frac{m_{np}}{m_{bf} + m_{np}} \times 100 \quad (2)$$

Where m_{bf} and m_{np} are the masses of the base fluid and the nanoparticles, respectively. The mixture properties of the nanofluids are usually expressed in terms of the volume fraction, Φ , or volume concentration, ϕ , (vol%), as expressed by Equation3.

$$\phi = \frac{V_{np}}{V_{bf} + V_{np}} \times 100 \quad (3)$$

Where V_{bf} and V_{np} are the volumes of the base fluid and the nanoparticles, respectively. The mass fraction X can be converted to ϕ , as expressed by Equation4[21].

$$\frac{1}{\rho_{nf}} = \frac{X}{\rho_{np}} + \frac{(1-X)}{\rho_{bf}} \quad (4)$$

Where ρ_{nf} , ρ_{np} and ρ_{bf} are the densities of the nanofluid, the nanoparticles and the base fluid, respectively. For the distilled water- Al_2O_3 (10 nm particle diameter) nanofluid studied, assuming a uniform particle volume fraction and knowing the constituent properties, the density of the mixture ρ_{nf} , can be evaluated using the equation, Equation5[29].

$$\rho_{nf} = (1-\phi)\rho_{bf} + \phi\rho_{np} \quad (5)$$

The specific heat of the mixture C_{nf} can be evaluated using the equation of, Equation6[29].

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$$C_{nf} = (1-\phi)C_{bf} + \phi C_{np} \quad (6)$$

Where C_{bf} and C_{np} are the specific heats of the base fluid and the nanoparticles, respectively. The experimentally measured dynamic viscosity of the nanofluid μ_{nf} can be validated by using the equations Equation 7[30].

$$\mu_{nf} = \mu_{bf} (123\phi^2 + 7.3\phi + 1) \quad (7)$$

Where μ_{bf} is the dynamic viscosity of the base fluid. The thermal conductivity of the nanofluid k_{nf} is measured experimentally and can be calculated by using the equations of, Equation 8 [31].

$$k_{nf} = k_{bf} (4.97\phi^2 + 2.72\phi + 1) \quad (8)$$

Where k_{bf} is the thermal conductivity of the base fluid. The thermo-physical properties of the nanofluid are summarized

Hydrodynamics of the problem

The jet velocity at the nozzle exit, V_j is obtained using the continuity equation at the nozzle exit. The jet velocity at the impingement point can be obtained by applying the energy equation between the nozzle exit and just before the impingement point V_i as expressed by Equation 9.

$$V_i = \sqrt{V_j^2 - 2gH} \quad (9)$$

Where g and H are the gravitational acceleration and nozzle-to-plate spacing, respectively. The jet impingement diameter D_i just before the impingement is calculated from the continuity equation just before impingement in terms of the nozzle diameter D_j as expressed by Equation 10.

$$D_i = D_j \left(\frac{V_j}{V_i} \right)^{0.5} \quad (10)$$

The impingement Reynolds number based on the jet impingement velocity and the diameter at the impingement point is given by Equation 11.

$$Re_i = \frac{\rho V_i D_i}{\mu} = \frac{V_i D_i}{\nu} \quad (11)$$

Where ρ is the density of the cooling fluid and μ is the dynamic viscosity of the cooling fluid.

Heat exchange of the problem

The average heat transfer coefficient $\bar{h}$ between the cooling fluid jet and the target disc with diameter d is calculated via the average Nusselt number $\overline{Nu}_d$ as given by Equation 12.

$$\overline{Nu}_d = \frac{\bar{h}d}{K} \quad (12)$$

Where k is the thermal conductivity of the cooling fluid. The average Nusselt number grows as the Reynolds number increases. Finally, the total electric input power $\dot{Q}$ is defined by heat balance Equation 13 [9], [15], [18], [19], [22], [25] and [28].

$$\dot{Q} = \bar{h}A(\bar{T}_{s_i} - T_{f_i}) = \dot{m}C(T_{f_o} - T_{f_i}) \quad (13)$$

Where $\dot{m}$ and C are the mass flow rate and the specific heat of the cooling fluid, respectively. $\bar{T}_{s_i}$, T_{f_o} and T_{f_i} are the initial average temperature of the heated plate, the average temperature of the cooling fluid after impingement at outlet and the average temperature of the cooling fluid at the nozzle exit, respectively. The average heat transfer coefficient of the cooling fluid is calculated according to Equation 14.

$$\bar{h} = \frac{q''}{(\bar{T}_{s_i} - T_{f_i})} \quad (14)$$

Where q'' is the heat flux calculated as $\dot{Q}/A$.

Calibration and uncertainty analysis

The thermocouples, temperature probes and data acquisition system are calibrated at freezing and boiling points. At the freezing point, the error in thermocouple readings is in the range from -0.2 to 0.1°C. At the boiling point, the error in the thermocouple readings is in the range from 0.15 to 0.4°C. Also, thermocouples were calibrated using a standard liquid thermometer and measuring water at temperature 50°C with an accuracy of 2°C. This calibration ensured that all thermocouples' readings are accurate.

The uncertainties associated with different equipment used in the impingement jet experiment are enlisted in Table 4. To find the uncertainties in the measurements of temperature, velocity, and time, propagation of error is performed. The uncertainty in jet velocity is calculated as $\pm 1.8\%$, whereas the maximum uncertainty in temperature and time measurements is found to be $\pm 1.3^\circ\text{C}$ and ± 0.07 s, respectively. The accuracy of the remaining items of equipment is taken directly from the relevant technical specifications.

Table 4: The measurement range, accuracy and error

Equipment/sensors	Measurement range	Accuracy ($\pm$)	Error ($\pm$)
Hall effect water flow meter	0.3 -10 l/min	1%	0.003 l/min
Diaphragm pump	0-8.5 bar	1%	0.05 bar
Electrometer	0-20 mils	1%	0.01 mils
Mica round plate heater	300 °C	± 0.5	1.0°C
K-type thermocouple	-270 -1260°C	2.2°C	1.1°C
Data acquisition system	-10 -135°C	0.2°C	0.1°C
Thermocouple time response		0.06 s	0.03 s
Data acquisition system time response		0.01 s	0.02 s

EXPERIMENTAL RESULTS AND DISCUSSION:

The results are divided into two sections. The first section presents the cooling curves; details of cooling time, the second section discusses the results of the enhancement of the heat transfer in terms of Nusselt number and the influence of the investigated parameters on Nusselt number.

Cooling curves: -**The effect of the volume flow rate of the cooling fluid:-**

To select the best volume flow rate of the cooling fluid that enhances the heat transfer between the jet and the heated copper plate, the following figure summarizes the results of the investigated five volume flow rates of distilled water jet from (0.6 to 1.4 l/min).

Figure8:Effect of volume flow rate on plate cooling, $X=0$, & $\bar{T}_{si}=175^{\circ}\text{C}$

Figure 8 shows that increasing the volume flow rate of the cooling fluid enhances the heat transfer between the cooling fluid and the plate. Increasing the volume flow rate decreases the cooling time.

The effect of the plate's surface geometry, using distilled water:-

The effect of the plate's surface geometry on the enhancement of the heat transfer between the jet and the heated plate. The following five plate surface geometries are examined: flat,

concave, convex, waved and, corrugated, the following figure summarizes the results of the different geometries.

Figure9:Effect of plate geometry on plate cooling, $X=0$, $\dot{V}=1.4$ l/min & $\bar{T}_{s_i}=175^\circ\text{C}$

Figure 9 shows that increasing the surface area enhances the heat transfer between the cooling fluid and the plate. Also, increasing the surface area decreases the cooling time.

The effect of the plate's surface geometry, using 2% Al_2O_3 concentration:-

The effect of the plate surface geometry on the enhancement of the heat transfer between the jet and the heated plate, while using the nanofluid with Al_2O_3 mass concentration of 2%.

Figure10:Effect of plate geometry on plate cooling, $X=2.0\%$, $\dot{V}=1.4$ l/min & $\bar{T}_{s_i}=175^\circ\text{C}$

Figure 10 shows that adding the nanoparticles to the distilled water enhances the heat transfer between the cooling fluid and the plate.

The effect of Al_2O_3 concentration:-

The effect of mass concentration of the nanofluid on the enhancement of the heat transfer between the jet and the heated plate. The following four Al_2O_3 concentrations are examined: 0.2%, 0.5%, 1%, and, 2%.

Figure 11: Effect of Al_2O_3 fraction on corrugated plate cooling, $\dot{V} = 1.4 \text{ l/min}$ & $\bar{T}_{s_i} = 175^\circ\text{C}$
Figure 11 shows that increasing the concentration of the nanofluid enhances the heat transfer between the cooling fluid and the plate. Increasing the concentration of the nanofluid decreases the cooling time.

The influence of investigated parameters on Nusselt number

The influence of the volume flow rate of the cooling fluid:-

Figure 12 shows that increasing volume flow rate increases Re_i that enhances $\overline{Nu}_d$. As the increase in Re_i from 7565 to 18460 results in 36.8% enhancement in Nusselt number. This is because higher Re_i increases turbulence. Subsequently, the boundary layer breaks away from the plate surface and mixes with the bulk of the cooling fluid further from the plate surface. The figure also compares the experimental results to the Teamah and Farahat's and Zeitoun model.

Figure 12: Effect of Re_i on $\overline{Nu}_d$, for flat surface, $X = 0$, & $\bar{T}_{s_i} = 175^\circ\text{C}$

This figure shows that the two previous models under-predict the current experimental data by about 25% and 10.7% respectively, especially at $Re_i = 13128$

The influence of the plate's surface geometry, using distilled water

Figure 13 shows that increasing the surface area enhances the heat transfer between the cooling fluid and the plate. The figure shows that increasing the surface area increases $\overline{Nu}_d$. Increasing the surface area for the same flux increases the average heat transfer coefficient

Figure 13: Effect of plate's surface geometry on $\overline{Nu}_d$, $X = 0$, & $\overline{T}_{Si} = 175^\circ\text{C}$

This figure shows that changing the plate surface geometry from flat to corrugated increase $\overline{Nu}_d$ by 58.7%.

The influence of the plate's surface geometry, using 2% concentration nanofluid:-

Figure 14 shows that increasing the surface area enhances the heat transfer between the cooling fluid and the plate that increases $\overline{Nu}_d$. The figure shows that changing the plate surface geometry from flat to corrugated increases $\overline{Nu}_d$ by 57.1%, for the 2% concentration.

Figure 14: Effect of plate's geometry on $\overline{Nu}_d$, $X = 2.0\%$, & $\overline{T}_{Si} = 175^\circ\text{C}$

The kinematic viscosity of the 2% Al_2O_3 concentration nanofluid is higher than that of distilled water by 2.58%. This increase in kinematic viscosity decreases the impingement Reynolds number with a subsequent decrease in $\overline{Nu}_d$.

The influence of Al_2O_3 concentration

Figure 15 shows that, for the same concentration, increasing Re_i increases $\overline{Nu}_j$. Also, increasing the mass concentration of Al_2O_3 increases $\overline{Nu}_d$ by 32.2%

Figure 15: Effect of Al_2O_3 fraction on $\overline{Nu}_d$, for flat surface & $\overline{T}_{\text{Si}} = 175^\circ\text{C}$

Figure 16: Effect of Al_2O_3 fraction on $\overline{Nu}_d$, for corrugated surface & $\overline{T}_{\text{Si}} = 175^\circ\text{C}$

Figure 16 shows that increasing the concentration of Al_2O_3 decreases Re_i . This is due to the increase in the kinematic viscosity of the nanofluid, increasing Re_i increases $\overline{Nu}_j$.

The influence of adding dispersant (Polyethylene glycol) to Al_2O_3 nanofluid

Figure 17 shows the effect of adding Poly ethylene glycol (PEG) on the enhancement of heat transfer between the jet and the heated copper plate with different geometries, with 2% Al_2O_3 concentrations, in terms of the impingement Reynolds number and the average Nusselt number.

Figure17: Effect of plate's geometry on $\overline{Nu}_d$, $X = 2.0\%$, with PEG 600 & $\overline{T}_{Si} = 175^\circ\text{C}$
This figure shows that adding dispersant (PEG 600) to Al_2O_3 decreases heat transfer by 2.5% but keeps suspension state and prevent agglomeration.

CONCLUSION

Increasing the volume flow rate of the cooling fluid decreases the cooling time by 26.76% and enhances the Nusselt number by 36.8%.

using nanofluid jet enhances the heat transfer that may reach up to 32.2% at 2% concentration in all geometries.

Increasing the surface area enhances the heat transfer between the cooling fluid and the plate and decreases the cooling time.

The corrugated plate has the shortest cooling time, with a cooling time that is less than the flat surface by 38.5%.

Changing plate surface shape can enhance heat transfer by 58.7% in case of water jet and by 57.1% in case of nanofluid jet with concentration 2%.

The accumulative effect of using nanofluid jet 2% in corrugated surface with higher flow rate on Nusselt number can reach 185% compared with water jet on flat surface with lower flow rate.

The obtained correlation is:-

$$\overline{Nu}_d = Pe_i^{0.36} (1 - X)^{-14.1} A_r^a$$

Where A_r is the relative area defined as $A_r = D_j^2 / 4r^2$, where D_j is the nozzle diameter and r is the impingement surface radius and constant a equal as the following:-

For flat plate surface geometry zero, For concave plate surface geometry -0.0204, For convex plate surface geometry -0.0312, For waved plate surface geometry -0.0474 and For corrugated plate surface geometry -0.0696.

On the other hand, using a dispersant for the nanofluid as polyethylene glycol keeps the nanofluid suspension and prevents agglomeration, but reduces its heat transfer enhancement effect.

RECOMMENDATIONS:

The experiments are performed using different volume flow rates and initial plate temperatures. However, further research is required to address other issues including the following:

Using suitable numerical model to study the effect of nanofluid Jet Impingement on Heat Transfer of Surfaces with different Shapes.

Study the effect of different types of jets on heat transfer enhancement.

Study the effect of different jet diameters on heat transfer enhancement.
Study the effect of different types of nanofluids on heat transfer enhancement.
Study the effect of other shapes of the heating surface on heat transfer enhancement.

Symbols and Abbreviations

A	The relative area - m^2
C_c	Specific heat of the cooling fluid - J/Kg.K
C_m	Specific heat of the mixture - J/Kg.K
C_b	Specific heat of the base fluid - J/Kg.K
C_n	Specific heat of the nanoparticles - J/Kg.K
a	Target disc diameter - m
L	Jet impingement diameter, just before the impingement - m
L_n	Nozzle diameter - m
g	Gravitational acceleration - m/s^2
$\bar{h}$	Average heat transfer coefficient - $W/m^2.^\circ C$
H	Nozzle-to-plate spacing - mm
k_c	Thermal conductivity of the cooling fluid - $W/m.^\circ C$
k_b	Thermal conductivity of the base fluid - $W/m.^\circ C$
k_n	Thermal conductivity of the nanofluid - $W/m.^\circ C$
$\dot{m}_c$	Mass flow rate of the cooling fluid - Kg/s
m_b	Mass of the base fluid - Kg
m_n	Mass of the nanoparticles - Kg
$\bar{Nu}$	Average Nusselt number
Pe	Peclet number $Re \cdot Pr$
Pr	Prandtl number $\mu C_p / K$
q	Heat flux - W/m^2
$\dot{Q}$	Total electric input power - W
Re_i	Impingement Reynolds number
R_p	Maximum peak high of the roughness profile - μm
R_v	Maximum valley depth of the roughness profile - μm
R_z	Maximum height of roughness profile - μm
R_c	Mean height of the roughness profile elements - μm
R_t	Total height of roughness profile - μm
R_a	Arithmetic mean deviation of the roughness profile - μm
R_q	Root-Mean-Square (RMS) deviation of the roughness profile - μm
R_{sk}	Skewness of the roughness profile - μm
R_{ku}	Kurtosis of the roughness profile - μm
T_c	Temperature of the cooling fluid at the nozzle exit - $^\circ C$
T_o	Temperature of the cooling fluid after impingement at outlet - $^\circ C$
$\bar{T}_p$	Initial average temperature of the heated plate - $^\circ C$
V_j	Jet velocity at the nozzle exit - m^2/s
V_i	Impingement velocity - m^2/s
V_b	Volume of the base fluid - litre
V_n	Volume of the nanoparticles - litre
$\dot{V}_c$	Volume flow rates of cooling fluid - l/min
ϕ	Mass concentration - %

Greek Symbols

μ	Dynamic viscosity of the cooling fluid - Kg/m.s
μ	Dynamic viscosity of the base fluid - Kg/m.s
μ	Dynamic viscosity of the nanofluid - Kg/m.s
ν	Kinematic viscosity of the cooling fluid - m^2/s
ν	Kinematic viscosity of the nanofluid - m^2/s

- ρ Density of the cooling fluid - Kg/m³
 ρ Density of the base fluid - Kg/m³
 ρ Density of the nanofluid- Kg/m³
 ρ Density of the nanoparticles - Kg/m³
 φ Volume concentration - %

Abbreviations

AFM	Atomic force microscopy
CHF	Critical heat flux
DI	Deionized
PEG	Polyethylene glycol
SEM	Scanning electron microscopy
TEM	Transmission electron microscopy
vol%	Percentage volume concentration
XRD	X-ray powder diffraction

REFERENCES

1. P. Anand, K. Lam, and K. A. Prakash, "Thermodynamic investigation and multi-objective optimization for jet impingement cooling system with Al₂O₃/water nanofluid," *Energy Convers. Manag.*, vol. 111, pp. 38–56, 2016.
2. R. Gugulothu, K. V. KumarReddy, S. S. Naga, and E. L. Adithya, "A Review on Enhancement of Heat Transfer Techniques; Volume 4," Issue, vol. 2 Part A, pp. 1051–1056, 2017.
3. O. Manca, Y. Jaluria, and D. Poulikakos, "Heat Transfer in Nanofluids," *Adv. Mech. Eng.*, vol. 2, p. 380826, 2010.
4. C. Maradiya, J. Vadher, and R. Agarwal, "The heat transfer enhancement techniques and their Thermal Performance Factor," *Beni-Suef Univ. J. Basic Appl. Sci.*, vol. 7, no. 1, pp. 1–21, Mar. 2018.
5. D. Dey, P. Kumar, and S. Samantaray, "A review of nanofluid preparation, stability, and thermo-physical properties," *Heat Transf. Res.*, vol. 46, no. 8, pp. 1413–1442, Dec. 2017.
6. Stephen . U. S. Choi and J. A. Eastman, "Enhancing thermal conductivity of fluids with nanoparticles," *ASME International Mechanical Engineering Congress & Exposition*, San Francisco, CA, pp. 2-9, November 12-17, 1995.
7. S. Mukherjee and S. Paria, "Preparation and Stability of Nanofluids-A Review," *IOSR J. Mech. Civ. Eng. (IOSR-JMCE)*, vol. 9, no. 2, pp. 63–69, 2013.
8. X.-Q. Wang and A. S. Mujumdar, "A review on nanofluids - part II: experiments and applications," *Brazilian J. Chem. Eng.*, vol. 25, no. 4, pp. 631–648, Dec. 2008.
9. C. T. Nguyen, G. Roy, C. Gauthier, and N. Galanis, "Heat transfer enhancement using Al₂O₃-water nanofluid for an electronic liquid cooling system," *Appl. Therm. Eng.*, vol. 27, no. 8–9, pp. 1501–1506, Jun. 2007.
10. K. Shukla and A. Dewan, "Flow and thermal characteristics of jet impingement: comprehensive review," *Int. J. Heat Technol.*, vol. 35, no. 1, pp. 153–166, 2017.
11. J. H. Lienhard V and J. Haderl, "High Heat Flux Cooling by Liquid Jet-Array Modules," *Chem. Eng. Technol.*, vol. 22, no. 11, p. 967, Nov. 1999.
12. K. C. Jagtap, N. B. Kale, V. V. Kale, K. S. Pawar, and S. A. Deshmukh, "Heat Transfer Enhancement Through Liquid Jet Impingement," *Int. Res. J. Eng. Technol.*, vol. 10, pp. 2395–56, 2017.
13. G. Roy, I. Gherasim, F. Nadeau, G. Poitras, and C. T. Nguyen, "Heat transfer performance and hydrodynamic behavior of turbulent nanofluid radial flows," *Int. J. Therm. Sci.*, vol. 58, pp. 120–129, Aug. 2012.

14. C. H. Li and G. P. Peterson, "Experimental investigation of temperature and volume fraction variations on the effective thermal conductivity of nanoparticle suspensions (nanofluids)," *J. Appl. Phys.*, vol. 99, no. 8, p. 084314, Apr. 2006.
15. C. T. Nguyen, G. Laplante, M. Cury, and G. Simon, "Experimental investigation of impinging jet heat transfer and erosion effect using Al_2O_3 -water nanofluid," in *Proceedings of the 6th IASME/WSEAS International Conference on Fluid Mechanics and Aerodynamics (FMA'08)*, pp. 44–48, August 20–22, 2008.
16. C. H. Li and G. P. Peterson, "Experimental investigation of temperature and volume fraction variations on the effective thermal conductivity of nanoparticle suspensions (nanofluids)," *J. Appl. Phys.*, vol. 99, no. 8, p. 084314, Apr. 2006.
17. P. Keblinski, S. R. Phillpot, S. U. S. Choi, and J. A. Eastman, "Mechanisms of heat flow in suspensions of nano-sized particles (nanofluids)," *Int. J. Heat Mass Transf.*, vol. 45, no. 4, pp. 855–863, 2002.
18. C. T. Nguyen, N. Galanis, G. Polidori, S. Fohanno, C. V Popa, and A. Le Behec, "An experimental study of a confined and submerged impinging jet heat transfer using Al_2O_3 -water nanofluid," *Int. J. Therm. Sci.*, vol. 48, no. 2, pp. 401–411, 2009.
19. P. Tie, Q. Li, and Y. Xuan, "Heat transfer performance of Cu--water nanofluids in the jet arrays impingement cooling system," *Int. J. Therm. Sci.*, vol. 77, pp. 199–205, 2014.
20. F. Afroz and M. A. R. Sharif, "Numerical study of heat transfer from an isothermally heated flat surface due to turbulent twin oblique confined slot-jet impingement," *Int. J. Therm. Sci.*, vol. 74, pp. 1–13, 2013.
21. O. Zeitoun and M. Ali, "Nanofluid impingement jet heat transfer," *Nanoscale Res. Lett.*, vol. 7, no. 1, p. 139, 2012.
22. B. Jaber, T. Yousefi, B. Farahbakhsh, and M. Z. Saghir, "Experimental investigation on heat transfer enhancement due to Al_2O_3 --water nanofluid using impingement of round jet on circular disk," *Int. J. Therm. Sci.*, vol. 74, pp. 199–207, 2013.
23. S. K. Nayak, P. C. Mishra, and S. K. S. Parashar, "Enhancement of heat transfer by water-- Al_2O_3 and water-- TiO_2 nanofluids jet impingement in cooling hot steel surface," *J. Exp. Nanosci.*, vol. 11, no. 16, pp. 1253–1273, 2016.
24. V. B. Jaware and G. P. Bhagat, "Heat transfer enhancement by using nanofluid jet impingement," *Int. J. Eng. Technol. Manag. Appl. Sci.*, vol. 3, no. 5, pp. 50–60, 2015.
25. M. A. Teamah, M. M. K. Dawood, and A. Shehata, "Numerical and experimental investigation of flow structure and behavior of nanofluids flow impingement on horizontal flat plate," *Exp. Therm. Fluid Sci.*, vol. 74, pp. 235–246, 2016.
26. J.-B. Huang and J.-Y. Jang, "Numerical study of a confined axisymmetric jet impingement heat transfer with nanofluids," *Engineering*, vol. 5, no. 01, p. 69, 2013.
27. G. Roy, C. T. Nguyen, and P.-R. Lajoie, "Numerical investigation of laminar flow and heat transfer in a radial flow cooling system with the use of nanofluids," *Superlattices Microstruct.*, vol. 35, no. 3–6, pp. 497–511, 2004.
28. I. Gherasim, G. Roy, C. T. Nguyen, and D. Vo-Ngoc, "Experimental investigation of nanofluids in confined laminar radial flows," *Int. J. Therm. Sci.*, vol. 48, no. 8, pp. 1486–1493, 2009.
29. Pak, B.C., Cho, Y.I., "Hydrodynamic and heat transfer study of dispersed fluids with submicron metallic oxide particles. Experiment". *Heat Transfer* 11 (2), 151–170. 1998.
30. Masuda, H., Ebata, A., Teramae, K., Hishinuma, N., "Alteration of thermal conductivity and viscosity of liquid by dispersing ultrafine particles (dispersion of γ - Al_2O_3 , SiO_2 and TiO_2 ultra-fine particles)." *Netsu Bussei* (in Japanese) 4 (4), 227–233. 1993.
31. Hamilton, R.L., Crosser, O.K., "Thermal conductivity of heterogeneous two-component systems" *I & EC Fundamentals* 1 (3), 187–191. 1962.