A Review on Nanofluid Impingement Jet Heat Transfer

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Abstract

This review paper contains a large number of experimental and numerical investigations about using single or multiple nanofluid jet impingement on a hot surfaces as a heat transfer enhancement technique which employed in many industrial applications. The results of heat transfer, fluid flow characteristics and effects of important parameters were presented and analyzed for all the previous studies. Physics, correlations, properties and applications of nanofluids and nanofluid jet impingement were reported here.

Keywords: heat transfer, fluid flow, jet impingement, nanofluid, single jet, multiple jets

1- Introduction

Heat transfer enhancement is considered to be one of the most important issues which has many industrial applications and this enhancement includes: increasing the heat transfer efficiency, decreasing the cost, size, weight of the cooling or heating systems.

According to Lasance (1997) report about cooling techniques of electronics cooling, Figure 1 obtain the relation between cooling rate in the form of heat transfer coefficient with the corresponding cooling technique. It can illustrated that; liquid cooling methods (heat pipes, cold plates, spray cooling, immersion cooling and jet impingement) enhance heat transfer more than air cooling(piezo fans, synthetic jet cooling, air jets and nanolighting) and conventional cooling(natural and forced convection), in addition, the highest heat transfer coefficients were achieved by boiling and condensation followed by jet impinging. Heat transfer rate achieved by impinging jets is three times higher than the conventional cooling techniques (Jambunathan et al., 1992).

Jet impingement (gas or liquid) is one of the active heat transfer enhancement techniques. Impinging jets have received considerable attention as they are provide large localized heat and mass transfer for a lot of applications such as; electronics cooling, drying of papers and textiles, food processing, processing of metal and plastic, glass tempering, gas turbine cooling, see Figure 2 (Cho et al., 2011)

2- Jet impingement technique

Liquid jets can be classified as shown in Figure 3 (Molana and Banooni ,2013) as free surface, confined, plunging, wall(free surface) and submerged jet impingement. The submerged jet is formed when liquid jet is discharged in to the same liquid medium while the free surface jet if formed when a liquid is discharged in a gas medium.

Jet impingement systems can also be classified as: confined, semi confined and unconfined (free surface) jet. The confined jet impingement, the liquid can get recirculated and be entrained back in to the impinging jet, this causes the formation of recirculation zones in the outlet flow regions (Fitzgerald and Garimella, 1998) In the unconfined jet the heated fluid is not return back in to the jet, in this case the jet interacts with the ambient air and has associated with it higher heat transfer coefficients because the heated fluid is not entrained back into the jet, as is sometimes the case for confined jets (Lupton et al., 2008) . The semi confined jets have a characteristics of both confined and unconfined jets. The selection of liquid jet impingement type is depending on the industrial application.

In addition, the flow of the jet may be dissipated laminar jet (Re<300), fully laminar jet (300<Re<1000), transition jet (1000<Re<3000) and fully turbulent (Re>3000) based on nozzle diameter and velocity at the nozzle exit (Gauntner et al., 1970). Turbulent jets demonstrate superior heat and mass transfer characteristics compared with laminar jets. Figure 4 shows the flow field of an impinging jet and can be analyzed in to three regions:

1. Free jet region: the jet from nozzle spreads and may be fully developed or not as it reaches the stagnation point this is according to the jet to surface distance. The free jet region may be divided to three parts; potential core, developing flow and fully developed part. In the potential core, the velocity is constant and equal to the velocity at the nozzle exit. In the developing flow part, there is a decay of the centerline velocity and this followed by a fully developed flow part where the velocity profiles are similar, the length of this region depends on the shape the jet shape, the nozzle exit conditions and nozzle to plate spacing.

2. Stagnation (impingement) region: the flow is subjected to a strong curvature and very high strain due to the presence of the physical boundary (impingement wall). This region extends to the point where the pressure gradients on the target plate is zero.

3. Wall-jet region: in this region the boundary layer thickness increases while the flow is leaving the stagnation region.

Investigated parameters, heat transfer enhancement using jet impinging technique was investigated either experimentally or numerically trying to find the effects of some parameters on the flow and heat transfer characteristics. These parameters can be concluded as: geometry of the nozzle (circular or slot jets) and shape(long or short, straight or contoured nozzles), heated plate to jet distance (H), slot width for slot jets(W) or nozzle diameter for circular jets(D), the dimensionless number H/W or H/D, jet orientation(vertical, horizontal or angled), target shape(circular, rectangular or square), single jet or multi jets effects, flow type (laminar or turbulent), range of Reynold's number(Re), effect of jet-to-jet spacing dimensionless number (X_n/D), effect of crossflow(minimum, intermediate and maximum crossflow), effect of spent fluid, effect of free and submerged jets, effect of confined and unconfined jets and effects of hole arrangement (inline or staggered).

Jet Reynolds number can be calculated from the following equation:

$$Re_{jet} = \frac{\rho \times v_{jet} \times D_{jet}}{\mu} \quad (2.1)$$

Where:
ρ= base fluid or nanofluid density (kg/m3)
v_{jet}=jet velocity (m/s)
μ=base fluid or nanofluid dynamic viscosity (Pa.s)
D_{jet}= jet diameter (m)

Heat transfer coefficient can be calculated from:

$$h = \frac{q_{net}}{T_{Avg} - T_{jet}} \quad (2.2)$$

Where:
Q_{net}=the net heat flux from the heated plate to the fluid, q_{net}=q_{in}-q_{loss} (q_{in} is the total power from the heater and q_{loss} is the total heat loses from the back surface of heated plate) (W/m^2)
T_{Avg}= average temperature of the impingent surface (˚C)
T_j= jet temperature (˚C)

Average Nusselt number, based on jet diameter, can be calculated from:

$$Nu_{Avg(jet)} = \frac{hD_j}{K} \quad (2.3)$$

Where:
h= average heat transfer coefficient (W/m^2 .̊C).
D_j=jet diameter (m).
K=thermal conductivity of base fluid or nanofluid (W/m ̊C).
Local Nusselt number, based on the jet diameter, can be calculated from equation 13:

$$Nu_x = \frac{h_x D_j}{K} \quad (2.3)$$

Where:
Nu_x= Nusselt number at point x on the plate.
h_x= heat transfer coefficient at point x on the plate $h_x = \frac{q_{net}}{T_x - T_{jet}}$ (W/m^2 .̊C).
K=thermal conductivity of base fluid or nanofluid (W/m ̊C).

There are many reviews focused on the history of impinging jets on a surfaces, the first reviews was made by Martin (1977), Button and Wilcook. (1978), Hrycak 1981, Button et al. (1989), Jambunathan et al. (1992) and Jambunathan et al. (1994). There are many important reviews and investigations were conducted after these reviews which summarized in Molan and Banooni (2013).

Effect of crossflow, H/D and Xn/D investigated by Han and Goldstein (2001), Goldstein and Timmers (1982), Goldstein and Timmers (1982), Obot and Trabold 1987, Metzger and Korstad (1972), Kercher and Tabakoff (1970), Hollworth, and Cole (1987), Hollworth and Berry (1978), Florschuetz and Metzger (1984), Huang (1963), Koopman and Sparrow (1976), Florschuetz and Tseng (1985), Behbahani and Goldstein (1983), Goldstein and Timmers (1982), Florschuetz et al. (1981), Friedman and Mueller (1951) Allande et al. (1961), Gardon and Cobonpue (1962), Chance (1974), Florschuetz et al. (1980), Saas et al. (1980), Florschuetz et al. (1981), Zuckerman and Lior (2006) and Robinson and Schnitzler (2007).

Effect of cuved and concave surfaces studied by Metzger et al. (1969) and Metzger and Korstad (1972), Dyban and Mazur (1970), Gau and Chung (1991), Kornblum and goldstein (1997), Taslim et al. (2000), lee et al. (1996), Bunker and Metzger (1990), Hrycak (1980), Founti (2004).

Effects of nozzle configurations studied by, Royne and Dey (2006), Geers et al. (2004), Geers et al. (2008), Whelan and Robinson (2009). Micro jets arrays investigated by , Leland et al. (2002), Womac et al. (1994), Michna et al. (2011), Fabbri and Dhir (2005).

Other investigations of single jet and multi jets impingement experimental analysis of heat transfer and fluid flow was conducted by Wang et al. (2004), Rallabandi et al. (2010), Sagot et al. (2008), Womac et al. (1994), Paison (2010). Numerical studies were conducted by, Shariatmadar et al. (2016), Brakmann et al. (2015), Li et al. (2015), Isman et al. (2008), Shih and Lumley (1995), Hofman et al. (2007), Kubacki et al. (2011), Wan (2013), Zuckerman and Lior (2005).

3- Jet impingement using Nanofluid

The working fluid using in liquid jet impingement system is usually be water or ethylene glycol, these traditional fluids cannot meet the requirements of high heat flux removal because of its low thermal conductivity. To meet the needs of heat transfer enhancement, an innovative category of heat transfer fluid called nanofluid has been proposed with development of nanomaterials technology. Nanofluid is a mixture of nanoparticles with average size smaller than 100 nm, dispersed in the base fluid (water or ethylene glycol). Nanofluid jet impingement technique is considered to be one of the compound heat transfer enhancement technique as the effect of both jet impingement and nanofluid can be achieved. Nanofluid can enhance heat transfer process due to;

1. Enhance the thermal transport properties of the base fluid.
2. Increase the surface area and heat capacity of the fluids
3. Increase the thermal conductivity.
4. The interaction collision between particles are intensified and reduced particles clogging compared to conventional slurries.

The amount of heat transfer enhancement using nanofluids depends on some parameters such as; the nanoparticles size and type (Al_2O_3, ZnO, Fe_2O_3, CuO, SiO_2, TiO_2…..etc.) and nanofluid concentration (φ). Comparing milli, micro and nano fluids, the nanofluids show more stability and high thermal conductivity with negligible pressure penalty.

3.1 Nanofluids preparation

Preparation of nanofluid is the first stride to the exploratory investigation of the nanofluids. There are two essential strategies to prepare nanofluids: the single-step preparation technique and the two steps preparation technique. The single step direct evaporation method which simultaneously makes and disperses the nanoparticles directly into the base fluid (Figure 5). The two steps process which first makes nanoparticles and then disperses them in to the base fluid (Figure 6), in either case, a well-mixed and uniformly dispersed is a must. For the single step or two step method, the particles must be wetted by the medium and the agglomerations must be prevented. Since, the purity of the nanofluid is important. The most effective method of breaking and dispersing the powder in a fluid is through the application of ultrasonic vibration (also high speed stirring works well) for 12 hours or more. As an alternative for producing nanofluids, direct mixing of nanoparticles in the base fluids may be used. However, using these means of production would be necessary in order to obtain a stable suspensions. A lot of surfactants and dispersants have been used in nanofluids systems which include organic acids, polymers surfactants and salts, these surfactants increase the stability of the nanofluid and decrease the re-agglomerations.

3.2 Nanofluids properties

After preparing a stable nanofluid, it is important to measure and calculate its thermal conductivity, viscosity and density. From previous experiments, physical and transport properties can be measured and validated by comparing the results with the previous studies correlations. there are a lot of measurement devices used for measuring physical and transport properties of the nanofluid such as; Visco lite 700 (hydramotion Ltd, http://www.hydramotion.com) for dynamic viscosity, DMA 35 N density meter (Anton-paar, http://www.anton-paar.com) for density and KD2 pro (Decagon, http://www.decagon.com) to measure the thermal conductivity All of these measurement devices will be calibrated

Nanofluid concentration by volume can be calculated from:

$$\varphi = \frac{Volume\ of\ nanoparticles \times 100}{Volume\ of\ nanopartiles + Volume\ of\ base\ fluid} \quad (3)$$

Where:

φ = nanofluid volume fraction percentage

Volume of nanoparticles= $\frac{W_{np}}{\rho_{np}}$ (m^3)

Volume of base fluid=$\frac{W_{bf}}{\rho_{bf}}$ (m^3)

W_{np}= mass of nanoparticles (kg)

W_{bf}= mass of base fluid (kg)

ρ_{np}= density of nanoparticles (kg/m^3)

ρ_{bf}=density of base fluid (kg/m^3)

The measured dynamic viscosity of the nanofluid can be validated by using the equation 2 of Williams et al. (2008) [and equation 3 of Wang et al. (1999):

$$\mu_{nf} = \mu_{bf} e^{4.98\varphi/(0.2092-\varphi)} \quad (4)$$

$$\mu_{nf} = \mu_{bf}(123\varphi^2 + 7.3\varphi + 1) \quad (5)$$

Where:
μ_{nf}= nanofluid dynamic viscosity (ps.s).
μ_{bf}=base fluid dynamic viscosity (ps.s).
φ=nanofluid concentration %.

The nanofluid density can be calculated using the equation of Williams et al. (2008):

$$\rho_{nf} = \varphi\rho_{np} + (1-\varphi)\rho_{bf} \quad (6)$$

Where:
ρnf=nanofluid density (kg/m3).
ρnp=nanoparticles density (kg/m3).
ρbf=base fluid density (kg/m3).
φ= nanofluid concentration %.
The thermal conductivity of the nanofluid can be calculated by using equation 5 of Williams et al. (2008) and equation 6 of Hamilton and Crosser (1962)

$$K_{nf} = K_{np}(1 + 4.5503\varphi) \quad (7)$$

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

Where:
K_{nf}= nanofluid thermal conductivity (W/m.°C).
K_{np}=nanoparticle thermal conductivity (W/m.°C).
K_{bf}=base fluid thermal conductivity (W/m.°C).

Some experiments and measurements were conducted on the nanofluids to obtain the physical (dynamic viscosity, density) and thermal (thermal conductivity and specific heat) properties for different types of nanofluids and select the best heat transfer carrier. These studies were focused also on the preparation method of the nanofluids to get the maximum nanofluid stability for a long time, some of their results can be concluded as: Masuda et al. (1993), measured the thermal conductivity of Al_2O_3 in water nanofluid for 13 % volume fraction and particle size was 13 and 33 nm, they found a 30 % increase in thermal conductivity for alumina-water compared with water. Park et al. (1998) reported that the best concentration was 4.3 % for 13 nm alumina which gave an enhancement percentage of 32 %. Das et al. (2003) found that the best volume fraction was 4 %, the best particle size 38 nm which delivered 25 % increase in thermal conductivity. Lee et al. (1999) found 10 % increase in thermal conductivity for 24.4 nm alumina in water with 4.3 concentration. Xie et al. (2003)found 20 % enhancement using 5 % nanofluid concentration and 60 nm size. Prasher et al. (2005) estimated that the best concentration was 0.5 % with 10 nm particle size which gave 100 % thermal conductivity enhancement. Krishnamurthy (2006) found 16 % enhancement for 20 nm size and 1 % concentration. Choi (1995), used 36 nm particles to obtain 60 % enhancement with 5 % concentration. Wang and Choi (1999found 35 % enhancement for 10 % concentration and 23 nm size. Lee and Eastman (1999), found 10 % increase in thermal conductivity was obtained using 18.6 nm size and 4.3 % concentration. Chon et al. (2005 concluded that the best concentration and size were 4 % and 28.6 nm respectively which gave 36 % enhancement in thermal conductivity. Experimental and numerical investigations which conducted from 1995 to 2007 of nanofluids properties, preparation methods, measurements and applications were reported in Wang (2007), it can be concluded from the previous studies on heat transfer enhancement of nanofluids , that, the thermal conductivity increase with increasing the nanofluid concentration , Figure 7 show experimental results of thermal conductivity of different types and sizes of nanofluid versus the concentration of nanoparticle in the base fluids.

Many studies about the nanofluid thermal and physical properties were reported by Shanthi et al. (2012) and Mohammed et al. (2011) We can concluded from these studies that; particles size, nature of material, operating temperature, thermal conductivity and PH value of the base fluids, all have influence on the thermal conductivity of the nanofluids. The metallic nanoparticles (Cu, Al, Ag… etc.) in the base fluid having much higher thermal conductivity than oxides at the same concentration. Development, preparation methods, potential features, stability, Heat transfer performance, Thermal and rheological properties, Friction loss and Applications of the Nanofluids till 2015 were reported in Solangi et al. (2015). Latest developments on the viscosity of nanofluids till 2012 were reported in Mahbubul et al. (2012), they also analyzed the effect of nanofluid temperature, particles size and shape, effect of volume fraction on heat and mass transfer of the nanofluids.

3.3 Nanofluid jet impingement experimental analyses

Heat transfer enhancement using nanofluid jet impingement was affected by; nanofluid concentration (volume fraction or mass fraction), nanoparticles type and size, and turbulence models for numerical investigations. These new parameters were added to the previous parameters mentioned above and investigated experimentally by:

Zhou et al. (2015) investigated liquid jet impingement with the use of silver-water nanofluid, the target was a flat plate and finned plate. Nano particles size was 4.8 nm. The results show that, about 6.23, 9.24 and 17.53 % increasing in heat transfer coefficients were obtained at nanofluid weight fraction of .02 %, 0.08 % and 0.12 % respectively compared to the base fluid(water and surfactant) and increasing of 6.61 % compared to water as a base fluid. Average Nusselt number of the heat sink was higher than a flat plate. Nanofluid jet impingement using array of jets investigated by Kumar and Mulugeta (2014), they investigated inline array of jets impinging on a finned plate heat sink, Al2O3-water nonofluid used as impinging fluid, ultrasonic bath was used to disperse the nanoparticles in to the base fluid for 6 hrs., two step method was used to prepare the nanofluid with φ=0.1 % concentration, nanoparticles properties were; size=50 nm, ρ_{np}=3800 kg/m^3, k_{np}= 40 W/m.K and Cp_{np}=773 J/kg.K, nanofluid properties were ρ_{nf}=1002.8 kg/m^3, K_{nf}=0.6148 W/m.K and Cp_{nf}=4174.06 J/kg.K. Target was consisted of an electronic heat sink with aluminum plate fins, two rows of plate fins with 29 mm×24 mm and 0.56 mm thickness, constant heat flux was the condition generated from the target. Jets plate contain 110 holes of D=2.5 mm, H/D=13, jet-to-jet spacing was 2 mm., Q=1.315 l/min to 2.778 l/min. For the CDF study, nanofluid was considered as a single phase fluid, free jet impingement was employed. Results of heat transfer coefficients, interface temperature and pressure drop were calculated. They concluded that, adding nanoparticles to the water increase heat transfer coefficients by 32.92 % at 0.1 % concentration of Al2O3 in water, interface temperature between the test block and heater block reduced by 0.4 C with the use of nanofluid.

Pressure drop increased by 6.6 % due to nanofluid impinging and this in turn lead to increase the pumping power. Tie et al. (2014) investigated jet impingement techniques which employed to enhance heat transfer on a flat plate. They conducted their experiments using four different Cu-water nanofluid concentrations; 0.17 % to 0.64 %. Five nozzles radial staggered arranged jets plates used in this study. Sodium dodecyl benzoic sulfate (SDBS) dispersant with two weight fractions was employed. Results show that, using dispersant in the nanofluid enhance heat transfer greater than using nanofluid without dispersant. Chang and Yang (2013) investigated heat transfer and fluid flow of Al2O3-water nanofluid jet impingement with boiling condition using different additions concentrations. They found that, in case of evaporation and boiling of nanofluids, a nono-sorption layer is formed by nanoparticles on the hot surface and this is decreasing the cooling performance. They tried to enhance heat transfer by applying ultrasonic vibration to the surface and they found an increasing in heat transfer cooling rate. Yousefi et al. (2013) performed an experimental analysis on heat transfer and fluid flow characteristics using single slot jet impinging vertically on a V-shaped plate. Al2O3-water nanofluid is used as a working fluid, φ=0.02 %, 0.05 %, 0.1 % and 0.15 %, Re=1732 to 2719 (laminar flow). They stated that, low nanofluid concentration enhance heat transfer more than high concentrations. Local heat transfer coefficients enhanced by 11.7 % at φ=0.02 % and 21.7 % at φ=0.05 %. Maximum average heat transfer coefficient enhanced by 13.9 % compared to water. Jaberi et al. (2013) studied heat transfer enhancement using Al2O3-water nanofluid single jet impinging on a flat circular plate. Reynolds number varied from 4200 to 8200, the concentration by weight varied from 0.0198 % to 0.0757 %. SDBS is used as a dispersant. Their results show that, using nanofluids jet impingement eanhaced heat transfer coefficients at low concentrations. Maximum enhancement was 50 % at Re=4200 and 0.0597 % weight fraction. The value of enhancement stilled sonstant with increasing Reynolds numbers. Zheng et al. (2013) used nanofluids jet impingement technique to enhance cooling capacity inside the diesel engine. Cu, MgO and Al2O3 in water nanofluids were used as a cooling liquid where impinging on the cylinder head. The results stated that, using nanofluids can eliminate heat from the high heat density regions more than the traditional fluids. The maximum enhancement was 110 %. Zeitoun and Ali (2012), studied heat transfer between single alumina-water nanofluid jet and a horizontal circular plate with different diameters, different jet flowrates and different nozzle diameters, the concentration was 0.66 and 10 %. They found that heat transfer carrier can enhance heat transfer process by using nanofluids, the Nusselt number increased up to 100 % for some higher concentrations, increasing the disk diameter decreases heat transfer coefficients. They compared their results of Nusselt number with the correlation equations of zhao et al. (2002) and Teamah and Farahat (2003) and their results were in good agreement with the imperial formulas. Paisarn and somchai (2011) conducted an experiments on electronics cooling using nanofluid jet impingement on a heat sink. Mass flowrate varied from 0.008 to 0.02 kg/s, Djet= 1, 1.4 and 1.8 mm. results show that, the average surface temperature reduced by 3 % compared with water jet impingement and 6.25 % compared to conventional cooling systems. Another experimental study of Nguyen et al. (2009) , they conducted their experiments using 36 nm alumina-water impinged on a flat, horizontal and circular heated surface, the nozzle diameter was 3 mm, the distance between nozzle and heated pate varied as 2, 5 and 10 mm, Reynold's number range was varied from 3800 to 88000, prandl's number varied from 5 to 10 and the nanofluid concentration was varied from 0 to 6 %. High concentrations were not suitable for heat transfer enhancement purpose under the confined impinging jets. They stated that, using 2.8 % concentration of Al2O3-water nanofluid enhanced heat transfer coefficients at H=5 mm (intermediate height), while, heat transfer coefficients decreased at H=2 mm (very small height) or H=10 mm (large height). The reduction in heat transfer rate is due to the separation flow and the recirculation zones. Gherasim et al. (2009) conducted their experiments on a confined jet using Al2O3- water nanofluid (47 nm size), the jet Reynold's number range was 500 to 946 (laminar flow). They found that, the best concentration was 6 %, Nusselt number increases with particle volume fraction and Reynold's number and decreases with increasing the disk spacing (H). Di Lorenzo et al. (2012) also investigated laminar flow jet impingement using alumina-water nanofluid (30 nm size), Re was less than 500, they used jet slots, the maximum increase of 32 % in terms of heat transfer coefficients was detected at 5 %

volume fraction and H/W=8. Nguyen et al. (2008) studied impingement jet heat transfer and erosion effect using Al_2O_3-water nanofluid (36 nm particle size) with different concentration in a submerged jet, the range of Re=1700 to 20000. They found that the best nanofluid concentration was 5 % which gave an enhancement of heat transfer of 72 %, they concluded that nanofluids have a potential to cause an early wear of mechanical components due to erosion. An experimental study of impingement using CuO-water nanofluid with particle size of 50 nm was conducted by. Liu and Qiu (2007) studied the free surface jet boiling heat transfer of CuO-water nanofluid jet impingement on a flat plate, mass concentration varied from 0.1 % to 2 %, Re=2.5 ×104 to 4×105, Djet=4 mm. results show that critical heat flux (CHF) increased with increasing nanofluid concentrations up to 1 % and still constant for mass concentrations higher than 1 %. CHF increased by 25 % compared with water. A nano-sorption layer is formed on the surface during the jet boiling which reduced the surface roughness. Table 1 summarized parameters and results of the experimental investigations.

3.4 Nanofluid jet impingement numerical analyses

Nanofluid jet impingement was studied numerically in several studies such as:

Yousefi et al. (2016) investigated heat transfer enhancement using Al2O3-water nanofluid jet impingement, the target temperature kept in 340 K and the jet temperature was 293 K, single slot and confined jet was used, they installed obsticles in the channel with different angles (0˚to 60˚), two dimensional simulation and k-ω turbulence model was employed . They concluded that the highest Nusselt number obtained at H/W=1. Increasing nanofluid volume fraction lead to increase the average Nusselt number and the optimum volume fraction found to be 6%. Increasing h/W decreasing average Nusselt number. Maximum Nusselt number occurred at the stagnation point and its is value increased by increasing the obstacle angle, however the average Nusselt number was decreased by increasing the angle from 0 to 15, while, the average Nusselt number started to increase again at angles from 15 to 60. Lam and Prakash (2016) performed a numerical investigation for the Al2O3-water nanofluid jet impinging in a channel with height H, Multiple heat sources of length L were installed and used as a heat flux generator, the simulation was conducted using finite element method . Their results show that, main and secondary recirculation zones were obtained on the upper surface of the chancel for 800 ≥Re≥ 500 in the fourth heat source and, for Re≥800 in the fifth heat source, these bubbles increase the heat accumulation on the wall. Maximum local Nusselt number obtained at the stagnation point and the minimum values of local Nusselt numbers were found in the secondary bubbles regions. Average Nusselt number increased with increasing Re and φ and decreasing H/L. Teamah et al. (2015) investigated numerically and experimentally heat transfer and fluid flow on a flat plate exhibited to Al2O3-water nanofluid jet impingement, Re varied from 3000 to 32000, φ varied from 0 % to 10 % , H/D= 3 (constant), the effect Al2O3- TiO3 and CuO in water on heat transfer and flow characteristics was studied . Their results showed a good agreement between numerical and experimental results. Heat transfer enhanced with increasing the nanofluid concentration, about 62 % increasing in the heat transfer coefficients was obtained at φ=10 % and Re=24000 and with the use of CuO in water. Using CuO-water nanofluid enhanced heat transfer with about 12 % compared to Al2O3 and TiO3 in water nanofluids. An interesting results were obtained by Senkal and Torii (2015), they investigated experimentally and numerically heat transfer and fluid flow of a multi free jet impingement cooling system using Al2O3-water nanofluid, three jet arrangement configurations were considered (9, 17 and 25 radial staggered jets), target was a circular flat plate had the same diameter of the jets plate (31.5 mm), Djet=1.5 mm, Re=1300 to 6000, nanoparticle size= 31 nm, φ=0.5 %, 2% and 4.9%. They summarized that, heat transfer coefficients slightly enhanced at low alumina-water concentrations, but at higher alumina-water concentrations, no heat transfer enhancement was found compared with water, these undesirable results make authors to recommend that using alumina-water nanofluids is not suitable in jet arrays cooling systems. Authors return the reduction in heat transfer to two reasons, first; the recirculation regions (formed between two adjacent jets) where the liquid entrapped and because of the thermal conductivity of the nanofluid is greater than water, the surface temperature increased which reduced the heat transfer coefficients in this zone. The second reason is the separation flow regions which increase the dynamic viscosity and lead to reduce the outflow and make it to accumulate in the recirculation zones. Actually their results is at the same trend of Nguyen et al. (2009) results. The comparison between experimental and numerical results showed a good agreement and small deviation between each other in the low Reynolds number range (low jet velocity), this deviation increased by increasing the jet velocity (high Reynolds number range), the authors return this difference to fluid splashing which is more significant in the experimental analysis than in the numerical simulation (increasing Reynolds number increase the splashing effect). Li et al. (2015) employed a confined single jet of Al2O3-water nanofluid impinging on a dimpled target to enhance heat transfer, Re=10000 to 20000, φ varied from 0 % to 5 %, H/D=6. The separation flow was found near the dimple edge and its effect decreased with increasing the jet velocity. Ahmadi et al. (2016) investigated Al2O3-water nanofluid jet impinging on a concave surface, standard K-ε model turbulence model was employed. Their results show that, optimum H/Djet=5. Increasing Reynolds number and nanofluid concentrations, increase the average Nusselt number. Pumping power increased as a result of the nanofluid viscosity. Dutta et al. (2016) investigated numerically heat transfer of Al2O3-water nanofluid jet impingement from a single slot jet on a flat plate, φ= 3 % to 6 %. Single phase model was employed, Reynolds number varied from laminar to turbulent. Target was kept at constant temperature of 313 K, 2-D flow simulation using RANS models (standard k-ε model, SST k-ωand v2f) was employed. They found that, the average Nusselt number increased by 27 % for laminar flow and 22 % for turbulent flow at φ=6 %. Pumping power increased by increasing the nanofluid volume fraction. Generally, SST k-ωwith transitional flow correction turbulence model was the best, however, for H/Djet > 5, standard k-εwith enhanced wall treatment turbulence model was the best. Selumfendigil and oztop (2014) performed a numerical study of pulsating rectangular jet on a flat plate. Their results show that, in steady case, maximum average Nusselt number increased by 18.8 % at φ=6 % and Re=200. Maximum local Nusselt numbers were found at the stagnation point. In steady case, increasing volume fraction, increased heat transfer enhancement, while in pulsating case the values of Nusselt numbers were found to be less than of steady case. Another important 2-D simulation using two-phase mixture model were conducted by Huang and Jang (2013) , they used Al2O3-water nanofluid with different nozzle to plate distance, different volume fraction, different Reynold's number, local, average and stagnation point Nusselt numbers were calculated and they found that the highest local Nusselt number were obtained at the stagnation point and the lowest value was obtained at the end of heated plate. The average Nusselt increased with increasing the concentration. The best H/D=5. 16 % heat transfer enhancement was obtained with 5 % concentration. They concluded that, SST (shear stress transport) k-ω turbulence model is an appropriate to determine the average Nusselt number, therefore, it is recommended to use this model for applications with separating flow. Manca et al. (2013) investigated heat and mass transfer of Al2O3-water nanofluid slot jet impingement. Laminar flow range was considered and constant surface temperature was applied to the target. The domain length= 031 m, w=0.0062 m, H/W = 4 and 6 and the volume fraction varied from 0 % to 4 %. They found a 22 % maximum enhancement at φ= 4 % and H/W=6. Li et al. (2012) used nanofluid jet impingement to eliminate heat from the electronics devices of high heat flux. Cu-water nanofluid was employed as a cooling fluid and compared with water. Nanoparticles sizes were 25 nm and 100 nm, H/Djet=1, 2, 3 and 4, φ=1.5%, 2%, 2.5% and 3 %. They concluded that, heat transfer enhanced by 52 % at φ= 3 %. Heat transfer enhancement using 25 nm size was more than using 100 nm size. Mitra et al. (2012) investigated experimentally and numerically the boiling phenomena of TiO2-water and multi-walled carbon nanotubes-water (MWCNT-water) nanofluids multi-jet impingement on a hot steel plate. Laminar flow region was considered. The nozzle plate consisted of 91 nozzle of 2 mm length and 5 mm diameter, Xn=15 mm. the steel plate was 295×125× 4 mm thickness, Ts=1200 K. TiO2 size =20- 70 nm and 0.1 % volume fraction, MWCNT size=100-500 nm and 0.01 % volume fraction. The results show that, heat transfer enhancement using CNT is more than using TiO2 in water nanofluid. The critical heat flux (CHF) using nanofluids is lower than that of water but this result doesn't affect the cooling performance. Armaghani et al. (2012) studied 2D flow and heat transfer characteristics of plane jet, they used DNS (direct numerical simulation) model. Al2O3 and CuO in water as a base fluid were employed as a heat transfer carriers, the concentration varied from 0 to 4 %, they found that Al2O3-water nanofluid enhance heat transfer greater than CuO for the same volume fraction and the turbulent intensities in Al2O3 was found to be higher than that in CuO. Rahimi et al. (2012) conducted a 2-D flow simulation of Al2O3-water nanofluid confined jet impingement, forced convection of the nanofluid in a rectangular duct was considered, Re varied from 25 to 275, φ=0% to 0.05 % . Results obtained a 60 % increase in average Nusselt number for Re=275 and φ=0.05 %. Lorenzo et al. (2011) investigated numerically slot jet impingement using Al2O3-water nanofluid as a working fluid, φ=0 % to 5 %, two dimensional flow simulation was considered, W=6.2 mm, H/W=4 to 8, Tw=313(constant), Re=100 to 400. The results show that, 32 % maximum increase in average heat transfer coefficients was obtained at φ=5 % and H/W=8. Pumping power increased with increasing the nanofluid concentrations. Manca et al. (2011) they studied heat transfer enhancement of a two-dimension nanofluid slot jet impingement using Al2O3- water with 38 nm particle size and different concentrations, the range of Reynold's number was 5000 to 20000, distance between jet and hot surface was 24.8 to 124 mm, jet width= 6.2 mm, H/W varied as 4, 6, 8, 10, 15 and 20, the domain length= 310 mm, boundary condition of the heated plate was selected as a constant temperature(343 K) with no slip condition, velocity profile at the inlet jet section selected to be uniform, the single-phase model was selected . The results of the stream function shows that the vortex intensity and size depend on H/W ratios, Reynold's numbers and volume fraction, they also concluded that, increasing nanofluid concentration causes an increase in the bulk temperature. The highest Nusselt number was obtained at the stagnation point and the lowest Nusselt number was obtained at the end of the heated plate, the best H/W was 10, the results obtained 18% increase in Nusselt number at 6 % volume fraction, pumping power increased with increasing Reynold's number and nanofluid concentration. Gherasim et al. (2011) investigated heat transfer in confined jet impingement using Al2O3 in water nanofluid, they found that 6% volume fraction was the best value for 47nm nanofluid. Yang and Lai (2010) used 47nm of 20% Al_2O_3 water nanofluid in laminar flow jet impingement and found 20% increase in Nusselt number for 10% concentration. Freng (2010) used Al_2O_3 water with 30 and 47 nm size and laminar flow jet impingement(Re <800) ,he found that the best concentration was 4% , nanofluids are a smoother mixture flow fields when compared with water as a base fluid . Vaziei et al. (2009) studied heat transfer and fluid flow characteristics of the confined and submerged jet impinging on a flat plate, size= 36 nm of Al2O3-water, φ=2.8 % and 6 %. For laminar flow, the results show that, the stagnation point Nusselt number increased by towice at H/Djet=2. For turbulent flow, Nusselt decreased increasing Reynolds numbers. Confinement surface plays an important role in the heat transfer enhancement. Wang Xiangqi (2007) used different nanoparticles (Al2O3, CuO and CNT) and different volume fraction for laminar flow confined jet impinging, the best concentration for Al2O3-water found to be 10 % which gave 30 % increase in Nusselt number, the best concentration for CuO-water was 1 % to achieve 100% increase in Nusselt number and for CNT- water the best concentration was 10 % achieving 80 % increase in Nusselt number. Palm et al. (2006), investigated Al2O3-water nanofluid with 38 nm in laminar flow jet impingement (Re=500 to 946), the best volume fraction was 7.5 % achieving 70 % Nusselt number. Roy et al. (2004) used Al2O3-water nanofluid with different volume fraction, Reynold's number varied from 200 to 2470 (laminar flow) they found that 10 % concentration is the best value achieving 110 % heat transfer enhancement. Table 2 summarized parameters and results of the numerical investigations.

4- Conclusions and recommendations

jet impingement cooling using air, water and nanofluids was reviewed here trying to study the effects of some important parameters on heat transfer and fluid flow characteristics, such as; geometry of the nozzle (circular or slot jets) and shape(long or short, straight or contoured nozzles), jet diameter, dimensionless numbers H/D, X_n/D, jet orientation(vertical, horizontal or angled), target shape(circular, rectangular or square), flow type (laminar or turbulent), range of Reynold's number(Re), effect of crossflow(minimum, intermediate and maximum crossflow), effect of spent fluid, effect of free and submerged jets, effect of confined and unconfined jets, open area (A_f), area ratio (A_r), nanofluid concentration(volume fraction), nanoparticles type and size, and turbulence models for numerical investigations.

A few number of studies were focused on the square inline arrays of impinging jets with the use of nanofluids and there isn't any previous investigations were conducted using square staggered arrays so; its highly recommended for future work to investigate either experimentally or numerically , heat transfer and fluid flow for inline and staggered arrays of nanofluid jet impingement.

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Table 1 nanofluid jet impingement experimental investigations.

Authors	*Jet type*	*D_{jet}*	*Re*	*H/D_{jet} or H/W*	*Type/ Size (nm)*	*φ %*	*X_n/D_{jet}*	*Single Or Arrays*	*Nu %*	*Notes*
Zhu et al. (2015)	Confined	--	--	---	Silver-water 4.8 nm	02 %, 0.08 % and 0.12 % Wt %	--	Single	6.23, 9.24 and 17.53 %	Flat plate and finned plate
Kumar and Mulugeta (2014)	Free jet	110 holes of D=2.5 mm	---	H/D=13	Al2O3-water50 nm	0.1 %	Xn=2 mm	inline array	32.92 % at φ=0.1 %	Pressure drop increased by 6.6 % due to nanofluid
Tie et al. (2014)	confined	---	---	---	Cu-water 26 nm	0.17 % to 0.64 %	---	single	---	(SDBS) dispersant with two weight fractions was employed
Chang and Yang (2013)	confined	---	----	---	Al2O3-water 27-43 nm.	0, 0.0001, 0.001 and 0.01 vol%	---	single	---	jet impingement with boiling condition ultrasonic vibration to the surface increase heat transfer cooling rate
Yousefi et al. (2013)	confined	---	1732 to 2719 (laminar flow)	---	Al2O3-water	0.02 %, 0.05 %, 0.1 % and 0.15 %	----	single slot	----	V-shaped plate Maximum average heat transfer coefficient enhanced by 13.9 %
Jaberi et al. (2013)	confined	---	4200 to 8200	----	Al2O3-water	0.0198 % to 0.0757 % wt%	----	single	50 % at Re=4200 and 0.0597 % weight fraction	SDBS is used as a dispersant
Zheng et al. (2013)	confined	----	----	----	Cu, MgO and Al2O3	----	----	single	110 %	-----
Zeitoun and Ali (2012)	confined	---	----	---	Al2O3	0.66 and 10 %.	---	single	100 %	horizontal circular plate with different diameters
Paisarn and somchai (2011)	confined	1, 1.4 and 1.8 mm	---	distance nozzle-to-fins tip is 2.00 mm	TiO2	----	----	array	temperatures obtained from the jet nanofluids impingement cooling system are 3.0%, 6.25% lower than water	conducted an experiments on electronics cooling using nanofluid jet impingement on a heat sink
Nguyen et al. (2009)	confined	3 mm	3800 to 88000	H=2, 5 and 10 mm	36 nm alumina-water	0 to 6 %	---	single	---	they concluded that , the highest surface heat transfer coefficients were obtained at φ=2.8 % and H=5 mm above the target
Gherasim et al. (2009)	confined	---	500 to 946 (laminar flow)	---	Al2O3-water nanofluid (47 nm size)	0 to 8 %	---	single	---	he best concentration was 6 %
Di Lorenzo et al. (2012)	confined	---	less than 500	---	alumina-water nanofluid (30 nm size)		---	Single slot	32 %	The best condititons5 % volume fraction and H/W=8
Nguyen et al. (2008)	submerged	---	1700 to 20000	---	Al_2O_3-water nanofluid (36 nm particle size)	---	---	single	72 %	he best nanofluid concentration was 5 %
Liu and Qiu(2007)	free surface jet	4 mm	2500 to 400,000	---	CuO- water nanofluid with particle size of 50 nm	0.1 % to 2 % Wt %	---	single	---	the surface tension of the nanofluid was about 75 % higher than water, the CHF for nanofluid increased about 25 %

Table 2 nanofluid jet impingement Numerical investigations.

Authors	Jet type	D_{jet}	Re	H/D_{jet} or H/W	Type and Size (nm)	φ%	X_n/D_{jet}	Single Or Arrays	Nu%	Notes
Yousefi et al. (2016)	confined	---	---	the highest Nusselt number obtained at H/W=1	Al2O3-water	optimum volume fraction found to be 6%	---	single slot	---	k-ω turbulence model was employed
Lam and Prakash (2016)	confined	---	100-1000	---	Al2O3-water	---	---	single	---	Maximum local Nusselt number obtained at the stagnation point and the minimum values of local Nusselt numbers were found in the secondary bubbles regions.
Teamah et al. (2015)	confined	---	3000 to 32000	H/D= 3 (constant)	Al2O3-TiO3 and CuO in water	0 % to 10 %	---	single	---	Numerical + Experimental Using CuO-water nanofluid enhanced heat transfer with about 12 % compared to Al2O3 and TiO3 in water nanofluids
Senkal and Torii (2015)	---	1.5 mm	1300 to 6000	---	Al2O3-water 31 nm	0.5 %, 2% and 4.9%	---	9, 17 and 25 radial staggered jets	---	Numerical + Experimental heat transfer coefficients slightly enhanced at low alumina-water concentrations
Li et al. (2015)	confined	---	10000 to 20000	H/D=6	Al2O3-water	0 % to 5 %	---	Single	---	dimpled target. The separation flow was found near the dimple edge and its effect decreased with increasing the jet velocity
Ahmadi et al. (2016)	confined	---	---	H/Djet= 2-10	Al2O3-water	---	---	single	---	concave surface. standard K-ε model optimum H/Djet=5
Dutta et al. (2016)	confined	---	Laminar + turbulent	---	Al2O3-water	3 % to 6 %.	----	single slot	27 % for laminar flow and 22 % for turbulent flow at φ=6 %	standard k-ε model, SST k-ωand v2f
Selumfendigil and oztop (2014)	confined	---	100- 400	---	Al2O3-water	Optimum concentration0%, 3 %, 6 %	---	single	18.8 % at φ=6 % and Re=400	pulsating rectangular jet
Huang and Jang (2013)	confined	---	---	best H/D=5	Al2O3-water	best φ=5	---	single	16 %	SST (shear stress transport) k-ω turbulence model is an appropriate to determine the average Nusselt number
Manca et al. (2013)	confined	w=0.0062 m	Laminar flow	H/W = 4 and 6	Al2O3-water	0 % to 4 %	---	Slot single jet	22 %	22 % maximum enhancement at φ= 4 % and H/W=6
Li et al. (2012)	confined	---	---	H/Djet= 1, 2, 3 and 4	Cu-water 25 nm and 100 nm	φ=1.5%, 2%, 2.5% and 3 %	---	single	52 %	heat transfer enhanced by 52 % at φ= 3 %. Heat transfer enhancement using 25 nm size was more than using 100 nm size
Mitra et al. (2012)	confined	5 mm	---	---	TiO2-water size =20- 70 nm and MWCNT-water size=100-500 nm	TiO2 0.1 % MWCNT 0.01 %	---	91 nozzle of 2 mm length	---	Experimental and Numerical. The critical heat flux (CHF) using nanofluids is lower than that of water but this result doesn't affect the cooling performance

Continuation Table 2

Authors	***Jet type***	***D_{jet}***	***Re***	***H/D_{jet} or H/W***	***Type and Size (nm)***	***φ%***	***X_n/D_{jet}***	***Single Or Arrays***	***Nu%***	***Notes***
Armaghani et al. (2012)	confined	---	---	---	Al2O3 and CuO in water	0 to 4 %	---	single	---	they found that Al2O3-water nanofluid enhance heat transfer greater than CuO
Rahimi et al. (2012)	confined	---	25 to 275	---	Al2O3-water	0% to 0.05 %	---	single	%60	Forced convection of the nanofluid in a rectangular duct. 60 % increase in average Nusselt number for Re=275 and φ=0.05 %.
Lorenzo et al. (2011)	confined	W=6. 2 mm	100 to 400	H/W=4 to 8	Al2O3-water	φ=0 % to 5 %		Single slot	32 %	32 % maximum increase in average heat transfer coefficients was obtained at φ=5 % and H/W=8
Manca et al. (2011)	confined	W=6. 2 mm	5000 to 20000	H=24.8 to 124 mm, H/W varied as 4, 6, 8, 10, 15 and 20	Al2O3-water 38 nm	--	--	Single slot	18%	the best H/W was 10, the results obtained 18% increase in Nusselt number at 6 % volume fraction
Gherasim et al. (2011)	confined	---	---	---	Al2O3 in water 47nm	0% to 10 %	--	single	---	6% volume fraction was the best value for 47nm nanofluid
Yang and Lai (2010)	confined	---	laminar flow		Al_2O_3 water 47nm	20%	---	single	20%	---
Freng (2010)	---	---	Re <800	---	Al_2O_3 water with 30 and 47 nm	---	---	---	---	he found that the best concentration was 4%
Vaziei et al. (2009)	confined and submerg ed	---	---	--	Al2O3-water 36 nm	2.8 % and 6 %.	---	---	---	the stagnation point Nusselt number increased by towice at H/Djet=2
Wang Xiangqi (2007)	confined	---	---	---	Al2O3, CuO and CNT	---	---	--	Al2O3-water 30 % Al2O3-water 30 % CuO-water 100% CNT- water 80 %	---
Palm et al. (2006)	confined	---	500 to 946	--	Al2O3-water with 38 nm	----	---	---	---	the best volume fraction was 7.5 % achieving 70 % Nusselt number
Roy et al. (2004)	---	---	200 to 2470	--	Al2O3-water	---	---	---	---	they found that 10 % concentration is the best value achieving 110 % heat transfer enhancement

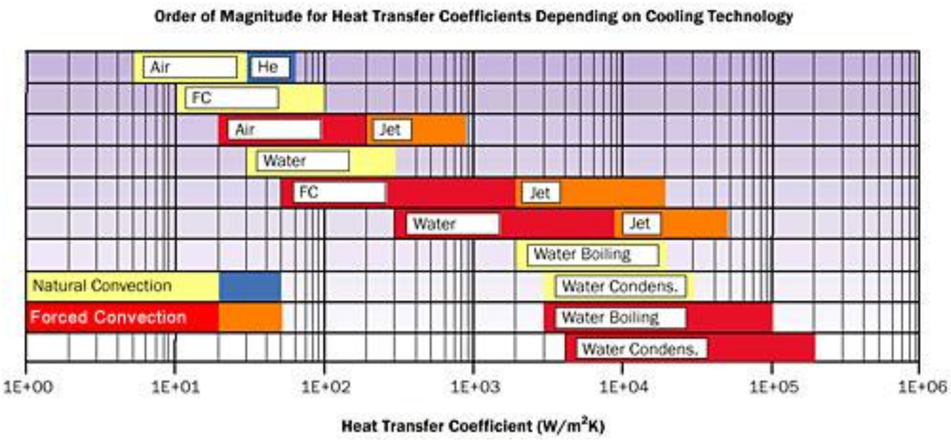

Figure 1 Relation of heat transfer coefficient with the corresponding cooling technique. (Lasance,1997)

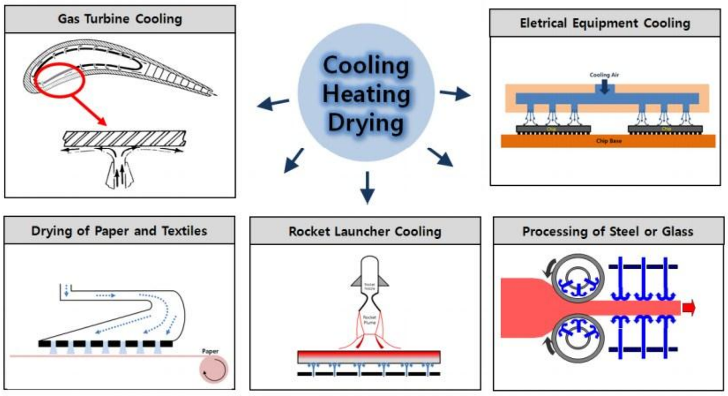

Figure 2: Impingement jet industrial applications. (Cho et al., 2011)

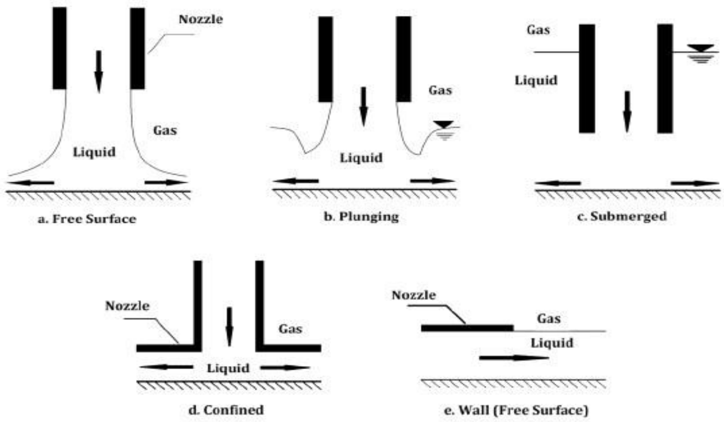

Figure 3 Different types of jet impingement (Molana and Banooni ,2013)

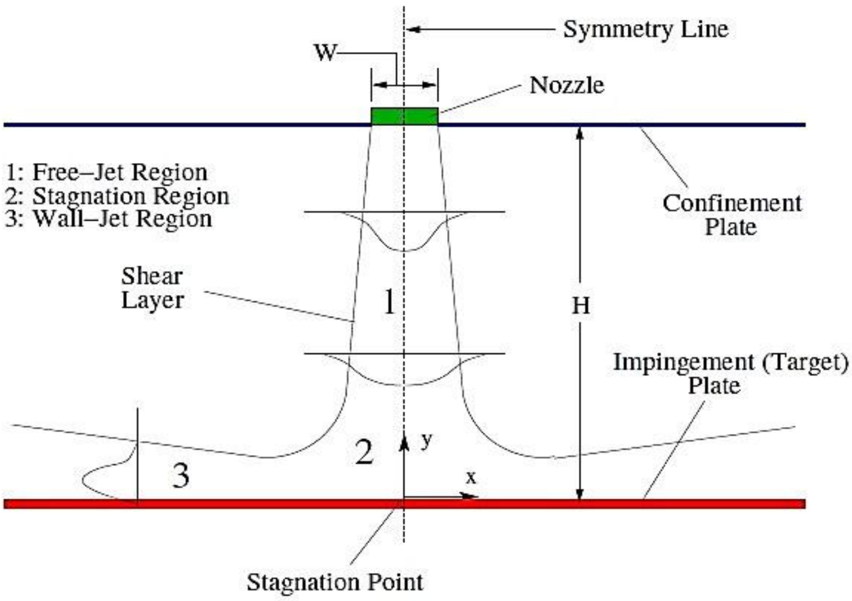

Figure 4 A schematic of jet impingement flow fields (adopted from Kabari, 1977)

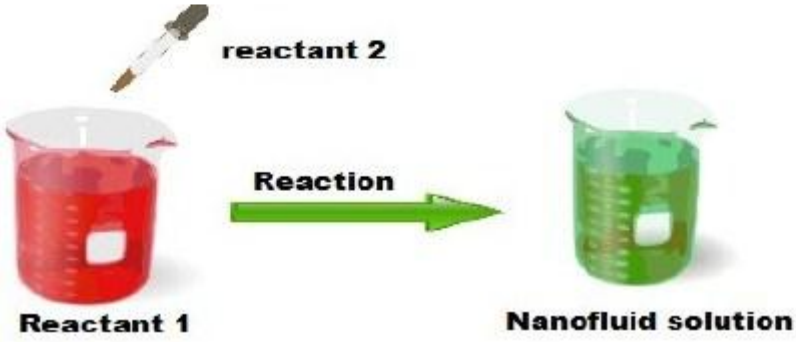

Figure 5 single step nanofluid preparation method, (Mukherjee and Paria, 2013)

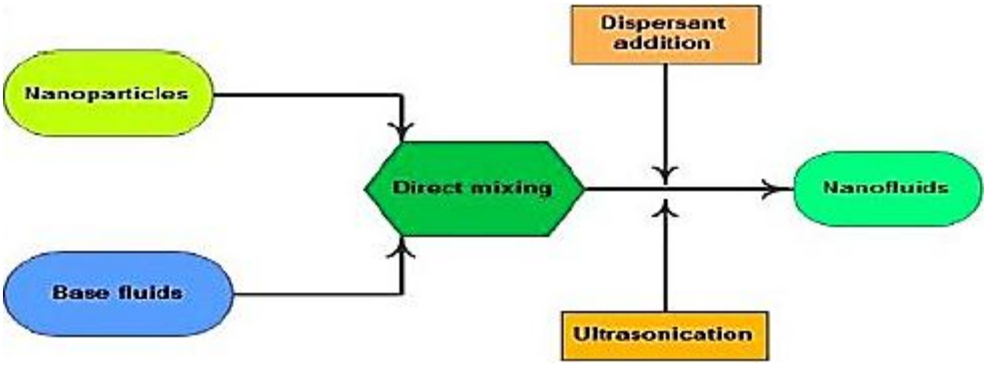

Figure 6 Two steps nanofluid preparation method (Mukherjee and Paria, 2013)

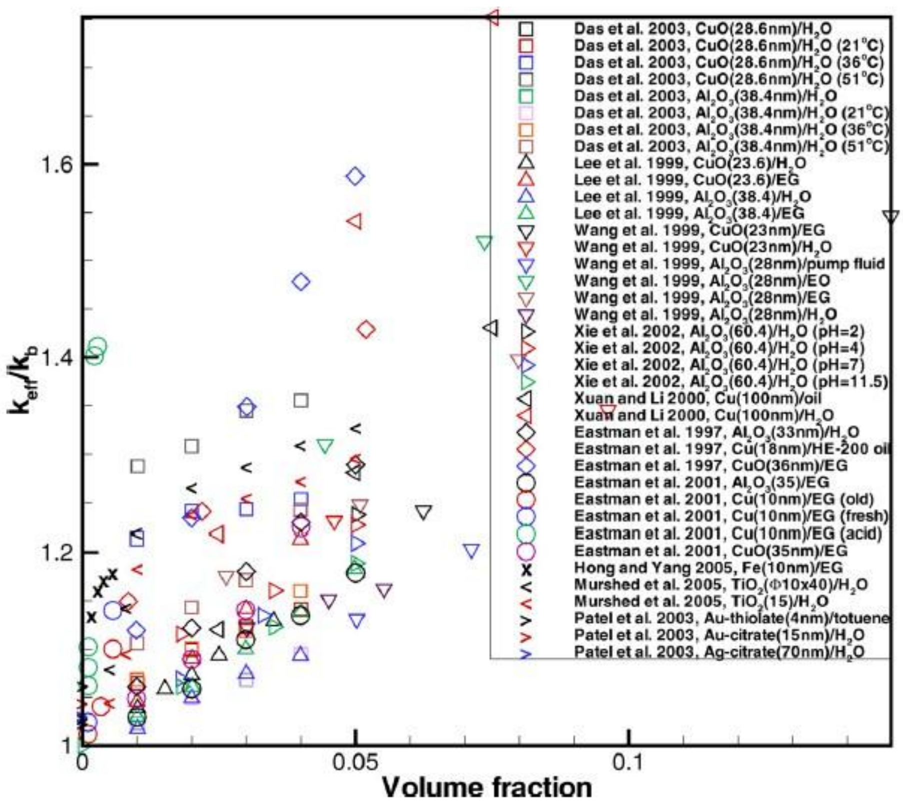

Figure 7 : Relation of thermal conductivity with nanofluid volume fraction (Wang ,2007)