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Research Article

Experimental Study of Quenching Progression on a Heated Flat Dimpled Surface with Water Jet Impingement

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ABSTRACT

Quenching progression on a flat surface and heat transfer enhancement with an impinging jet over smooth and dimpled surface modification is presented in this manuscript. With rapid advancements in today's electronic, electrical, and mechanical systems, the need for the removal of the associated heat generation rates is also increasing. Achieving that by jet impingement provides an economical and fast solution. A smooth flat plate is quenched repeatedly from three different initial temperatures of 300,350 and 400° C. The results in terms of re-wetting parameters viz., Re-wetting temperature, and wetting delay are reported. Parallelly, the effect of the hemispherical dimpled (array) surface with a pitch of 3 mm, and diameter of 2 mm with varying depths of 0.5 mm(d/t=10) and 1mm(d/t=5) are studied. The results are then compared to that of a smooth surface. Water is used as a coolant at a temperature of 17 ±2°C. A large deviation in results is reported when the plate surface was subjected to repeated trials due to a change in the metallurgical properties of the surface. The results of a dimple depth of 0.5mm show a higher heat transfer rate as compared to that of both the smooth surface and the dimple depth of 1 mm. A maximum of 40% and a minimum of 26% enhancement in heat transfer rate is reported for dimple depth 0.5mm compared to 1mm. Further, a 59.76% of heat transfer efficiency was recorded for the experimental setup and this efficiency was found to be increasing with an increase in the water pump pressure.

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1. Introduction

With the latest advancements in today's systems, heat dissipation is a must for further development. Various systems like pool boiling, jet impingement, and spray cooling is being in use today. Among them, jet impingement provided the most economical solution and is used for quenching. A few of the applications of cooling by jet impingement include cooling of electronic equipment [1], glass manufacturing [2], food processing [3], nuclear power plants in case of Loss-of-Cooling-Accident (LOCA) [4], steel industries during strip rolling [5], manufacturing of optical fiber [6], fire suppression mechanism [7], cooling of turbine blades [8], etc. With an increase in the requirement of effective heat dissipation, numerous analytical, experimental, and computational studies are conducted for quenching by jet impingement. Each study with its distinct flow characteristics and different objectives is conducted and the parameters such as rewetting temperature, rewetting velocity, wetting delay, etc. are evaluated to find out the cooling performance. For better heat transfer and faster cooling rates, various techniques such as the use of modified surfaces [9], flow control techniques [10], use of nano-fluids [11], and integration of PCM [12] with

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jet impingement technique is employed and better results are recorded. Among these, surface modifications like the use of extended surfaces, introduction of roughness elements, use of ribs, and use of surface coating have seen a lot of interest getting developed among researchers in modern times.

The quenching phenomenon by liquid jet impingement is explained in analogy with different boiling regimes observed in the flow boiling. In atmospheric conditions, when water is above the saturation temperature, film boiling takes place and the liquid is vaporized instantaneously. However, this behavior does not go as long continuously. Above a certain value of temperature, a vapor layer is formed over the targeted surface. After that, when the vapor film is fully developed, the droplet gets completely separated from the hot surface, and the liquid displays a minimum rate of evaporation for a temperature exceeding its normal boiling point. The corresponding temperature is known as the Leidenfrost Temperature (rewetting temperature). Once the Leidenfrost temperature is reached, the vapor layer breaks down and the establishment of liquid takes place. This phenomenon is known as the Rewetting phenomenon and the corresponding parameters associated with the phenomenon are known as rewetting parameters.

Some of the noteworthy terminology and fundamental concepts associated with these phenomena are:

Re-wetting Temperature (T_{rw}): The temperature corresponding to the rewetting phenomenon i.e., at which the liquid-solid re-contact takes place is known as Re-wetting temperature.

Re-wetting Velocity: The velocity of progression of the re-wetting front on the heated surface area is known as Re-Wetting velocity.

Wetting Delay/Re-Wetting time/Resident time: The duration of time from the starting of the impingement and the re-wetting to occur is known as Wetting delay or Re-Wetting Time.

For jet impingement, whenever a surface is heated to a certain degree of Temperature, cooling takes place by both natural and forced convection. Once the coolant contact with the surface is established, the cooling process begins. To cause early re-wetting, various modification of flow, surface, and coolant has been studied over the years. Surface modifications achieved by a change in geometry have seen little interest from the researchers. Disruption in the flow to cause turbulence leads to the early occurrence of the re-wetting. Hence, higher cooling rates are achieved. With that in mind, H. Leocadio et al. [13] studied the effects of rough elements on a stainless-steel surface and summarized that the induction of rough elements causes better cooling due to those elements acting as micro fins. Singh et al. [14] studied the effect of an array of roughness elements of cylindrical, cubic, and concentric shapes on a surface. The concentric-type elements resulted in the highest heat transfer rate showing an increment of 20-60%. Similarly, 20-40% for cubic and 10-30% for the cylindrical roughness elements in fin effectiveness were found when compared with the smooth surface. K Nagesha et al. [15] modified a heated surface with V-grooves/multi-protrusions. The heat transfer was characterized for Reynold's Number in the range 10000 to 27500. Due to an increase in the surface area caused by roughening the surface, enhancement in heat transfer over that of a flat plate for V-groove-type surface elements was found. Tepe et al. [16] studied the effects of extended jet holes on a rib-roughened surface. An increment of up to 40.32% in heat transfer was found due to the use of extended jet holes. Kim et al. [17] optimized the plate fin height and found a 50% increment in the fin heat sink performance than the conventional heat sink. To optimize the cooling performance, the study of the quenching process is often complimented by the study of pressure drop across the channel since pressure drop is related to the pumping power required. Han et al. [18] studied the heat transfer augmentation for a 90° and 45° attack angled ribs and concluded that for the same heat transfer through the target surface, the angled ribs had a low-pressure drop than the orthogonal rib. Jing et al. [19] did a numerical investigation for concave dimples and protrusions. The dense arrangement of the protrusions resulted in better heat transfer. Xie et al. [20] observed in their study that no flow separation happened in the dimpled space and reported an enhancement of up to 50% in Nusselt number in one of his arrangements. Kanokjaruvijit and Botaz [21-23] conducted a lot of experimental studies studying three impingement arrangements. Reynolds number was varied from 5000 to 11500, jet spacings(z) to diameter ratio(d) of 2,4,8,12 for varying depths(t) of dimples(t/d=0.15,0.25,0.29). Jet diameter(d) to dimple diameter(D) ratios were varied from 0.25 to 0.5 to 1.15. Hemispherical dimples and cusped elliptical dimple surfaces were studied for these experiments. They concluded that the dimpled surfaces lead to better heat transfer performance when compared with that flat surfaces and cusped elliptical dimples. This happens since a dimpled surface induces higher energetic vortices. Schukin et al. [24] studied hemispherical dimples for heat transfer enhancement. It was reported that by using dimples, there was an increase in heat transfer. Ekkad and Kontrovitz [25] studied the effect of rotation of dimpled and smooth surfaces under rotation. The rotation enhanced the heat transfer in the case of the smooth surface in leading and trailing portions but was found to have a reverse impact on the dimpled surface leading to a decrease in heat transfer. Vinze et al. [26] studied the effect of a dimpled surface on heat transfer for multiple jets. Reynolds's number was kept between 5000-40000. For d= diameter of the jet, the effect of various dimple depths (0.25*d* and 0.5*d*) and dimple pitch (*p*=3*d,* 4*d,* and 5*d*) is also studied. For greater pitches($p \geq 4d$), better heat transfer characteristics were recorded for the dimpled surface than the smooth surface. In another study, Wright et al. [27] presented a study on single and multi-jet arrays for heat transfer enhancement. The Parametric analysis for the surface-to-jet spacing and jet velocity was reported. Kishan et al. [28] in their CFD analysis also performed an analysis on the effect of the curvature on the boiling heat transfer. For the case of curved surface, Rakhsha et al. (29) in their parametric study analyzed the pulsating effect of cooling on a concave surface, comparing the results with the steady jet. The author group reported an enhancement in Nusselt Number by 22% for their case. Similarly, Ataei et al. (30) in their study, studied heat transfer enhancement for double pulsating jet on a flat surface, reporting the increment in average Nusselt Number. However, the pulsating jet requires higher energy consumption and must be taken into account too. This led to the present study with following objectives.

2. Motivation and Objectives

It is evident after going through the literature, a lot of research is conducted related to the effect of dimple configuration. Various dimple configurations along with a change in various parameters like Reynolds number, pitch, and depth have been explored. Along with that, citing a change in properties of the surface, a very little number of studies and comprehensive reviews were found about the re-wetting behavior progress when a material is subjected to repeated cooling. So, an experimental setup was fabricated with the following objectives:

a) To study the re-wetting progress for a surface when subjected to repeated heating and cooling.

b) To study the effect on the rewetting behavior on a flat smooth plate and compare it with that of a dimpled surface of varying depths.

c) To perform the heat transfer efficiency on the flat plate when heated up to 300° C.

3. Experimental Setup and Procedure

3.1 Preparation of the temperature controller assembly

For heating the surface, four cartridge heaters of 500 watts capacity are attached in parallel. The plate surface is heated from the bottom. Since, temperatures as high as 400 °C are involved, a failsafe mechanism for safety must be used. For that purpose, by employing various electrical and electronic devices, a connection board is prepared. The Miniature Circuit Breaker (MCB) is used to prevent damage to the setup due to overcurrent. Equipped with a K-type thermocouple, a thermostat (REX-C100) controller is used to maintain the initial surface temperature to our desired value once it is reached. A Solid-State Relay device is connected to the thermostat which is then used to operate the cartridge heaters. The cartridge heaters are then connected to the switch plug. All these devices are attached in the illustration as shown in Figure 1.

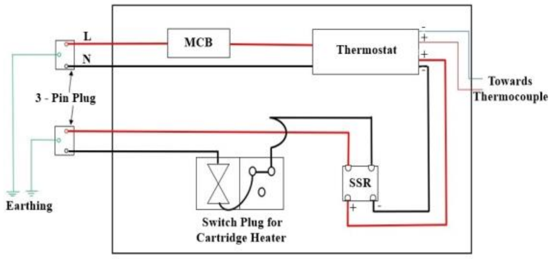


Figure 1. Illustration of the connection board

3.2 Test Sections and Initial Conditions

A detailed schematic along with a photograph of the setup is provided in Figure 2. A plane flat and dimpled aluminum plate 25 × 25 cm^2 of 4mm thickness is used as the test section. The smooth plate is heated repeatedly to three different initial temperatures of 300, 350 and 400°C.

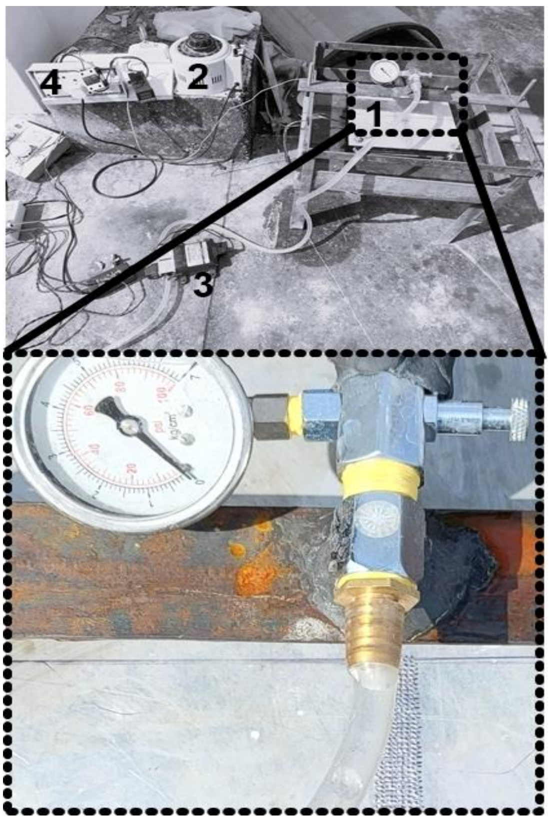


Figure 2. Experimental Setup (1) Test plate (2) VARIAC connection (3) Water Pump (4) Thermostat connection board

Furthermore, the effect on the heat transfer due to dimple depth is investigated. For that purpose, a new flat plate of the same material is used. Dimples of thickness 0.5mm and 1mm are prepared and the results are reported. Eight K- type thermocouples at eight different locations are attached under the plate with a spacing of 8mm between them. Water at a temperature of 17 ±2°C is used as the coolant. A water pump of capacity 2 hp attached with a nozzle of diameter 5mm is used for impinging water onto the heated surface. The plates are insulated from all four directions and the bottom direction as well to avoid heat losses due to convection. A data logger (HUATO S220-T8) is used to record the temperatures at fixed intervals. The cooling graph is plotted, and the results are then interpreted in terms of re-wetting temperature with wetting delay.

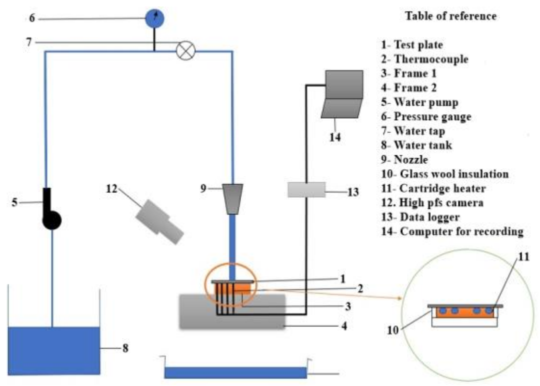


Figure 3. Representation of the experimental setup

Table 1. Boundary Conditions

Boundary Conditions	
Coolant	Water
Coolant initial temperature	17 ±2°C
Initial surface temperature	300,350,400°C
Mass flow rate	0.0015 kg. /sec.
Initial pressure of the coolant	20kPa
Distance from the jet to the plate surface	17mm
Position of the impinging jet	Centre of the plate

3.3 *Experimental Procedure*

A U- tube mercury manometer in conjunction with a control valve was used on both sides of the water pump to regulate the mass flow rate. A control valve was used to increase or decrease the flow rate. The manometer was then used to measure the pressure difference by knowing the difference in height between both columns of the manometer. This knowledge of the pressure difference was then used to calculate the mass flow rate. The control valve was then set, and the experiment was performed.

See Figure 3 for the Representation of the experimental setup. A standard 230V household power supply is used for both the thermostat and the cartridge heating element. The underside of the plate is drilled to a depth of 3.8 mm, and thermal wax is used to position and secure the thermocouples. The top surface temperature of the plate is determined by the temperature measured by thermocouples. Here, the thermostat acts as the temperature controller and is responsible for maintaining the set temperature value. Once the set temperature is achieved, the water pump is turned on and the power supply to heaters is cut-off. The temperature value is recorded at a time interval of two seconds during the cooling process.

Table 2. Accuracy of the instruments

Name of the instrument	Specification	Accuracy
Data logger	0°C -1350°C	±0.2% ±0.8°C
Thermocouples	0°C -1200°C	±0.1%
Thermostat	0°C -1300°C	±0.5%

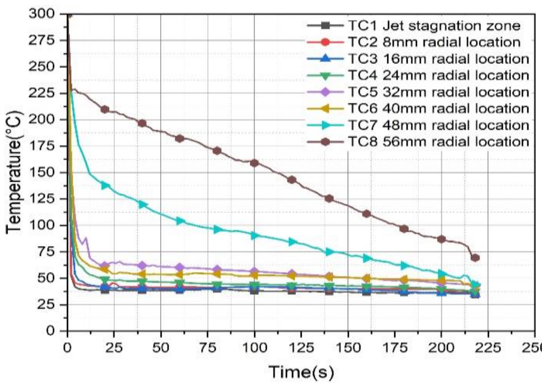
The uncertainty of the experimental reading is a must to take into account the estimated accuracy of the procedure. Table 2 provides the uncertainty associated with each instrument. Following the schematic in Figure 1, the data logger, thermocouple, and thermostat have been joined in a series connection. Hence, for our system, this has been done according to the method provided by Kline and McClintock [31]. The uncertainty with our temperature readings was within the range of ±0.8%. This shows that the results are fairly accurate and well within the acceptable range. Moreover, the accuracy level associated with various formulas presented in section 4.3 for heat transfer calculation has been published based on experimental measurements that vary from 15-20% in degree of accuracy.

4. Results and Discussion

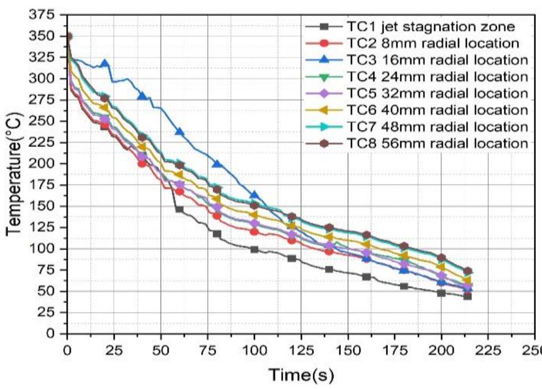
4.1 *Smooth plate subjected to repeated heating and cooling*

A smooth aluminum flat plate is subjected to repeated cooling at three different temperatures 300,350 and 400°C. The cooling curve graphs obtained are plotted as shown in Figure 4. It shows the re-wetting progress of the same test plate when subjected to 300°C, 350°C, and 400°C as initial temperatures. When the coolant first encounters the heated plate, the coolant forms a small region (around two to three times the diameter of the jet).

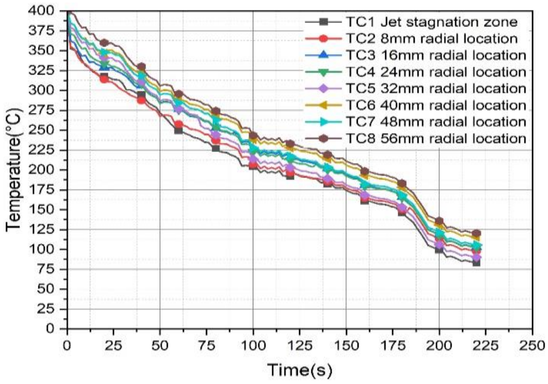
Pretty soon, a circular wet patch is formed in the jet stagnation zone. It is observed that the wet patch progresses with time with its area continuously increasing uniformly in all directions. The formation of this wet patch is depicted by the sharp drop in the cooling curve graph (Figure 4(a)). This sharp drop observed on the curve is the Re-wetting Temperature point. A significant increase in the cooling time was recorded when the flat plate was subjected to a consecutive cooling process. This observation was driven by two factors. Firstly, an increment in the initial temperature by 50°C for every observation recorded, the energy removal to cool the plate surface back to the initial temperature increases (proportional to the temperature). With all other parameters like mass flow rate, jet diameter, and jet-to-plate distance kept constant, it will take more time to cool down the plate which is here observed. Secondly, the subsequent heating of the same plate surface changes the metallurgical properties of the surface which causes a change in the behavior of the cooling characteristics. Moreover, the bending of the plate which was observed after heating for the first time leads to uneven distribution of the coolant for the next trial. This effect was observed at an elevated rate for the third trial. This uneven bending of the plate surface led to non-uniform cooling curves. Moreover, the rate of cooling observed at the start of the impingement is much higher than that observed at the later stages. This happens due to the reason that once the wetting zone is formed, the coolant layer present on the plate will cause the further incoming flow of coolant to slip away. For the same mass flow rate, this leads to less heat removal. This effect can be visualized in Figure 4.



(a) Initial temperatuire, 300 °C



(b) Initial temperature, 350°C



(c) Initial temperature, 400 °C

Figure 4. Cooling graph when the smooth flat plate is subjected to cooling from (a) 300 °C; (b) 350°C; (c) 400 °C

4.2 Dimpled plate subjected to cooling

Two hemispherical dimpled surfaces are prepared on the aluminum plate. They are subjected to cooling from the initial temperature of 300°C. With the same boundary conditions as before, both the horizontal and the vertical pitch are kept at 3mm, and the diameter of the dimple is kept at 2mm. The Cooling graphs obtained are plotted below. Here, the dimpled surfaces with 1mm and 0.5mm depth are used i.e., the diameter of the nozzle(d) to depth (t) ratio of 10 and 5 are investigated and reported.

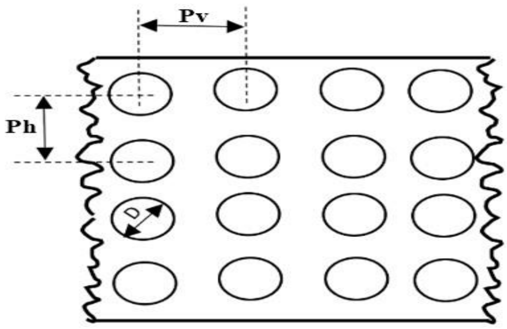
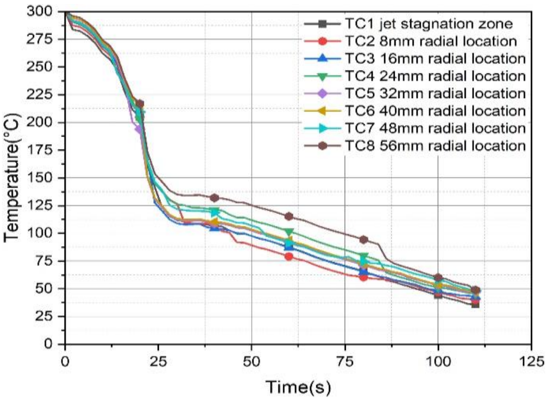
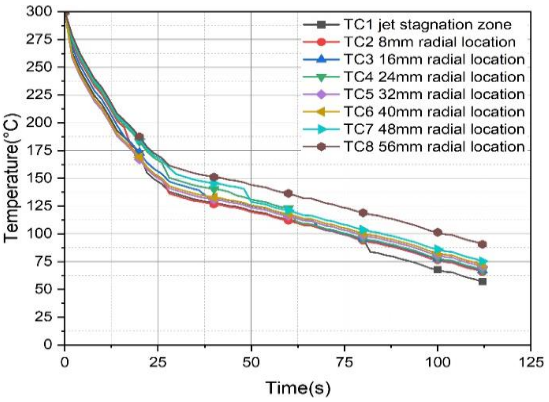


Figure 5. Schematic of the Dimpled plate
(P_h : Horizontal pitch, 3 mm; P_v: Vertical pitch, 3 mm;
and D: Diameter of dimple, 2 mm)

On first look, the time taken to cool down the dimpled surface was less compared to that of the smooth flat surface. From the previous literature, it is known that the dimples act as an obstruction and cause early turbulence. This leads to early breakage of the laminar sub-layer and formation of the highly energetic vortices. These high energy vortices result in a higher heat transfer rate, thus a lower time to reach the same temperature as compared to a smooth surface. As a result of which, higher re-wetting temperatures and shorter wetting delays are observed. The trends were in line with the observations from [21-24] and can also be observed by comparing Figure 4(a) with Figure 6.



(a) Dimple depth, 0.5 mm



(b) Dimple depth, 1 mm

Figure 6. Cooling graph when the dimpled flat plate is subjected to cooling from 300 °C with dimple depth (a) 0.5 mm; (b) 1 mm

For the diameter of the jet(d) and the depth of the dimple(t), the dimple depth of d/t=10 and 5 are investigated. Dimples with 0.5mm thickness (d/t=10) have shown a higher heat transfer rate than the ones with 1mm thickness(d/t=5). Levels to reach the temperature of 100°C are compared for both cases. For the initial temperature of 300°C to reach the 100°C temperature levels, a maximum of 40% and a minimum of 26% reduction in time has been observed for the jet stagnation zone and the thermocouple at the 56mm location (farthest from the stagnation zone). The reason behind this observation is twofold. Firstly, there is a loss in momentum of the coolant when it strikes the surface. With the presence of deep dimples on the heated surface, a comparative more mass flow rate of coolant is required for the same level of cooling. Secondly, this loss in momentum due to deep dimples causes more resistance to the incoming flow of fresh coolant. Thus, more time is required to cool to the same level of temperature. Better heat transfer characteristics are observed for d/t=10. Here too, the same result of a slower heat removal rate at the later stages is seen. The cooling rates are lesser at the later stages of the impingement as compared to the early stages.

4.3 Heat transfer Efficiency

After obtaining the data from the experiment, four boiling regimes were identified during the cooling of the hot flat plate namely the film boiling regime, transition boiling regime, nucleate boiling regime, and single-phase convection regime. General empirical co-relations for all four boiling regimes were identified in the error band of ±15% and then compared with the total heat loss by the plate.

Liu and Wang [32], who performed a theoretical and experimental study on film boiling heat transfer for a hot horizontal flat plate, the heat flux in the said regime is approximated as equation 1:

$$q_{\mathrm{f}} = \left(\frac{\Delta T_{\mathrm{sat}}\lambda_{\mathrm{v}}}{d}\right)^{\frac{3}{4}} \left(\frac{\frac{1}{3}h_{\mathrm{fg}}V_{\mathrm{s}}Re_{l}\mu_{\mathrm{l}}}{v_{\mathrm{v}}}\right)^{\frac{1}{4}} \tag{1}$$

According to Marie et al. [33], The co-relation for heat flux in transition boiling regime is obtained analytically as (equation 2):

$$q_{tr} = 1.6510^{4}\frac{V_j^{0.83}}{d^{0.42}}(1 + 0.235\Delta T_{\mathrm{sub}}) \tag{2}$$

Similarly, Rohensow et al. [34] proposed the relation for heat flux during nucleate boiling (see equation 3):

$$q_{\mathrm{n}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{\frac{1}{2}}\left[\frac{C_p(T_s-T_{\mathrm{sat}})}{C_{sf}h_{fg}\mathrm{Pr}_l^n}\right]^3 \tag{3}$$

For Single phase convection heat transfer, the heat flux was determined according to equation 4 as:

$$\mathrm{qs} = \frac{\mathrm{mcp(Tsat - Tcoolant)}}{\mathrm{As}} \tag{4}$$

The heat fluxes associated with various boiling regimes are stated above and cumulative, they tell about the total heat flux required to be removed to achieve the desired temperature and can be written as:

$$\mathrm{qt = qf + qtr + qn + qs} \tag{5}$$

The total heat transfer to cool a surface from an initial surface temperature of 300 degrees to the coolant temperature can be written as (see equation 6):

$$\mathrm{qal} = \frac{\mathrm{mcp(Tini - Tcoolant)}}{As} \tag{6}$$

Heat loss to the surroundings is then written as:

Heat loss= q_{al}- q_t

Heat transfer Efficiency= $\frac{Heat\ loss}{Heat\ supplies} * 100$

A 59.76% of heat transfer efficiency was recorded in the said experiment after going through the methodology. This happens primarily due to two reasons. First, a lot of splashing was observed during the performance of the experimental setup. This is directly related to the wasted potential of the coolant which could have been used to cool down the plate. Secondly, Heat losses to the surroundings also play a dominant role when dealing with temperatures as high as 300 degrees. Methods to prevent these losses such as designing insulations by considering both the economic aspect and the power requirement are important.

Further, a variation in the water pump pressure with pressures 81.325kPa (actual pressure used for the experimental setup), 86.325, 91.325, 96.325, and 101.325 kPa were also recorded and plotted. An increase in heat transfer efficiency was seen (see Figure 7). However, this increment was very small compared to the increase in heat transfer efficiency. The added pumping power hence is not justified.

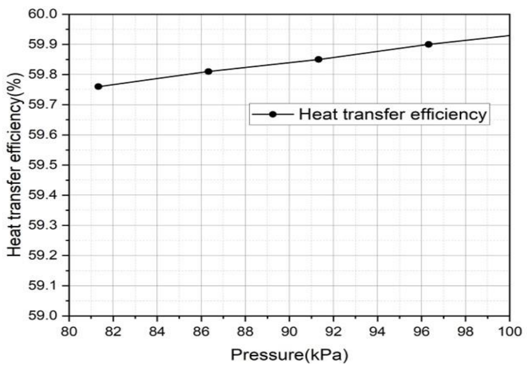


Figure 7. Heat transfer efficiency with variation in pump pressure

Conclusions

Re-wetting progression in aluminum flat plates of dimensions 25 ×25mm^2 with 4mm thickness is studied. The same smooth plate surface is subjected to cooling from the initial surface temperature values of 300,350 and 400°C. The dimpled surfaces of depth 0.5mm(d/t=10) and 1mm(d/t=5) are cooled from the initial surface temperature of 300°C. After going through the above experiment, the following conclusions were observed:

- The presence of dimples causes early turbulence. This leads to early breakage of the laminar sub-layer and higher heat transfer rates. The d/t case of 10 was reported to have the highest value of heat transfer rate.
- A maximum and minimum of 40% and 26% increment in the heat transfer rates were found for the d/t case of 10 as compared to that of 5 at the jet stagnation zone and 56mm location away from the jet stagnation zone (farthest point).
- Under repeated heating, the change in the properties and bending of the surface cause change in re-wetting behavior.
- A 59.76% of heat transfer efficiency was recorded for the experimental setup and this efficiency was found to be increasing with an increase in the water pump pressure.

Although the changes under consecutive quenching are attributed to the change in properties, what metallurgical properties cause these deviations? How will these properties behave with various surface modifications? Moreover, the trend observed due to dimples force us to think of the optimum value of the depth to achieve maximum heat transfer rate. These questions need to be further answered and can be considered as future work in continuity with this experiment.

Nomenclature

q	Heat flux [W/m^2]
ΔT	Temperature difference
λ	Thermal conductivity [W/m.K]
d	Diameter of jet [m]
h_{fg}	Latent heat of vaporization [kJ/kg]
t	Depth of the dimple [m]
V	Velocity at the stagnation zone [m/s]
Re	Reynold's number
C_{sf}	Experimental constant that depends on a surface fluid combination
n	Experimental constant that depends on the surface
Pr	Prandtl number
A_s	Surface area [m^2]
c_p	Specific heat [kJ/kg-K]
h	Heat transfer Coefficient [W/Mk]
μ	Dynamic viscosity [Pa- s]
g	Acceleration due to gravity
σ	Surface tension of liquid-vapor interface [N/m]

Symbols

f	Film boiling regime
tr	Transition boiling regime
n	Nucleate boiling regime
s	Single-phase convection regime
sat	Saturation
sub	subcooling
v	vapor
j	jet
l	liquid

Conflicts of Interest

The autor declares no conflict of interest.

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