

Experimental Investigation of Heat Transfer Enhancement Using Impingement of Multiple Water Jets

Niranjana Murthy,¹ V. Krishnan,¹ and A. Chennakesava Reddy²

¹Department of Mechanical Engineering, M.S. Ramaiah Institute of Technology, MSR Nagar
Vidhya Soudha Bangalore, 560054 India

²Department of Mechanical Engineering, JNTU Anantapur, India

The problem of cooling electronic components has become a subject of special interest in recent years due to the increasing capacity and rapidly decreasing size of electronic components. Direct contact cooling using multiple jet impingement is considered the most effective method. The heat transfer problem is complex and a better understanding of the jet impingement method is essential for the proper application of this method for electronic cooling. Investigations were carried out using an electrically heated test plate. Heat flux in the range of 25 to 200 W/cm², which is a typical requirement for cooling high power electronic components was dissipated using 0.5-mm diameter water jets arranged in a 7 × 7 array with a pitch of 3 mm. Temperature difference between the test plate and water was within 30 °C. Tests were performed in the flow rate range of 22 to 40 ml/min, resulting in a Reynolds number range of 1100 to 1750. Results show a significant increase in the heat transfer coefficient or Nusselt number with an increase in heat flux. The effect of the flow rate or Reynolds number on the heat transfer coefficient is found to be negligible. © 2010 Wiley Periodicals, Inc. *Heat Trans Asian Res*, 39(4), 222–231, 2010; Published online 7 April 2010 in Wiley InterScience (www.interscience.wiley.com). DOI 10.1002/hjt.20291

Key words: heat transfer enhancement, heat transfer

1. Introduction

With the advances in electronics and communication technology, smaller and more powerful components are being introduced in the market. The demand for high-performance electronics has increased; several applications require electronic components to be faster, smaller, able to handle a higher amount of power, and be more reliable. Small size and high power unfortunately lead to higher heat fluxes that need to be removed from the components to avoid high temperatures and failure. The requirements of high power dissipation and miniaturization demand cooling rates which cannot be obtained using traditional cooling methods such as forced convection, boiling, and evaporation. Several methods have been developed to meet the present day demands of high cooling rates. Direct contact cooling using multiple jet impingement is considered the most effective solution. The multiple jet impingement heat transfer problem is complex and a systematic study to assess its effectiveness for cooling electronic components is essential.

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Impinging jets for direct liquid cooling of microelectronic components has been investigated by many researchers. Elison and Webb [1] experimentally investigated the heat transfer associated with a single water jet. Authors have tested three different nozzles having jet diameters of 0.584, 0.315, and 0.246 mm. The nozzles were long enough to allow a fully developed velocity profile in any regime. The range of the Reynolds number (Re) was between 300 and 7000. The Nusselt number (Nu) varied as $Re^{0.8}$, whereas the previous studies [1] have shown that (Nu) is proportional to $(Re)^{0.5}$. Authors have attributed the enhancement in heat transfer to the surface tension effects at the nozzle exit, which is found to increase with an increase in jet diameter.

Fabbri and colleagues [2] conducted a study of cooling using both jets and sprays. The authors have investigated the performances of both sprays and micro jets and a comparison of results has been presented. The following conclusions were made: (a) micro-jet arrays can have the same or better performance compared to jet sprays with similar flow rate, but with lower pumping power; (b) for the same pumping power and wall-to-liquid temperature difference of 76 °C, the jets could remove heat fluxes as high as 240 W/cm², while the spray could remove only 93 W/cm²; (c) in practice, there is always a combination of jet diameter and jet spacing that can yield the same heat transfer coefficients as that of the spray, but at much lower pumping energy costs; and (d) the highly populated micro-jet arrays were much more preferable than the arrays with few and large jets, because they needed a much lower flow rate to obtain the same heat removal rate.

Jiji and Dagan [3] studied the heat transfer associated with multiple single-phase free surface jets impinging on an array of microelectronic heat sources. They used square arrays of 1, 4, and 9 jets using FC 77 as the cooling fluid on multiple heat sources of size 12.7 × 12.7 mm² and found that the surface temperature uniformity improved as the spacing between the jets decreased. It was also observed that multiple jets impinging on a heater surface can improve the spatial uniformity of the heat transfer coefficient on the surface.

Womac et al. [4] performed experiments with 2 × 2 and 3 × 3 jet arrays using water and FC77 as cooling fluids. The impinging surface made of copper was 12.7 × 12.7 mm². Tests were conducted with jet diameters of 0.513 and 1.02 mm and pitches of 5.08 and 10.16 mm. It was found that for a given flow rate, the heat transfer improved with the increase in jet velocity. It was also observed that the reduction in heat transfer that occurs with lowering the flow rate becomes more pronounced at very low flow rates and the authors have attributed this effect to the bulk heating of the fluid.

Wang et al. [5] used micro-jet heat sinks to cool VLSI chips. Micro-jet heat sinks can improve temperature uniformity in the presence of chip hot spots. Experimental comparisons of micro-jet and micro-channel performance showed that micro-jets have improved thermal uniformity. The results suggested that micro-jet arrays are the preferred micro-heat sinks for effective device cooling.

Fabbri et al. [6] have used a circular array of free surface micro jets. Experiments were conducted by employing jet pitches of 1, 2, and 3 mm and jet diameters of 50, 100, 150, and 250 μm. Deionized water and FC40 were used for cooling. The jet Reynolds number range was varied between 90 and 2000 and the Prandtl number was varied from 6 to 84. Heat flux up to 250 W/cm² could be removed with water. The authors have observed that the Nusselt number is strongly dependent on the Reynolds number.

Shang et al. [7] investigated heat transfer from single-phase and two-phase boiling free jet impingement. Experiments were carried out using a free jet impinging on the surfaces of 2- and 3-mm-thick film resistor. Effects of jet velocity and sub-cooling were studied. It was found that higher velocities resulted in higher heat transfer coefficients and the jet diameter had very little effect in single-phase heat transfer. However, higher velocities and larger jet diameters resulted in higher heat transfer coefficient in boiling heat transfer.

Lin et al. [8] experimentally studied the performance of confined slot jet impingement for an electronic cooling application. They explored the effects of jet Reynolds number and jet separation distance on heat transfer in a heated target surface. Authors have found that the effect of jet separation distance on the stagnation, local, and average Nusselt number was insignificant and the heat transfer increased with an increasing jet Reynolds number.

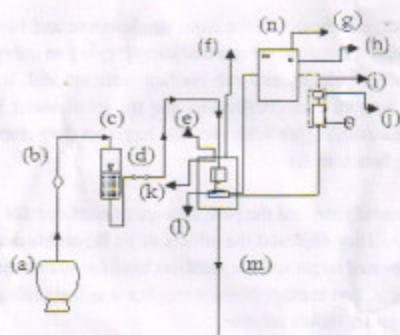
The problem of cooling by jet impingement is complex because of several parameters affecting the heat transfer process. Systematic study of the effect of various factors on the heat transfer phenomena is essential to understand the cooling process. Parametric investigation of cooling of a 2×2 cm² heated copper plate has been carried out using a 7×7 array of multiple water jets. The test plate was selected to simulate the cooling requirement of a typical electronic device used for example in telecommunication equipments or power electronics. The test plate also simulates a triangular or square fin mounted on a tube carrying a very hot fluid or a molten metal. The jets have a diameter of 0.5 mm and a pitch of 3 mm. Tests were conducted in the flow rate range of 22 to 34 ml/min. Heat flux and the distance between the test plate and nozzle were varied in the range of 25 to 200 W/cm² and 10 to 20 mm, respectively.

Nomenclature

- A: hot plate surface area, cm²
d: nozzle diameter, mm
h: heat transfer coefficient, W/cm²°C ($q/(T_c - T_w)$)
Nu: Nusselt number, hd/k
P: total heat transfer, W
q: heat flux, W/cm² (P/A)
Q: total flow rate, ml/min
Re: Reynolds number, Vd/ν
T_b: bulk fluid temperature, °C
T_c: chip surface temperature, °C
T_w: inlet water temperature, °C
V: jet velocity, m/s
 ν : kinematic viscosity, Ns/m²
z: nozzle height from chip surface, mm
 ΔT : $(T_c - T_w)$, °C

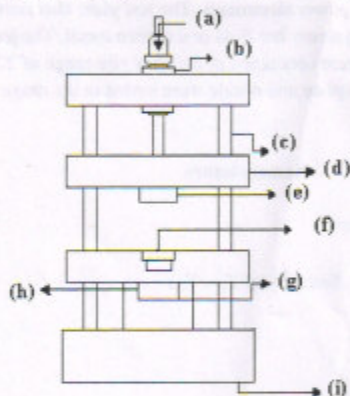
2. Experimental Apparatus and Test Procedure

The test apparatus is shown schematically in Fig. 1. The apparatus is designed and fabricated to carry out tests using different types of nozzles. The arrangement consists of an air compressor, a



(a) Schematic arrangement

- (a) Compressor (b) Filter (c) Auxiliary Reservoir (d) Flow control valve (e) Test stand
 (f) Nozzle head (g) Control unit (h) Dimmer stat (i) Variable voltage Transformer
 (j) AC power supply (k) Copper chip (l) Heater block (m) Drain out (n) Display unit



(b) Heater block assembly

- (a) Water inlet (b) Circular scale (c) Supporting rods (d) Movable plate (e) Nozzle head (f) Copper hot plate
 (g) Thermocouples (h) heater block (i) Base tray

Fig. 1. Schematic diagram of the experimental setup.

fluid delivery system, a heater assembly, a jet nozzle head, and the test stand. The fluid delivery system consists of a water reservoir, auxiliary reservoir, flow control valves, pressure gauge, filter, and piping systems. The auxiliary reservoir acts as a buffer and smooths out flow fluctuations to provide steady flow at the nozzle. It also serves to achieve fine control over the flow rate. A safety valve is provided to prevent excess pressure build-up in the system.

The heater assembly consists of a hot plate, heating element, thermocouples, variable voltage transformer, and a control-display system. The hot plate represents the surface of a typical electronic component and is made of copper. Copper was selected because of its high thermal conductivity. The hot plate is 20 mm × 20 mm in size with a thickness of 1 mm. The hot plate was mounted on the heating element. The heating element is a Nichrome wire of 16 gauge, 2 ohm, and wattage capacity of 1 kW. The power to the heater was controlled using a variable voltage transformer. Two thermocouples were mounted underneath the hot plate on the center line and insulated with ceramic insulation. These thermocouples also provide an indication of surface temperature uniformity on the plate. The complete heater assembly was mounted and insulated using a Teflon jacket. The leads from the thermocouples were connected to the control display system. The functions of the control and display system includes: (a) to vary the heat input to the hot plate using the transformer, (b) to display the hot plate surface temperatures, input voltage, and current using a digital temperature indicator, voltmeter, and ammeter, and (c) limit the maximum surface temperature and automatically cut off the power supply when the hot plate temperature exceeds a set value.

The jet nozzle head is made of stainless steel and it consists of the nozzle chamber and nozzle plate. The nozzle chamber was connected to the reservoir through a connecting tube. The nozzle plate is made of 3-mm-thick stainless steel plate. The nozzle plate was designed to cover the nozzle chamber making it a single leakproof unit. The nozzle plate had 49 holes, each 0.5 mm in diameter, which are laser drilled and arranged in a square array of 7 × 7 with a pitch distance of 3 mm between the holes. The distance between the nozzle plate and the hot plate surface can be varied in the range of 10 to 20 mm. The test stand consists of a base tray, mounting plate, hot plate, movable nozzle plate, and a top plate held together by vertical support rods. The nozzle head is attached to the nozzle plate which could be moved vertically. A calibrated screw thread assembly is provided along with a circular scale on the top plate. The nozzle plate can be fixed at the desired height by accurate positioning of the calibrated screw head.

The hot plate surface is cleaned to remove residual adhesive stains and dust on the surface before starting the experiment. Compressed air is passed through the tube connecting the reservoir and the nozzle head to remove any dust particles which could block the nozzle. The filter is installed in the flow line prior to the auxiliary reservoir. The flow rate, power input, and distance between nozzle exit and test hot plate were varied during the experiments. The hot plate is allowed to reach a steady state before the acquisition of test data on water flow rate, power dissipation, and temperatures from thermocouples. Experiments were conducted by positioning the jets and the hot plate in both horizontal and vertical positions.

3. Results and Discussion

The heat transfer enhancement using impingement of multiple water jets is complex because of several parameters affecting the heat transfer process.

Figure 2 shows the variation of heat transfer coefficient (h) with heat flux for various flow rates. It is observed that (h) increases significantly with heat flux at all flow rates. Similar trends in the variation of (h) with heat flux have been observed with different values of (Z) for both horizontal and vertical positioning of the jets and the test plate. The heat transfer results have been plotted in the non-dimensional form, $\ln(Nu/Re^{0.25})$ versus $\ln(q)$, in Figs. 3 to 6 by varying the parameters

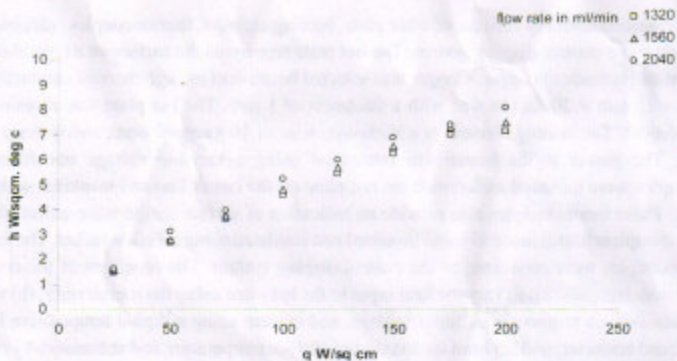


Fig. 2. Variation of heat transfer coefficient with heat flux, $Z = 20$ mm, horizontal position.

independently. The effect of (Z) and the horizontal or vertical positioning of the jet and the test plate seems to be less important compared to the effect of heat flux.

Figure 7 is a composite plot of the heat transfer results obtained by the variation of (a) heat flux, (b) flow rate, (c) distance between the test plate and nozzle exit, and (d) positioning of the jets and the test plate in the horizontal and vertical position. It is interesting to observe that the $\ln(Nu/Re^{0.25})$ versus $\ln(q)$ plot has a relatively small scatter about a mean line and the overall heat transfer results can be represented by the relationship given below.

$$Nu = 0.17q^{0.8}Re^{0.25}$$

This correlation relating the Nusselt number, heat flux, and the Reynolds number shows that heat flux is a dominating factor which determines the heat transfer coefficient and the Reynolds

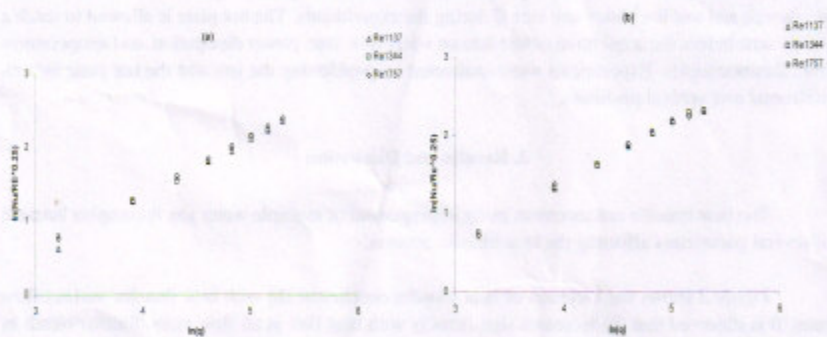


Fig. 3. Variation of $Nu/Re^{0.25}$ with heat flux for various Reynolds numbers, horizontal position: (a) $Z = 10$ mm, (b) $Z = 20$ mm.

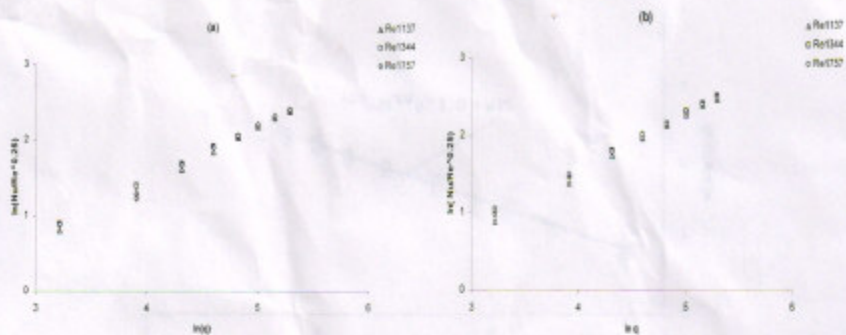


Fig. 4. Variation of $Nu/Re^{0.25}$ with heat flux for various Reynolds numbers, vertical position: (a) $Z = 10$ mm, (b) $Z = 20$ mm.

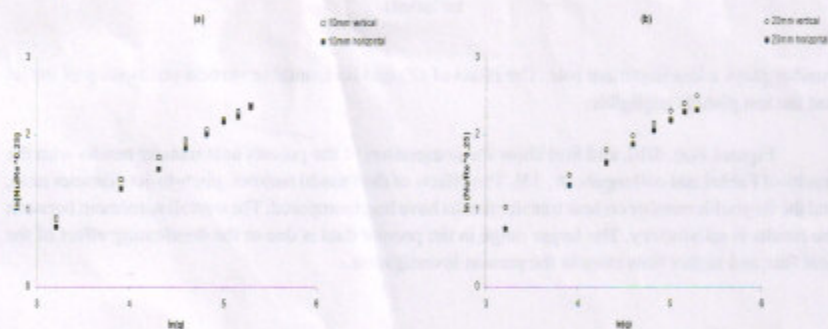


Fig. 5. Variation of $Nu/Re^{0.25}$ with heat flux, $Re = 1344$, jet in vertical and horizontal position: (a) $Z = 10$ mm, (b) $Z = 20$ mm.

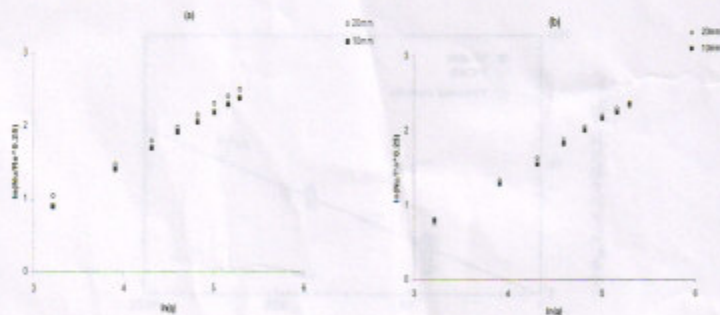


Fig. 6. Variation of $Nu/Re^{0.25}$ with heat flux, $Re = 1344$: (a) $Z = 10$ and 20 mm vertical position, (b) $Z = 10$ and 20 mm horizontal position.

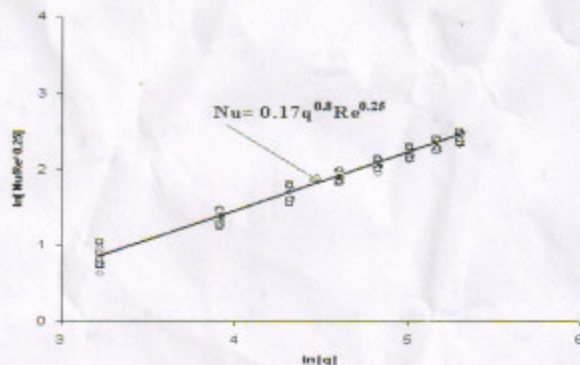


Fig. 7. Variation of $Nu/Re^{0.25}$ with heat flux, various Reynolds numbers, and (Z) Reynolds number range = 1100 to 1750, $Z = 10$ and 20 mm (data for both vertical and horizontal positioning of jets included).

number plays a less important role. The effect of (Z) and horizontal or vertical positioning of the jet and the test plate is negligible.

Figures 8(a), 8(b), and 8(c) show the comparison of the present heat transfer results with the results of Fabbri and colleagues [6, 15]. The effects of the Prandtl number, pitch-to-jet diameter ratio, and the Reynolds number on heat transfer results have been compared. The overall agreement between the results is satisfactory. The larger range in the present data is due to the dominating effect of the heat flux and higher flow rates in the present investigation.

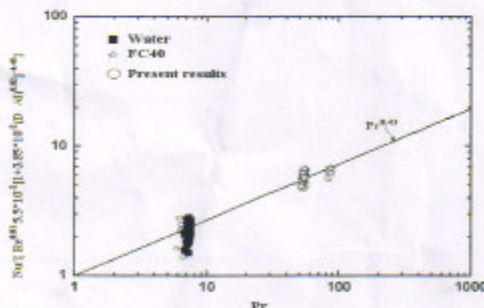


Fig. 8. (a) Comparison of the present heat transfer results with the data available in the literature [6].

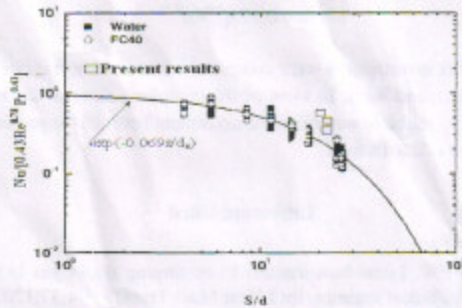


Fig. 8. (b) Variation of $Nu_{jet}/[0.043Re^{0.78}Pr^{0.48}]$ with Prandtl number for varying Reynolds numbers and ratio of pitch to jet diameter. Comparison of the present results with the data available in the literature [15].

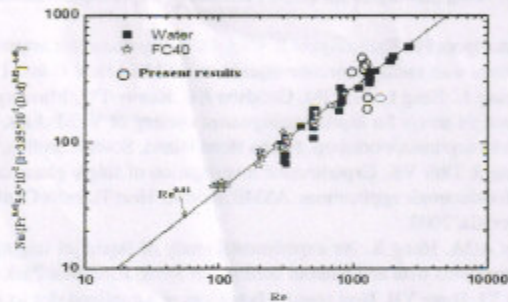


Fig. 8. (c) Comparison of the present heat transfer results with the data available in the literature [6], effect of Reynolds number.

4. Conclusion

Experiments were conducted to study the enhancement of heat transfer using impingement of multiple water jets on an electrically heated test plate. Heat flux in the range of 25 to 200 W/cm², which is typical for high-power electronic components was dissipated using multiple water jets of 0.5 mm in diameter. The temperature difference between the test plate and water is within 30 °C. Tests were conducted by varying the heat flux, flow rate, distance between the heated test plate, and the nozzle exit and by keeping the jets and the test plate in both horizontal and vertical positions.

It is observed that the heat transfer coefficient is a strong function of heat flux. The effect of the Reynolds number is not significant in the range of testing. The effect of (i) the distance between the heated test plate and the nozzle exit and (ii) horizontal or vertical positioning of the jets and the test plates is found to be negligible. Overall, the heat transfer data can be correlated by the relationship

$$Nu = 0.17 q^{0.8} Re^{0.25}$$

The results of this investigation were compared with the heat transfer data available in the literature [Figs. 8(a), 8(b), and 8(c)]. In view of the complex nature of the heat transfer problem associated with cooling by multiple water jets, the agreement between the present results and the data available in the literature is satisfactory.

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