International Journal of Applied Engineering Research
ISSN 0973-4562 Volume 5 Number 21 (2010) pp. 3553–3564

http://www.ripublication.com/ijaer.htm

# Parametric Study of Heat Transfer Enhancement Using Impingement of Multiple Water Jets

**Niranjan Murthy[1], V. Krishnan[2] and A. Chennakesava Reddy[3]**

*[1]Senior Lecturer and [2,3]Professor*
*[1,2]Department of Mechanical Engineering, MSRIT, Bangalore-560054, India*
*[3]Professor, Department of Mechanical Engineering, JNTU, Ananthapur, India*
*[1]Corresponding Author E-mail: nrn_smit@yahoo.com*

**Abstract**

The problem of cooling of 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 as the most effective method. The heat transfer problem is complex and better understanding of jet impingement method is essential for proper application of this method for electronic cooling. Investigations were carried out using electrically heated test plate. Heat flux in the range of 25 to 200W/cm$^2$, which is a typical requirement for cooling high power electronic components was dissipated using 0.25mm and 0.5mm diameter water jets arranged in 7X7 array with a pitch of 3mm. Temperature difference between the test plate and water was within 30$^0$C. Tests were performed in the flow rate range of 14 to 40 ml/min. The distance between the nozzle and the test plate was maintained at 10 and 20mm with the nozzle placed in both in horizontal and vertical position. Results show significant increase in heat transfer co-efficient or Nusselt number with increase in heat flux. The effects of (i) Flow rate or Reynolds number (ii) Distance between the nozzle and test plate and (iii) The horizontal or vertical positioning of the nozzle are found to be small. A correlation has been developed to predict the heat transfer co-efficient with multiple water jet impingement. The heat transfer data from the present experiments agree satisfactorily with the data from similar experiments in the literature.

**Keywords:** Heat transfer enhancement, jet impingement cooling.

## 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 higher amount of power and 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 as the most effective solution. The multiple jet impingement heat transfer problem is complex and systematic study to assess its effectiveness for cooling electronic components is essential.

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. The authors have tested three different nozzles having jet diameters of 0.584, 0.315 and 0.246mm.The nozzles were long enough to allow a fully developed velocity profile in any regime. The range of Reynolds number (Re) was between 300 and 7000. Nusselt number (Nu) varied as Re $^{0.8}$, where as 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 increase in jet diameter.

***Matteo Fabbri, Shanjuan Jiang, Vijay K. Dhir*** [2] conducted a study of cooling using both jets and sprays. The authors have investigated the performances of both sprays and micro jets and the 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 $76^0$ C, the jets could remove heat fluxes as high as 240W/cm$^2$, while the spray could remove only 93 W/cm$^2$ (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) A 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 x 12.7 mm$^2$ 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 2x2 and 3x3 jet arrays using water and FC77as cooling fluids. The impinging surface made of copper was a 12.7x12.7 mm$^2$. 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 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 comparison 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 .Matteo Fabbri, Shanjuan Jiang and Vijay K. Dhir [6] have used a circular arrays of free surface micro jets. Experiments were conducted by employing jet pitches of 1, 2 and 3mm and jet diameters of 50, 100,150 and 250µm. De-ionised 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/cm$^2$ could be removed with water. Authors have observed that Nusselt number is strongly dependent on the Reynolds number.Yang Shang, Andrew A. O. Tay Xue Hong [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 2mm and 3mm thick film resistor. Effect of jet velocity and sub-cooling were studied. It was found that higher velocities resulted in higher heat transfer coefficients and jet diameter had very little effect in single phase heat transfer. However, higher velocities and larger jet diameters resulted in higher heat transfer co-efficient in boiling heat transfer.Lin et al [8] experimentally studied the performance of confined slot jet impingement for electronic cooling application. They explored the effects of jet Reynolds number and jet separation distance on heat transfer in 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 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 parameters on the heat transfer phenomena is essential to understand the cooling process. Parametric investigation of cooling of a 2X2 cm$^2$ heated copper plate has been carried out using a 7X7 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 equipment or power electronics. Experiments were conducted with jets having diameter of 0.25mm and 0.5mm arranged with a pitch of 3mm. Tests were conducted in the flow rate range of 14 to 34 ml/min. Heat flux was varied in the range of 25 to 200 W/cm$^2$.The distance between the test plate and nozzle is 10mm and 20mm. Tests were conducted by positioning the nozzle head in both horizontal and vertical positions. Heat transfer results were compared with data obtained from similar experiments in the literature.

## Experimental Apparatus and Test Procedure

The test apparatus is shown schematically as shown in Fig. 1. The apparatus is designed and fabricated to carry out tests using different types of nozzles. The arrangement consists of air compressor, fluid delivery system, heater assembly, 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 smooth out flow fluctuations to provide steady flow at the nozzle.

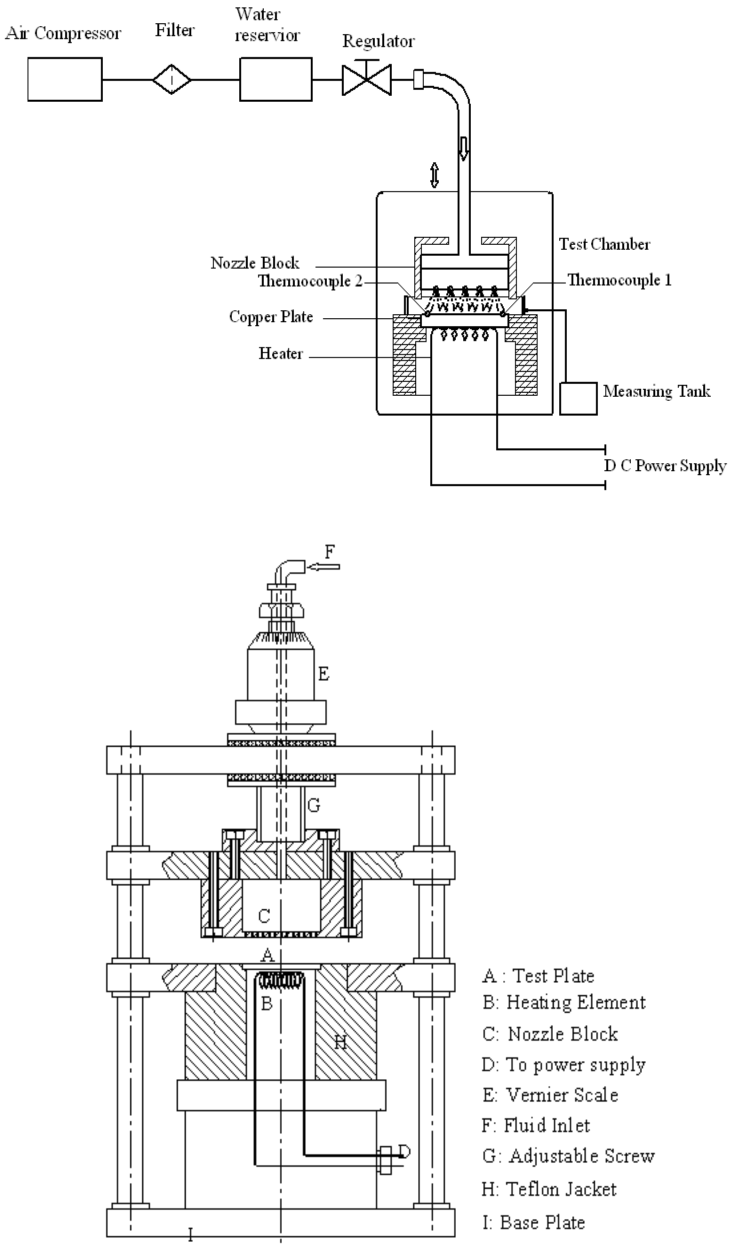


**Figure 1:** Schematic diagram of the experimental set up(a) Schematic arrangement b) Heater block assembly.

It also serves to achieve fine control over the flow rate. Safety valve is provided to prevent excess pressure build-up in the system.

The heater assembly consists of 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 is selected because of its high thermal conductivity. The hot plate is of 20mm x 20mm size and thickness 1mm. 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 is controlled using the variable voltage transformer. Two thermocouples were mounted underneath the hot plate on the centre line and insulated with ceramic insulation. These thermocouples also provide indication of surface temperature uniformity on the plate. The complete heater assembly is mounted and insulated using a Teflon jacket. The leads from the thermocouples are 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 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 the set value.

The Jet nozzle head is made of stainless steel and it consists of the nozzle chamber and nozzle plate. The nozzle chamber is connected to the reservoir through a connecting tube. The nozzle plate is made of 3mm thick stainless steel plate. The nozzle plate is designed to cover the nozzle chamber making it a single leak proof unit. Two nozzle plates having 0.25mm and 0.5mm diameter holes were used. The holes are laser drilled and arranged in a square array of 7X7 with a pitch distance of 3mm between the holes. The distance between the nozzle plate and the hot plate surface is maintained at 10mm and 20mm. 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 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. The values of test parameters used in the present study are given below.

Jet diameter = 0.25mm, 0.5mm
Heat flux range =25 to 200W/cm$^2$
Flow rate range =14 to 40ml/sec
Distance between nozzle head and test plate =10mm, 20mm
Positioning of the nozzle =Horizontal, vertical

## Results and Discussions

Fig.[2] shows the variation of heat transfer co-efficient (h) with heat flux for various flow rates. It is observed that (h) increases significantly with heat flux at all flow rates. The effect of jet diameter on the heat transfer co-efficient can be easily noticed. Results show that smaller diameter jet is more effective in enhancing heat transfer. The increase in flow rate enhances the heat transfer co-efficient and the effect is higher with smaller jet diameter. 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 nozzle.The heat transfer results have been plotted in the non-dimensional form, ln $(Nu/Re^{0.25})$ V/s ln (q) in Figs. [3-6] by varying the parameters independently. The effect of (Z) and the horizontal or vertical positioning of nozzle seems to be less important as compared to the effect of heat flux and the jet diameter. Figs (3) and (4) show the variation of $\ln(Nu/Re^{0.25})$ V/s ln(q) for various Reynolds numbers with the jet diameter of 0.25mm, Z=10 and 20mm and with both horizontal and vertical positioning of the nozzle. Similar trends in the variation of $\ln(Nu/Re^{0.25})$ with ln(q) and the Reynolds number have been observed in all cases. The effect of Reynolds number in the lower range of heat flux can be easily noticed. Fig (5) shows the effect of Z on horizontal and vertical positioning of the nozzle with the jet diameter of 0.25mm and Reynolds number of 1450. It is evident that the heat transfer co-efficient or Nusselt number increases with increase in Z in both cases. Similar trends were observed at the Reynolds numbers of 1760 and 2276 as shown in Figs (6) and (7). Figs (8) and (9) show the results obtained when the nozzle is placed in horizontal and vertical positions respectively. The Reynolds number is around 1760 and jet diameters are 0.25 and 0.5mm. The variations show a similar trend. It is interesting to compare Fig (8) and (9) with Fig(2) which shows the strong effect of jet diameter on the heat transfer co-efficient when plotted in h V/s q co-ordinates. The effect of horizontal and vertical positioning of the nozzle can be easily noticed. Fig (10) shows the composite plot of $\ln(Nu/Re^{0.25})$ V/s ln(q) for d=0.5mm and d=0.25mm respectively with different flow Reynolds numbers, Z=10 and 20mm and with horizontal and vertical positioning of the nozzle. It is interesting to note that the results follow a similar trend and the average of the results obtained with d=0.5 and 0.25mm follow the relationship $Nu=0.17q^{0.8}Re^{0.25}$.

The plots also show the lines indicating the upper and lower bounds. Large variation of the heat transfer data about the mean line (A=0.17 and n=0.8) at lower values of q is due to the combined effects of Reynolds number, Z and the horizontal or vertical positioning of the nozzle It can be easily observed from fig(10) that the combined effect of these parameters is larger in the case of d=0.25mm. As heat flux is increased, the combined effects of these parameters decreases and beyond 400W/cm$^2$, the effect of these parameters may become negligible. Thus at the at lower values of heat flux, other parameters also play an important role in addition to heat flux in determining the heat transfer co-efficient. Figs. 11 (a) and(b) show the comparison of the present heat transfer results with the results of Matteo fabbri and Vijay K Dhir [6 ,15]. The effects of 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.

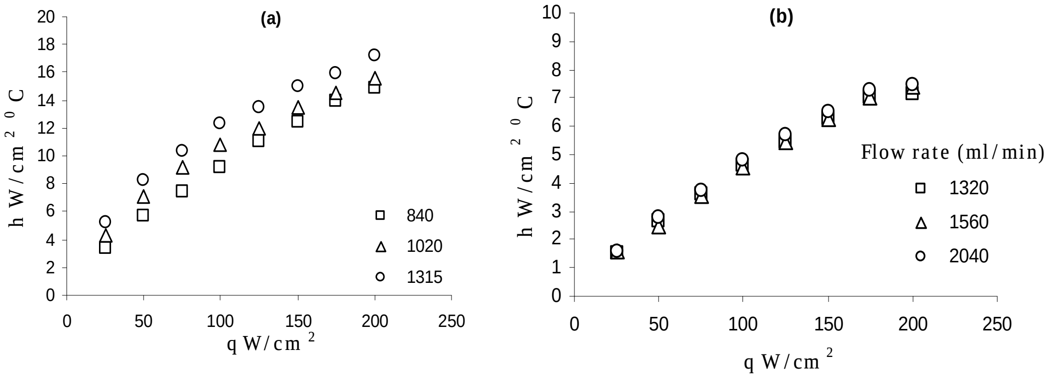


**Figure 2:** Variation of heat transfer co-efficient with heat flux. Z=20mm, Horizontal position (a) d=0.25mm and (b) d=0.5mm.

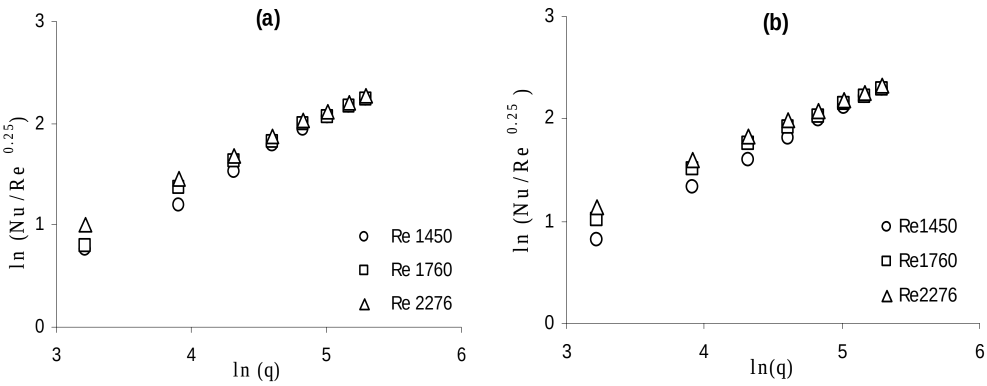


**Figure 3:** Variation of Nu/Re$^{0.25}$ with heat flux for various Reynolds numbers, Horizontal position (a) Z=10mm and d=0.25mm (b) Z=20mm and d=0.25mm.

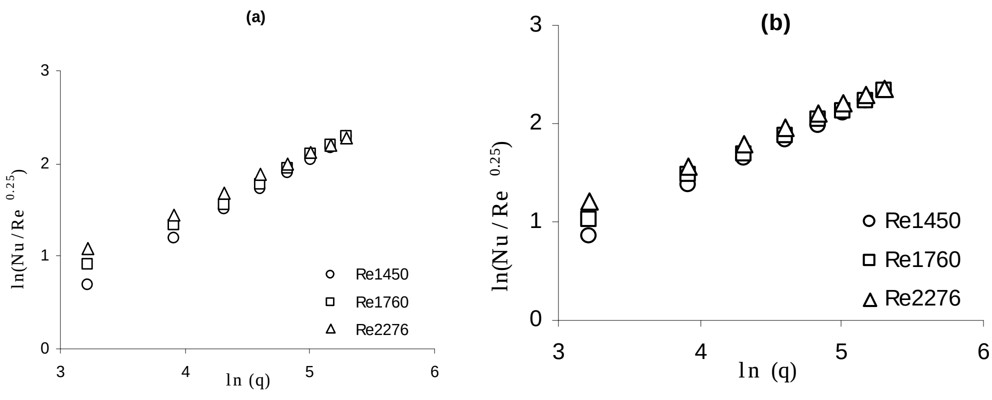


**Figure 4:** Variation of Nu/Re$^{0.25}$ with heat flux for various Reynolds numbers, Vertical position (a) Z=10mm and d=0.25mm (b) Z=20mm and d=0.25mm.

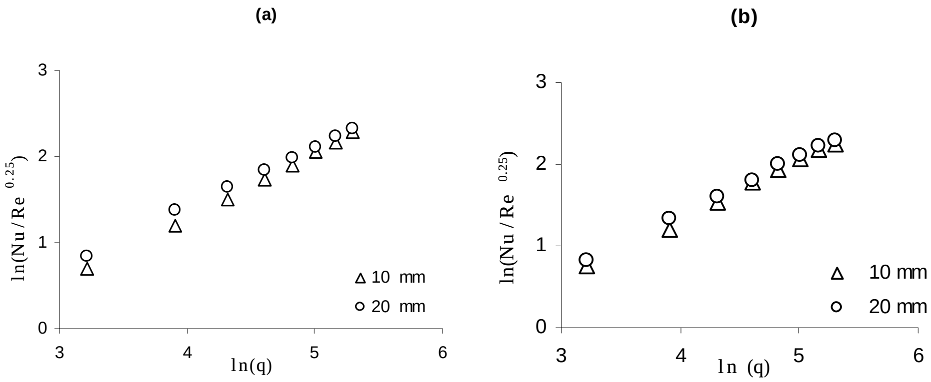


**Figure 5:** Variation of Nu/Re$^{0.25}$ with heat flux at Re= 1450, d=0.25mm, Z=10 and 20mm (a) Vertical position (b) Horizontal position.

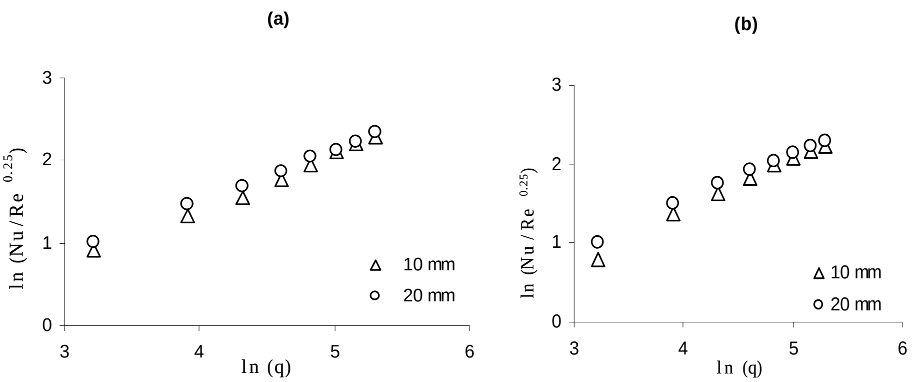


**Figure 6:** Variation of Nu/Re$^{0.25}$ with heat flux at Re=1760, d=0.25mm, Z=10 and 20mm Vertical position (b) Horizontal position.

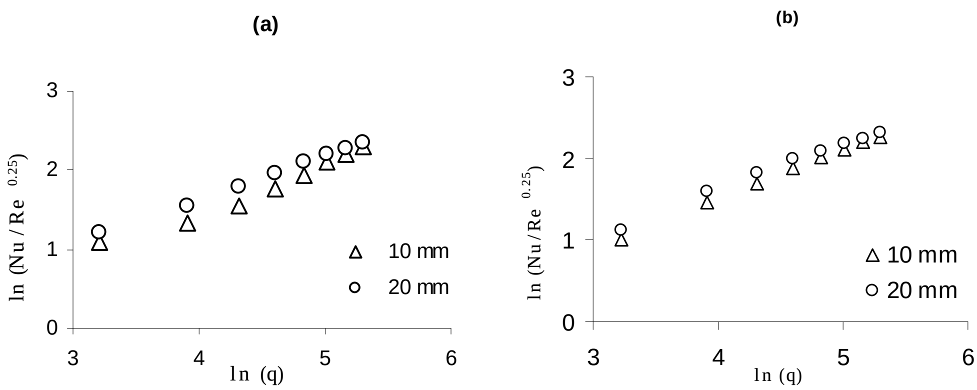


**Figure 7:** Variation of Nu/Re$^{0.25}$ with heat flux at Re=2276, d=0.25mm, Z=10 and 20mm Vertical position (b) Horizontal position.

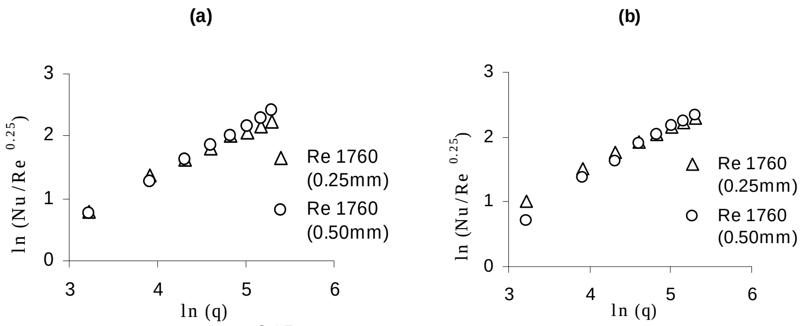


**Figure 8:** Variation of $Nu/Re^{0.25}$ with heat flux at Re=1760, Horizontal position d=0.25 and 0.5mm (a) Z=10mm and (b) Z=20mm.

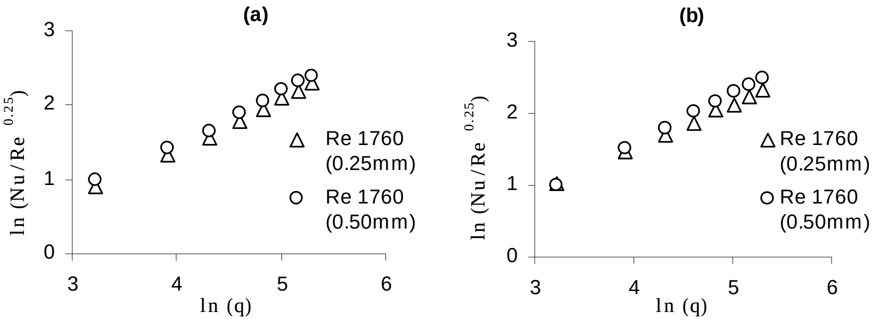


**Figure 9:** Variation of $Nu/Re^{0.25}$ with heat flux at Re=1760, Vertical position d=0.25 and 0.5mm (a) Z=10mm and (b) Z=20mm.

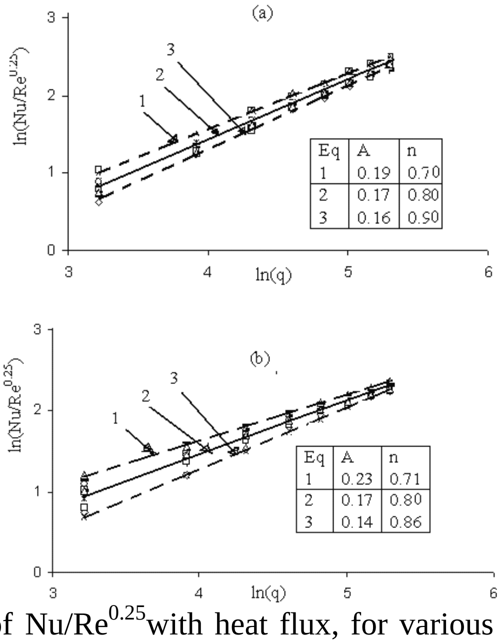


**Figure 10:** Variation of $Nu/Re^{0.25}$with heat flux, for various Reynolds numbers and (Z)Reynolds number range = 1100 to 2280, Z=10 and 20mm (a) d=0.5mm jet diameter (b) d=0.25mm jet diameter (Data for both vertical and horizontal positioning of nozzle included).

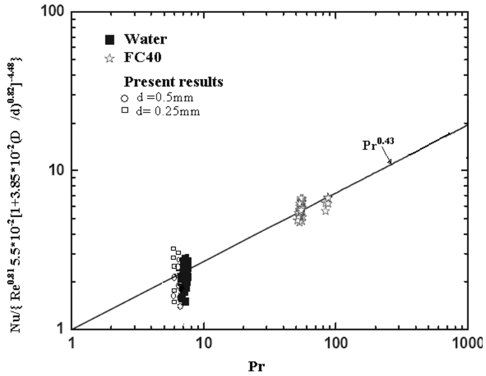


**Figure 11:** (a) Comparison of the present heat transfer results with the data available in the literature [6].

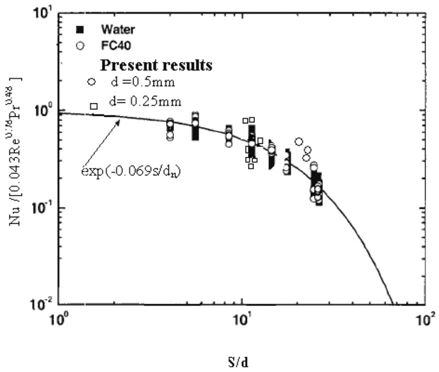


**Figure 11:** (b) Variation of $Nu_{dn}$ /[0.043$Re^{0.78}$ $Pr^{0.48}$ ] with Prandtl number for varying Reynolds numbers and ratio of pitch to jet diameter. Comparison of the present results with the data available in the literature [15].

## 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 200W/cm$^2$, which is typical for high power electronic components, was dissipated using multiple water jets of 0.25mm and 0.5mm diameter. The temperature difference between the test plate and water was within 30$^0$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 nozzle in both horizontal and vertical positions.

It is observed that the heat transfer co-efficient is a strong function of heat flux. Reynolds number plays an important role. The effects of distance between the test plate and the nozzle exit and the horizontal or vertical positioning of the nozzle can be noticed at the lower values of heat flux. The effects of all these parameters decrease when the jet diameter is increased from 0.25mm to 0.5mm.At higher values of heat flux say, about 400W/cm$^2$ ,the effect of these parameters reduce and the heat flux is found to play a dominant role. Overall, the heat transfer data can be correlated using 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 [Fig 11(a) and (b)]. 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. Further investigations on cooling using multiple air jets are in progress.

## Nomenclature

| | |
|---|---|
| A | Hot plate surface area (cm$^2$) |
| d | Nozzle diameter (mm) |
| h | Heat transfer coefficient (W/cm$^2$c) ( $q / (T_c - T_w)$ ) |
| Nu | Nusselt number ( hd/k) |
| P | Total heat transfer (W) |
| q | Heat flux (W/cm$^2$) (P/A ) |
| Q | Total flow rate (ml/min) |
| $R_e$ | Reynolds number (Vd/ν) |
| $T_b$ | Bulk fluid temperature ($^0$C) |
| $T_c$ | Chip surface temperature ($^0$C) |
| $T_w$ | Inlet water temperature ($^0$C) |
| V | Jet velocity (m/s) |
| ν | Kinematic viscosity (Ns/m$^2$) |
| z | Nozzle height from chip surface (mm) |
| ΔT | ($T_c - T_w$) ($^0$ C) |

## References

[1] Elison, B.and Webb, B.W. “Local heat transfer to impinging liquid jets in the initially laminar, transitional, and turbulent regimes.” International Journal of Heat and Mass transfer, 37, no 8, (1994) pp.1207-1216.

[2] Matteo Fabbri, Shanjuan Jiang, Vijay K. Dhir; Comparative Study of Spray and Multiple Micro Jets Cooling for High Power Density Electronic Applications; ASME Intenational Mechanical Engineering Congress and Exposition Washingtn, D.C., (Nov 2003).

[3] Jiji.L.J and Dagan.Z; Experimental Investigation of Single-Phase Multi Jet Impingement Cooling of an Array of Micro Electronic Heat Sources; Proceedings of the International Symposium on Cooling Technology for Electronic Equipment, W. Aung, ed., Hemisphere Publishing Corporation, Washington.D.C., pp. 333–351.

[4] Womac.D. J, Incropera.F. P and Ramadhyani.S; Correlating Equations for Impingement Cooling of Small Heat Sources With Multiple Circular Liquid Jets; ASME J. Heat Transfer, 116, (1994) pp. 482–486.

[5] Wang.E.N, Zhang.L, Jiang.L, Koo.J.M, Goodson.K. E, Kenny.T.W, Maveety.J. G, Sanchez.E.A; Micro-machined Jet Arrays for Liquid Impingement Cooling of VLSI Chips, Solid-State Sensor, Actuator and Microsystems Workshop; Hilton Head Island, South Carolina, June 2-6, 2002.

[6] Matteo Fabbri, Shanjuan Jiang, Vijay K. Dhir; "Experimental investigation of single – phase micro jets impingement cooling for electronic applications" ASME Summer Heat Transfer Conference July 21-23, Las Vegas, Nevada, USA 2003.

[7] Yang Cheng, Andrew A.O. Tay, Xue Hong " An experimental study of liquid jet impingement cooling of electronic components with and without boiling" International symposium on electronic materials and packaging (2001).

[8] Lin Z.H., Chou Y.J., and Hung, Y.H.. Heat transfer behaviors of a confined slot jet impingement, International Journal of Heat and M ass Transfer, Vol.40, (1997) pp. 1095-1107.

[9] A.G.Pautsch, T.A.Shedd - University of Wisconsin – Madison, USA; Spray Impingement Cooling with Single and Multiple Nozzle Arrays (2005).

[10] Shaw.M.C, Waldrop.J.R, Chandrasekaran.S, Kagalwala.B, Jing.X, Brown.E.R, Dhir.V. K and Fabbri.M; Enhanced Thermal Management by Direct Water Spray of High-Voltage, High Power Devices in a Three Phase, 18-hp AC Motor Drive Demonstration; Proceedings of the 8$^{th}$ Intersociety Conference on Thermal and Thermo-mechanical phenomena in Electronic Systems, San Diego, CA, (2002) pp. 1007–1014.

[11] Fundamentals of liquid cooling; A presentation in Thermal Management of Electronics, San José State University, Mechanical Engineering Department.

[12] 3M specialty fluids ;http://3m.com/market/industrial/fluid/library (2006).

[13] P.Ing, C.Sperry, R.Philstrom, P.Claybaker, J.Webster and R.Cree; Supercomputer Cooling System; in Proc 43$^{rd}$ electronic Components Technology Conf, 1993, pp.218 -237.

[14] Avram Bar – Cohen, Mehmet Arik and Michael Ohadi; Direct Liquid Cooling of High Flux Micro and Nano Electronic Components; Proceedings of IEEE, vol.94, No.8, Aug 2006.

[15] Matteo Fabbri, Vijay K. Dhir; "Optimized heat transfer for high power electronic cooling using arrays of microjets" Transactions of the ASME Vol. 127, July 2005.