Keynote Paper

A REVIEW OF THERMAL MANAGEMENT TECHNIQUES AND SOME RECENT DEVELOPMENTS ON JET IMPINGEMENT COOLING

Muhammad Mustafizur Rahman*
Department of Mechanical Engineering
University of South Florida
4202 E. Fowler Avenue, ENB 118
Tampa, Florida 33620, U.S.A.

Rengasamy Ponnappan
Energy Storage and Thermal Sciences Branch, Power Division
Air Force Research Laboratory, AFRL/PRPS
1950 Fifth Street, Bldg. 18
Wright Patterson AFB, Ohio 45433, U.S.A.

Abstract This paper presents a review of commonly used active thermal management techniques and discusses their advantages and limitations. These techniques are being used for the removal of high heat flux over a relatively small area that is typical in electronic equipment. An adequate thermal management scheme is needed to assure efficient and reliable operation of power electronics. Even though different cooling options can be tailored to the specific needs of an individual application, system considerations always play a paramount role in determining the most suitable technique. The review presented here covers a wide variety of applications including consumer electronics, computers and telecommunication equipment, automotives, aircraft, and spacecraft. In addition to the overall review of the field and identification of future challenges, some recent modeling and simulation efforts in the area of liquid jet impingement are presented. Even though experimental testing gives a first hand evaluation of the performance of any system, advanced modeling and simulation studies are frequently being used to predict the variations of different design parameters and to arrive at the optimum design of a system. The modeling of most engineering systems require the solution of conjugate heat transfer by including heat generation at discrete locations, conduction through the solid materials, and convection to the fluid at the heat sink. In addition to steady state operation, a transient or cyclic performance evaluation may be needed to predict the effectiveness of a design during start-up, shut down, or load variations. The variation of solid-fluid interface temperature, average Nusselt number, and the maximum temperature encountered by the device are explored.

Keywords: Thermal Management, Jet, Cooling

INTRODUCTION

An adequate scheme for thermal management is essential for proper operation of many engineering equipment. Cooling, heating, or thermal isolation is needed to maintain the required thermal environment. For example, the performance and reliability of electronic circuits are strongly influenced by its operating temperature. Exposure to temperatures beyond which the circuit is designed to withstand may result in severe performance degradation or failure. Therefore, it is needed to preserve the required operating temperature. Similarly, in a spacecraft design, the thermal management plays a very crucial role where solar thermal radiation is used to produce power, insulation and louvers are used to protect the spacecraft from adverse thermal environment, and waste heat produced in the spacecraft need to be rejected to space via radiator or expendable coolant. The objective of the present investigation is to perform a review of high heat flux thermal management schemes and to identify areas that need further technical development. Even though air cooling is the solution for most low heat flux applications and has been very widely used in engineering equipment, the maximum heat transfer coefficient that can be attained is only of the order of 0.02 W/cm^2 ^{o}C. Therefore, it cannot be used to cool high heat flux surfaces (heat flux > 10 W/cm^2) where one must resort to liquid or multi-phase cooling. In addition, the need for a compact and lightweight system may require an arrangement that can provide a high heat transfer coefficient at the solid-fluid interface in

*Email: rahman@eng.usf.edu

order to reduce the overall thermal resistance. Research in the area of high heat flux thermal management has resulted in a number of quite effective schemes. These include: single phase liquid pumped loop with a microchannel or very compact heat exchanger, single phase liquid jet impingement, flow boiling, jet impingement with evaporation or boiling, spray cooling, and heat pump or refrigeration.

PUMPED LOOP WITH MICROCHANNEL HEAT EXCHANGER

One of the most promising techniques for thermal management of high performance computers is direct forced convection liquid cooling. A high rate of power dissipation can be achieved by forcing the cooling fluid through microchannels fabricated directly in the wafer containing heat dissipating integrated circuit components. This cooling configuration results in much lower thermal resistance compared to conventional cooling methods because of shorter conduction path between heat sources and the sink and larger heat transfer coefficient at the solid-fluid interface.

Tuckerman and Pease [1981] designed and tested a compact water cooled heat sink that was an integral part of the silicon substrate. The channels were fabricated in the silicon substrate by precision sawing or orientation-dependent etching. The fourth side of the channel was formed by attaching a cover plate to the heat sink by anodic bonding or by using a suitable epoxy. The coolant flowed from the inlet plenum, through the channels, and out into the discharge plenum. Subsequent publications by Tuckerman and Pease [1982, 1983], Rittmann and Tuckerman [1983], and Bernhardt et al. [1990] outlined the same type of design methodology and rationale for laminar microchannel heat exchangers as applied to the packaging and cooling of high-powered electronic components.

Phillips [1987] came up with a method of determining the overall thermal resistance of such a heat exchanger as a function of pertinent variables. He performed a sensitivity analysis to evaluate various parametric effects on his test case and extended the analysis to include larger channel widths with moderate aspect ratios and for fully developed and developing flow in laminar and turbulent flow regimes. The test sections were fabricated using indium phosphide as the wafer material and water was used as the working fluid.

Nayak et al. [1987] worked on microchannel configurations in copper substrates. They came up with two designs: one for laminar flow and the other for turbulent flow. These were high aspect ratios channels, with the longer channel dimension aligned to the plane of the heater. Mundinger et al. [1988] fabricated and tested a microchannel heat exchanger package for the cooling of a edge-emitting laser diode array in a rack and stack architecture using water as the coolant. The results showed a thermal resistance of 0.04 (^{o}C cm^{2})/W for a single operating laser bar.

Walpole et al. [1988] presented the use of microchannel heat sinks as an effective heat extraction method for surface emitting two-dimensional laser arrays. Missaggia and Walpole [1991] incorporated a clever alteration of channel configuration as a means for improving the uniformity of temperature across the cooled device. This was done by altering the direction of coolant flow in adjacent channels. Knight et al. [1991] described an optimal geometrical design for fully developed flow in a closed finned channel in terms of relevant geometrical and flow parameters.

Weisberg et al. [1992] conducted a theoretical study for the thermal resistance of cooling channels integrated into silicon chips. A design procedure for the selection of channel dimensions in conformity with operational constraints was found for a flat plate heat exchanger consisting of rectangular channels fabricated in a silicon wafer and capped with a Pyrex plate. Beach et al. [1992] presented detailed performance results for an efficient and low thermal impedance laser diode array heat sink. The microchannels were fabricated in silicon using an anisotropic chemical etching process. A modular rack and stack architecture was adopted for the heat sink design.

Rahman and Gui [1993] fabricated and tested microchannel heat sinks using silicon <100> wafers. The reactive ion etching was used to develop trapezoidal channels of different depths keeping its outer surface width as 1 mm. A Pyrex cover plate was bonded to the wafer to cover the channels and to provide intake and exit manifolds for fluid flow. Resistance heaters were fabricated on the wafer to simulate electronics thermal load. Using water as the working fluid, tests were carried out to study the mechanical and thermal characteristics of the heat sink for different thermal loading and flow rate.

Arthur et al. [1992] used forced water cooling with 40 μm wide and 400 μm deep microchannels for the thermal management of x-ray monochromator crystals. The crystals were tested at a high-power wiggler beam line, using an x-ray beam having total power in excess of 250 W and normal incidence power density greater than 5 W/mm^{2}. The cooling minimized the amount of type-2 strain (thermal bump) produced by the hot x-ray beam. Arthur [1995] presented results of an experimental evaluation of microchannel and pin-post cooling techniques using water as the coolant for the silicon crystals for use with x-ray monochromators at high-power synchrotron radiation beamlines. Long parallel microchannels with approximately 50 μm in width were used. The pin-post geometry created a two-dimensional network of sub-millimeter water passages.

Both methods were found to be satisfactory for power levels of less than 5 W/mm^2.

Bowers and Mudawar [1994] studied flow boiling in mini- and micro-channel heat sinks with tube diameters of 2.54 mm and 0.51 mm respectively. Heat was applied over an area of 1cm x 1cm. Tests were performed using R-113 as the working fluid for a range of inlet subcoolings (10-35 ^oC) and flow rates (19-95 ml/min). Tests yielded critical heat flux in excess of 200 W/cm^2 in both heat sinks with a pressure drop of less than 0.35 bar. The two-phase pressure drop was found to be significantly lower than single-phase flow at comparable thermal loading. In addition, mini-channels had much lower pressure drop compared to micro-channels.

Wang and Peng [1994] reported experimental data for convection of water and methanol in rectangular microchannels. Channels with hydraulic diameters between 311 and 747 μm were tested. It was found that heat transfer characteristics can not be described well with standard liquid forced convection correlations for rectangular ducts. In a later study, Peng and Peterson [1996] reported experimental investigation with microchannels having hydraulic diameters between 133-367 μm. It was found that channel aspect ratio has very strong influence on convection heat transfer coefficient. The transition from laminar to turbulent flow occurred at smaller Reynolds number compared to ordinary channel flows.

Roy and Avanic [1996] developed a low-cost, high heat flux heat exchanger for the cooling of high power semiconductor laser diode arrays. The heat exchanger was made out of standard copper tubing and water was used as the working fluid. Two prototypes with dissipation rate of 1000 W over an area of 1 cm^2 and 200-300 W over an area of 0.5 mm^2 were tested. The results showed that the overall thermal resistance was 0.03 C/W, which was less than conventional heat sinks. Goodson et al. [1997] proposed a novel cooling system for high-power laser diode arrays using a microchannel heat sink made of chemical-vapor-deposited diamond. The high thermal conductivity increased the efficiency of the channel-wall fins and reduced the array-to-coolant thermal resistance.

Harms [1997] and Harms et al. [1997] presented experimental data for single phase forced convection in deep rectangular microchannels. The channels were fabricated in a 2 mm thick silicon substrate by means of chemical etching and covered a total projected area of 2.5 cm by 2.5 cm. A thin-film heater was deposited on the back side of the silicon substrate, corresponding to the entire projected channel area. The silicon substrate measured 2.9 cm by 2.9 cm. All tests were performed with deionized water as the working fluid, where the liquid flow rate ranged from 5.47 cc/s to 118 cc/s. A critical Reynolds number of 1500 for laminar-turbulent transition was found for this configuration.

Rujano and Rahman [1997] presented a numerical simulation model for transient heat transfer during start-up of power in a microchannel heat sink containing trapezoidal channels. The distribution of local heat transfer coefficient along the periphery of the channel as well as its variation along the direction of the flow were explored with the advancement of time. Calculations were done for different values of Reynolds number, channel depth, spacing between channels, and wafer thickness. In addition, the response of the heat sink under pulsed power was investigated.

Vidmar and Barker [1998] presented design calculations for the application of microchannel cooling technique with turbulent liquid flow for applications in high-energy particle transmission window, RF transmission window, and VLSI. Calculations were done for water and heptane and a simple prototype test was done to verify certain design aspects. A heat dissipation rate of 2700 W/cm^2 for electron beam window, 722 W/cm^2 for RF window, and 950 W/cm^2 for VLSI was predicted.

Kim and Kim [1999] modeled the microchannel heat sink as a fluid-saturated porous medium, and presented analytical and numerical results for velocity and temperature distribution. An expression for the total thermal resistance of the device was developed. Fedorov and Viskanta [2000] presented a three-dimensional numerical solution for conjugate heat transfer in a rectangular channel. For laminar fluid flow within the channel and uniform heating at one of the side walls, the heat flux distribution at the solid-fluid interface was presented. The predicted thermal resistance and friction coefficient were compared with available experimental data for the same geometrical arrangement.

Rahman [2000] presented a detail experimental study of heat transfer in trapezoidal microchannels. Two channel patterns: parallel and series were tested using water as the working fluid. From the measured temperature distribution, local heat transfer coefficient at the solid-fluid interface was evaluated at two locations along the flow length. In addition, the average heat transfer coefficient was determined. It was observed that the parallel pattern provided better overall heat transfer characteristics with smaller pressure drop.

Ozmat et al. [2000] investigated an advanced power module technology by combining Chip on Flex Power Overlay (POL) with double sided integral compact heat exchangers. Their analysis showed a heat removal rate of 2500 W/in^2 with junction to ambient temperature difference of 100 ^oC. The targeted applications for this were Integrated Gate Bipolar Transistor (IGBT), Metal Oxide Semiconductor Field Effect Transistor

(MOSFET), and MOS controlled Thyristor (MCT) that are pushing the current densities, switching speeds, and frequencies to new dimensions. The superior performance of power device was due to elimination of bond wires and the planar geometry that offer cooling from top, bottom, or both sides.

Tso and Mahulikar [2000] presented a theoretical analysis of combined meniscus driven convection and radiation in an annular space by discretizing the domain into annular elements and making mass, momentum, and energy balance at each element taking into account the surface radiation. The benefits of high heat transfer coefficient associated with liquid cooling and phase change was demonstrated.

Ambatipudi and Rahman [2000] developed a numerical simulation model for fluid flow and heat transfer processes in a silicon wafer containing integrated circuits (heat source) and microchannels. The wafer was modeled exactly by taking into account heat generation in circuit components, conduction of heat through the solid, and convection of heat to the coolant. Rectangular microchannels fabricated using silicon <110> as the wafer material was considered in this study. The numerical results were validated by comparing with the experimental data [Harms, 1997; Harms et al., 1997]. A detailed parametric study was carried out to explore the effects of channel depth, channel width, and Reynolds number.

Jiang et al. [2001] presented the design and open loop test results of a two-phase microchannel heat sink for possible application in a electrokinetic VLSI chip cooling system. Their design used 40 channels with 350 μm pitch and each channel had 100 μm width, 100 μm depth, and 15 mm length. Water was used as the test fluid. Both single and two-phase tests were carried out. A periodic temperature oscillation was observed when two-phase flow developed in the channels. Also, starting at the onset of two-phase flow, the pressure drop increased rapidly with increasing input power due to the acceleration of the fluid. Periodic pressure oscillation was also observed in the two-phase condition.

In summary, the following highlights the current state of the microchannel heat exchanger technology.

1. The fabrication technique for microchannels in a silicon wafer and the bonding of a cover plate, which appeared to be a significant challenge in earlier studies has now been perfected.

2. Using single phase flow, a microchannel heat exchanger is capable of removing a large heat flux, particularly if that appears over a small area. The values reported are on the order of 500 W/cm^2 [Arthur et a1., 1992] to 1000 W/cm^2 [Roy and Avanic, 1996; Vidmar and Barker, 1998].

3. The primary design constraint appears to be large pressure drop that happens in channels with small hydraulic diameter. One possible solution in this regard is to design for a larger hydraulic diameter and enhance heat transfer by using other surface enhancement techniques.
4. Evaporation within a microchannel and associated two-phase flow is still a major technical challenge, which need to be resolved in near future. The temperature oscillation [Rahman and Gui, 1993; Jiang et al., 2001] appear to make is unusable for electronic cooling. One obvious solution is to use larger channel as proposed by Bowers and Mudawar [1994]. However, this study reported a maximum heat flux of 200 W/cm^2 which is lower than that obtained by single phase flow.

5. The two-phase flow may also result in lower pressure drop compared to single phase flow [Bowers and Mudawar, 1994] with same heat removal rate.

LIQUID JET IMPINGEMENT

Jet impingement heat transfer is known for its ease of implementation and high heat transfer coefficients. Free-surface jets are formed when a liquid issues from a nozzle or orifice into a gas environment. The free surface forms immediately at the nozzle exit and prevails through the impingement region and into the wall jet region. The shape of the free surface depends on gravitational, surface tension, and pressure forces. The jet speed, size, and orientation determine the magnitude of these forces. In the past, several studies have been carried out on heat transfer during free jet impingement.

Glauert [1956] considered the flow due to a jet spreading out over a plane surface, either radially or in two dimensions. Solutions to the boundary layer equations were sought, according to which the form of the velocity distribution across the jet did not vary along its length. A solution for the laminar flow was obtained explicitly using a similarity transformation and, for turbulent flow, eddy diffusivity was introduced but the final result showed that the complete similarity solution was not attainable. Olsson and Turkdogan [1966] studied the hydrodynamics of a jet impinging over a flat plate, in the region before the hydraulic jump. The experiment was carried out to determine the thickness of the film, the boundary layer, and the diameter of the potential core.

Metzger et al. [1974] presented results for heat transfer characteristics of circular liquid jets impinging normally on plane surfaces. The experimental procedure was developed to cover single axisymmetric liquid jets formed by water or synthetic-based lubricating oil, and included a significant Prandtl number range so that extension to other liquids would be indicated. Nakoryakov et al. [1978] presented a study, theoretically and experimentally, of hydrodynamics and mass transfer from an axisymmetric liquid jet impinging onto a horizontal plate. Simple formulae were obtained for the calculation of friction factor, liquid layer thickness, liquid surface velocity, and mass transfer coefficient as a function of discharge parameters. Hrycak [1983] carried out an investigation of heat transfer from round jets impinging normally over a flat plate. This study was done for various nozzle-to-target distances, with Reynolds number varying from 14000 to 67000 (turbulent impinging flow), and different nozzle diameters.

Garg and Jayaraj [1988] theoretically analyzed the laminar boundary layer flow when a two dimensional slot jet impinges over a flat plate at some angle. The analysis was performed using a finite-difference technique, and the results were presented for impinging angles of 0° and 90°. The presence of a stagnation point when the plate is not parallel to the flow was found to considerably affect the local Nusselt number and the skin friction coefficient.

Liu and Lienhard [1989] investigated a circular subcooled liquid jet impinging on a surface maintained at a uniform heat flux. They used an integral method to obtain analytical predictions of temperature distribution in the liquid film and the local Nusselt number. They also carried out experiments to test the predictions of the theory.

Wang et al. [1989a, 1989b] presented an analytical study of heat transfer between an axisymmetrical free impinging jet and a solid flat surface with a non-uniform wall temperature or wall heat flux. The results obtained showed that the non-uniformity of wall temperature or wall heat flux has a considerable effect on the stagnation point Nusselt number. For the boundary layer region, a superposition solution was obtained beginning with the solution of a problem with a step change in the wall temperature or wall heat flux. The results indicated that the Nusselt number for increasing wall temperature or wall heat flux can be considerably higher than that for constant wall temperature or wall heat flux outside the stagnation region. Wang et al. [1989c] investigated the conjugate heat transfer between a laminar free impinging liquid jet and a laterally insulated disk with arbitrary temperature or heat flux distribution prescribed at the non-impingement surface. Their analytical solution showed that the heat transfer coefficient is influenced by the Prandtl number of the fluid, the ratio of the fluid to solid thermal conductivity, the ratio of the thickness to the radius of the disk, and the prescribed temperature or heat flux profile. Wang et al. [1990] applied the previously developed analytical solution to the conjugate heat transfer problem of a laminar jet impingement cooling of a microelectronic chip. They presented results for two different nozzle diameters.

Wadsworth and Mudawar [1990] presented an experimental investigation of cooling a 3x3 array of heat sources simulating chips by a similar array of two-dimensional jets of dielectric Flourinert FC-72 issuing from a thin rectangular slot into a channel confined between the chip surface and the nozzle plate. The heat transfer was not very significantly affected by nozzle shape or channel height. The average Nusselt number was strongly dependent upon jet velocity and jet width.

Schaffer et al. [1991] presented the results of an experimental study measuring the average heat transfer coefficient for discrete sources located under a liquid jet issuing from a rectangular slot. The experiment was conducted for heat sources mounted on a channel (submerged jet). Besserman et al. [1991] presented a numerical simulation of an axisymmetric, laminar jet impingement cooling of a circular heat source. The simulation includes the effect of the discharge fluid when redirected 180° to an annular exit. Their results demonstrated that the flow is strongly influenced by two recirculation zones near the exit and at the corner of the outside annulus. For nozzle parabolic velocity profile, the local Nusselt number presented a maximum at the stagnation point.

Wolf et al. [1990] performed experiments on a planar, free surface jet of water to investigate the effects of a nonuniform velocity profile on the local convective heat transfer coefficient for a uniform heat flux surface. The heat transfer coefficient was measured for different heat fluxes and Reynolds numbers. Vader et al. [1991] measured temperature and heat flux distribution on a flat, upward facing, constant heat flux surface cooled by a planar, impinging water jet. The velocity at the exit of the nozzle, the temperature of the fluid, and heat flux were varied. They found that the stagnation convection coefficient exceeded those predicted by laminar flow analysis and this was caused by the existence of free stream turbulence.

Polat et al. [1991a] measured local and average heat transfer coefficient for a confined turbulent slot jet impinging on a permeable surface with thorough flow. Measurements were carried out for a wide range of jet Reynolds number and thorough flow velocity. Polat et al. [1991b] measured local and average heat transfer coefficient for a confined turbulent slot jet impinging on a moving surface considering through flow.

Stevens and Webb [1991] carried out an experimental investigation to characterize local heat transfer coefficient for round, free liquid jet impinging normally over a flat uniform heat flux surface. The parameters varied were Reynolds number, nozzle to plate distance, and jet diameter. Liu et al. [1991] presented an analytical and experimental investigation for jet impingement cooling of uniformly heated surfaces to determine local Nusselt numbers from the stagnation point to radii up to 40 diameters. Turbulent transition in the film flow was observed experimentally at certain radius; beyond this point, a separate turbulent analysis was constructed. Pan et al. [1992] presented a study on the flow structure and heat transfer characteristics of turbulent, free surface liquid jets.

Al-Sanea [1991] developed a finite-difference numerical model to calculate the steady state fluid flow and heat transfer characteristics for a laminar slot jet impinging on an isothermal flat surface. The study was performed for free jet, semiconfined jet, and semiconfined jet in cross flow. The study showed that cross flow can degrade the average Nusselt number by as much as 60%. Chou and Hung [1994a] presented an analytical study for cooling of an isothermal heated surface with a confined slot jet. Chou and Hung [1994b] performed a numerical study for fluid flow and heat transfer of slot jet impingement with an extended nozzle. The parametric study included jet Reynolds number, nozzle-to-surface distance, and nozzle length.

Elison and Webb [1994] studied transport from a small diameter, fully developed liquid jet impinging normally over a constant heat flux surface. The study focussed on varying Reynolds number spanning the laminar, transitional, and turbulent flow regimes at the nozzle exit. Both free-surface and submerged jets were studied and correlations for Nusselt number as a function of the Reynolds number were developed.

Womac et al. [1993] obtained experimental data for liquid jet impingement cooling of small square heat sources resembling integrated circuit chips. Both free surface and submerged jet configuration were studied for a range of velocities, nozzle diameters, and nozzle-to-heater distance. Two different liquids, water and FC-77, were used as coolants. Womac et al. [1994] carried out an experiment to investigate single-phase heat transfer from a heat source to an array of free surface and submerged jets. They found that for a constant volumetric flow rate, the heat transfer for submerged jets exceeded or were approximately equal to those for the free surface jets. The average heat transfer coefficient increased with reduction in nozzle diameter.

Alkam and Butler [1994] applied an explicit finite difference technique to solve the case of transient, forced convective conjugate heat transfer between an axisymmetric incompressible laminar impinging jet and a solid disk at the stagnation zone. Constant properties were considered and no viscous dissipation was taken into account. Rice and Garimella [1994] reported an experimental study to determine the local heat transfer coefficient distribution for a submerged liquid jet impinging perpendicularly on a small, square heat source simulating an electronic cooling situation. They concluded that the local heat transfer coefficient at the stagnation point was independent of the nozzle-to-plate distance for small nozzle diameters, and for distances over 5 nozzle diameters the stagnation heat transfer coefficient decreased. Secondary peaks were observed as the fluid moved away from the stagnation point. Maddox and Bar-Cohen [1994] carried out a study to help design a jet impingement cooling system in attaining the targeted thermal performance while considering the available liquid pressure, liquid flow rate, and pumping power, as well as jet plate manufacturing and maintenance constraints. Garimella and Rice [1995] experimentally investigated the local heat transfer from a small heat source to a normally impinging axisymmetric and submerged liquid jet, in confined and unconfined configurations. Secondary peaks were more pronounced at smaller (confined) spacings and large nozzle diameters for a given Reynolds number. Correlations were presented for the average heat transfer coefficient and the Nusselt number.

Garimella and Nenaydykh [1996] reported an experimental investigation to explore the effects of nozzle diameter and its aspect ratio for submerged and confined liquid jet impingement. Tests were carried out for different values of nozzle to heat source spacing (1-14 jet diameters) and jet Reynolds number (4000-23000). Their results showed significant influence of nozzle diameter and its aspect ratio on the local heat transfer coefficient.

Seyedein et al. [1994] presented results of a numerical simulation of two dimensional flow field and heat transfer due to a turbulent slot jet discharging normally into a confined channel. Low and high Reynolds number versions of $k-\varepsilon$ turbulence models were used to model the turbulent jet flow. Seyedein et al. [1995] presented results of numerical simulation for turbulent flow field and heat transfer due to three and five turbulent slot jets discharging normally into a confined channel.

Laschefski et al. [1996] numerically analyzed the velocity field and heat transfer in rows of rectangular impinging jets in transient state. Axial and radial jets coming out of rectangular nozzles were considered. Cziesla et al. [1997] simulated turbulent flow issuing from a slot jet array using a subgrid stress model. The code showed good agreement with experimental data. Ashfort-Frost et al. [1997] experimentally investigated the velocity and turbulence characteristics of a semiconfined slot jet impinging over a plate. The measurements showed that the potential core of the jet is longer for a confined jet than for an unconfined one. Lin et al. [1997] presented an experimental study on heat transfer behavior of a confined slot jet using Reynolds number and nozzle-to-plate distance as parameters.

Ma et al. [1997a] performed an experimental study to investigate the local convective heat transfer from a vertical heated surface to an obliquely impinging circular free surface jet of transformer oil. The effect of jet inclination was examined in a range of Reynolds number between 235 and 1745, and inclination angles between 45 and 90 degrees. Ma et al. [1997b] performed an experimental study to characterize recovery factor and heat transfer coefficient on vertical heaters impinged by submerged circular transformer oil jets issued from both pipe and orifice nozzles. Local Nusselt number at the stagnation point was found to be proportional nearly to square root of jet Reynolds number in the range 220-1500 and essentially not affected by the plate-to-nozzle distance for Reynolds number less than 600. Ma et al. [1997c] carried out measurements to determine recovery factor and heat transfer coefficients resulting from the impingement of transformer oil jets. This study focussed on initially laminar jets in the range of jet Reynolds number between 55-415, and Prandtl number between 200-270. This study was performed for confined slot jets. Lee et al. [1997] conducted a numerical study to characterize the thermal behavior of laminar circular liquid jets. The effects of different parameters investigated included Reynolds number, Prandtl number, nozzle-to-plate distance, jet velocity, nozzle diameter, and velocity profile at the nozzle exit.

Ma et al. [1997d] performed experimental measurements to investigate the local behavior of the recovery factor and heat transfer coefficient when a free surface oil jet impinges on a vertical uniformly heated surface. The results showed that the recovery factor was nearly independent of both Reynolds number and nozzle-to-plate distance. The heat transfer coefficient at the stagnation point was found independent of the nozzle-to-plate distance but proportional to the square root of Reynolds number. The recovery factor dependence on Prandtl number was expressed in different correlations.

Leland and Pais [1999] performed an experimental investigation to determine the heat transfer rates for an impinging free surface axisymmetric jet of lubricating oil for a wide range of Prandtl numbers, and for conditions varying inside the fluid film. They concluded that the heat transfer surface configuration has an important effect on the Nusselt number. For constant flow rate or Reynolds number, larger nozzle diameters were shown to give higher heat transfer rates. Lee and Vafai [1999] presented a comparison of jet impingement and microchannel cooling techniques by performing analyses and optimization for both systems. It was found that microchannel cooling is preferable for a target dimension smaller than 7 cm x 7cm, whereas jet impingement is comparable or better for a larger target plate if proper treatment is applied for the spent flow after the impingement.

Rahman et al. [1999] presented the results of numerical simulation of a free jet of high Prandtl number fluid impinging perpendicularly on a solid substrate of finite thickness containing electronics on the opposite surface. The numerical model was developed considering both solid and fluid regions and solved as a conjugate problem. The influence of different operating parameters such as jet velocity, heat flux, plate thickness, nozzle height, and plate material were investigated. Computed results were validated with available experimental data. It was found that the local Nusselt number is maximum at the center of the disk and decreases gradually with radius as the flow moves downstream. The average Nusselt number and the maximum temperature occurring in the solid varied significantly with impingement velocity, disk thickness, and thermal conductivity of the disk material.

Bula et al. [2000a] investigated the free jet of high Prandtl number fluid impinging perpendicularly on a solid substrate of finite thickness containing small discrete heat sources on the opposite surface. Both solid and fluid regions were modeled and solved as a conjugate problem. Computations were carried out to investigate the influence of different operating parameters such as jet velocity, heat flux, plate thickness, and plate material. Numerical results were validated with available experimental data. The thickness of the disk as well as the location of discrete sources showed strong influence on the maximum temperature and the average heat transfer coefficient.

Bula et al. [2000b] studied conjugate heat transfer from discrete heat sources to a two-dimensional jet of a high Prandtl number fluid discharging from a two-dimensional slot nozzle. The variation of solid and fluid properties with temperature was taken into account in the numerical simulation. The geometry of the free surface was determined iteratively. It was found that in addition to jet Reynolds number, plate thickness and its thermal conductivity have significant influence on temperature distribution and average Nusselt number.

Rahman et al. [2000] investigated transient conjugate heat transfer during the impingement of a free jet of high Prandtl number fluid on a solid disk of finite thickness. When power was turned on at time = 0, a uniform heat flux was imposed on the disk surface. Computed results included the velocity, temperature, and pressure distributions in the fluid, and the local and average heat transfer coefficients at the solid-fluid interface as a function of time.

Hassaneen and Rahman [2000] presented results of CFD computation of the free surface flow characteristics in a radial impinging free liquid jet. The jet impinged on a flat circular disk and the flow downstream of the impinging area spreaded outward and inward on the disk. Different incidence angle of the jet and different flow Reynolds number were considered in the analysis. The effect of jet elevation from the disk was also discussed. The jet incidence angle and jet elevation were found to have strong effects on the velocity field and the free surface position of the spreading flow on the disk. The Reynolds number was also found to have a strong effect on the free surface position. Results were documented by plotting the distribution of velocity vectors and the streamline contours. The numerical visualization revealed the characteristics of the flow and the free surface during radial jet impingement.

Hassaneen and Rahman [2001] reported results of CFD computation of the heat transfer process in a radial impinging free liquid jet. The solution was made under steady state and laminar flow conditions. The solution was obtained for the axisymmetric radial jet with two free surfaces. The jet incidence angle, jet elevation, Reynolds number, and disc thickness were found to have strong effects on local and average Nusselt number.

The following can be some highlights of single phase liquid cooling by jet impingement.

1. The maximum heat transfer coefficient is obtained at the impingement location and it decreases significantly as the flow moves downstream and exits the heated plate. Therefore, the cooling performance of the system is directly associated with the size of the heater.

2. The capturing, treatment, and recirculation of the spent fluid is usually a design concern. Unlike an air jet, a liquid coolant cannot be depleted continuously in most engineering systems.

3. Enhancement of heat transfer by turbulence can be easily done. Pressure drop is low compared to other cooling systems and a very significant part of that happens right at the nozzle, which can be controlled if needed.

4. The heat transfer can be enhanced by using multiple jets [Wadsworth and Mudawar, 1990] or radial impingement [Hassaneen and Rahman, 2001] that produces a larger stagnation region.

JET IMPINGEMENT WITH EVAPORATION OR BOILING

At high heat flux and relatively lower flow rate, evaporation at the free surface or even boiling at the solid-fluid interface may take place. Monde and Katto [1978] and Monde [1987] reported the two-phase cooling characteristics of free circular jets. During boiling, the vigorous effusion of vapor within the wall jet splashed away a significant portion of the wall jet liquid flow. Further increases in heat flux resulted in the formation of dry patches in the outer circumference of the wall jet which propagated inward towards the impingement zone, causing the separation of the wall jet from the heated wall resulting in critical heat flux. Estes and Mudawar [1995a] showed that the critical heat flux for the free surface jet can be enhanced by jet velocity, jet diameter, or liquid subcooling. Increasing of subcooling was found to be effective in delaying the wall jet separation caused by the bubble growth.

Kumagai et al. [1995] presented experimental data for transient boiling heat transfer rate to a two dimensional water jet impinging over a thick rectangular plate. The cooling process was from 400 °C to 100 °C. The study showed that the cooling line moves from the impinging zone towards the edge of the plate in accordance with the boiling peak heat flux line. Temperature profiles inside the solid were calculated from the measured transient heat flux distribution at the solid surface. The cooling rate was very sensitive to jet velocity and degree of subcooling.

Inoue et al. [2000a] presented experimental results on the application of confined jet impingement for the removal of high heat flux from the diverter surface of a fusion reactor. Due to heat dissipation by charged particles in the reactor, the heat flux on diverter surface becomes locally near 2000 W/cm^2. Both planar impinging jet on a flat interface and jet impinging on a concave surface where a flow induced centrifugal force acts effectively along the curved surface were investigated. The distribution of critical heat flux in the confined jet was investigated as a function of distance from the center of impingement, amount of subcooling, flow velocity, and curvature of the concave surface.

Inoue at al. [2000b] studied the cooling of the divertor surface of a magnetic confinement fusion reactor that may locally reach heat fluxes as high as 3000 W/cm^2. It was necessary to develop a cooling technique that enhances the critical heat flux (CHF). A confined planar jet was applied to cool the concave surface where the centrifugal force is efficiently used to enhance the CHF. The concave impingement surface and confined flow channel provided several advantages over plane free slot jet impingement. These included: suppression of splash from liquid film, increase of radial pressure gradient normal to the curved surface that promoted departure of boiling bubbles from the heated surface as well as suppression of droplet entrainment and large bubble formation near the heated wall, variation of heat flux with angle, much reduced effects of gravity, and better control of flow velocity downstream of impingement. Based on the test data, a correlation for CHF was developed.

The following points can be noted from the above review.

1. The vigorous effusion of vapor within the thin film formed after the jet impingement may splash away a significant part of the liquid or even cause dryout.

2.. The critical heat flux can be enhanced by increasing the jet velocity, nozzle hydraulic diameter, or liquid subcooling.

3. The bubble separation and vapor transport can be enhanced by using a curved impingement surface and confining the jet.

4. Local heat flux on the order of 2000 W/cm^2 was successfully removed [Inoue et al., 2000a-b].

SPRAY COOLING

Sprays can be classified into pressure sprays and atomized sprays. Pressure sprays are formed by supplying liquid at high pressure through a small orifice while atomized sprays employ a high pressure air stream to assist the liquid breakup. Unlike jet impingement, fine droplets are impinged individually upon the heated wall, which enhances the spatial uniformity of heat removal.

Shembey et al. [1995] presented experimental results corresponding to heat transfer characteristics of liquid nitrogen spray cooling. Four different nozzles were used. The heat transfer coefficient as well as critical heat flux increased with mass flow rate for any particular nozzle. The highest heat flux was around $1.65x10^6$ W/m^2 at a superheat of approximately 16 K. At a given flow rate, the heat transfer coefficient increased with decrease in orifice size due to increase in the number of droplets as well as the impingement velocity.

Estes and Mudawar [1995b] and Mudawar and Estes [1996] investigated spray boiling and critical heat flux from a square heater that simulated an electronic device. Boiling curves were measured for different nozzles over broad ranges of flow rate and subcooling. It was observed that low wall temperature merit of nucleate boiling which is more common in jet impingement or flow boiling is not always realized with sprays. It was also found that nozzle to surface distance has a very strong effect on heat transfer for any given flow rate. The critical heat flux was found to be maximum when the spray impact area just inscribed the surface of the heater.

Oliphant et al. [1998] experimentally investigated single phase liquid jet array and spray impingement cooling techniques and compared the two options. Jet heat transfer was shown to be dependent on the number and the velocity of the impinging jets. It was found that sprays can provide the same heat transfer as jets at a significantly lower liquid mass flux. It was primarily from unsteady boundary layer resulting from the droplet impact and secondarily from evaporative cooling.

The following may be noted as highlights of the spray cooling option.

1. The spray cooling provides much more uniform distribution of heat transfer coefficient compared to other cooling options described above.

2. The best result is obtained when spray impact area just inscribes the surface [Estes and Mudawar, 1995b; Mudawar and Estes, 1996]. The nozzle to surface distance also has a very strong effect on heat transfer coefficient.

3. Compared to jet impingement, spray can provide the same cooling with lower mass flow rate [Oliphant et al., 1998].

REFRIGERATION

Phelan et al. [2001] presented a review of potential microscale and mesoscale refrigeration systems that may be used for the cooling of electronics. Utilizing refrigeration systems may provide the only means by which future high performance electronic chips can be maintained below predicted maximum temperature limit of 85 ^oC with chip level heat dissipation rate of 160 W. In contrast to other cooling techniques, a refrigeration system is capable of lowering the junction temperature below the ambient temperature. The low and constant junction temperature is expected to increase the life and reliability of the device. A number of refrigeration systems such as thermoelectric, vapor compression, Stirling, pulse tube, sorption, and reverse Brayton were discussed in terms of their technical maturity and expected developments for near future (10 years) applications. It was indicated that only thermoelectric cooler is currently commercially available for microscale application, which has low coefficient of performance (COP) and high cost.

In addition to the above research effects on specific thermal management concepts, some recent articles on comparison of different thermal management options were reviewed. Mudawar [2000] presented a very comprehensive review of phase change thermal management schemes. These include pool boiling, detachable heat sinks, channel flow boiling, microchannel and minichannel heat sinks, jet impingement, and sprays. The specific application of these techniques and their reliability and packaging concerns were explored. It was also pointed out that there is a great need for hardware innovation. Samson and Cutting [2000] presented a review of different thermal management techniques and challenges for spaceborne electronics. This include solid core thermal plane, heat pipes, sealed forced convection cooled system, advanced composite materials with high thermal conductivity, cold plate with space radiator, enclosure radiation, direct radiation integrated systems, low power electronics and optical solutions, superconducting materials, active cooling, thermal coatings, interstitial materials, thermal insulation, thermal shade, and thermal louvers. It was pointed out that the ultimate physical limitation of thermal management of spaceborne platforms is the temperature of the deep space and the ability to radiate heat to it. Elwell et al. [2001] presented an evaluation of different cooling techniques that could be used to maintain thermal equilibrium of an advanced linear motor for next generation shipboard aircraft launch and recovery system. A number of different potential approaches such as forced air cooling, direct water cooling, alternative cooling fluids, immersion cooling, heat pipes, phase change materials (PCM), vapor compression refrigeration, thermoelectric, Stirling refrigeration, Malone refrigeration, pulse tube refrigeration, thermionic cooling, thermoacoustic refrigeration, magnetic refrigeration, and ejector expansion refrigeration were evaluated. It was determined that direct liquid cooling is the best approach for this particular application. Liquid immersion cooling or PCM could be used but required a secondary loop to reject heat from the ship.

RESULTS ON CONJUGATE HEAT TRANSFER DURING LIQUID JET IMPINGEMENT

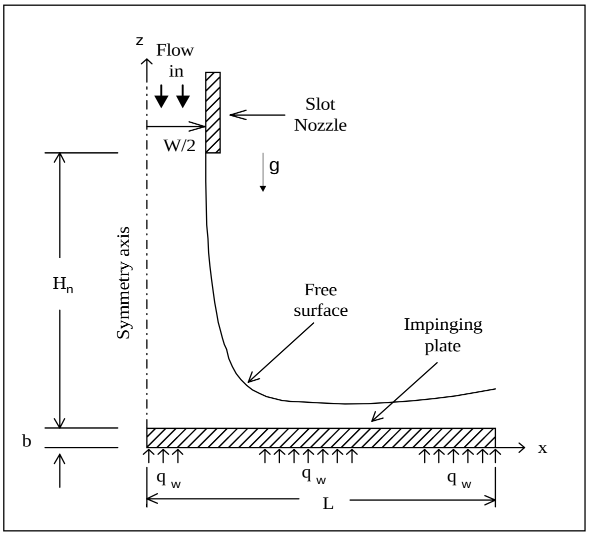

Fig. 1 Schematic of a free slot jet impinging over a solid plate with discrete heat sources

Rahman and co-workers reported a number of studies on conjugate heat transfer during liquid jet impingement. These studies focused on modeling the entire cooling system including the heat generation in power electronics, conduction of heat through the solid substrate and dissipation of it to the coolant impinging on the opposite side of the wafer. Another important feature of these studies is the variation of fluid and solid properties with temperature, which is very important for dielectric fluids (FC-77, FC-72) and lubricating oils (MIL-7808, PAO) being considered for future applications. Some recent results corresponding to the transient start-up of power in the discrete heat sources when a plate is impinged by a two-dimensional slot jet as shown in Fig. 1 is presented in this section.

Fig. 2 presents the interfacial temperature distribution at different time instants. It is noticed that at the early stages of the heat transfer process, the temperature at the interface rises uniformly at all locations resulting in a practically isothermal interface condition. This behavior is due to transient thermal storage in the fluid required to develop a thermal boundary layer starting with an isothermal initial condition. The thickness of this boundary layer increases with time and becomes significant only in the later part of the transient. The interface temperature responds accordingly. Since the leading edge of this boundary layer is located at the stagnation point and its thickness increases downstream; it can be noticed that in the later part of the transient, the temperature becomes minimum at the impinging point and maximum at the edge of the plate. The same figure presents the values for maximum-to-minimum temperature difference at the interface. As expected, the minimum to maximum temperature range at the interface increases with time.

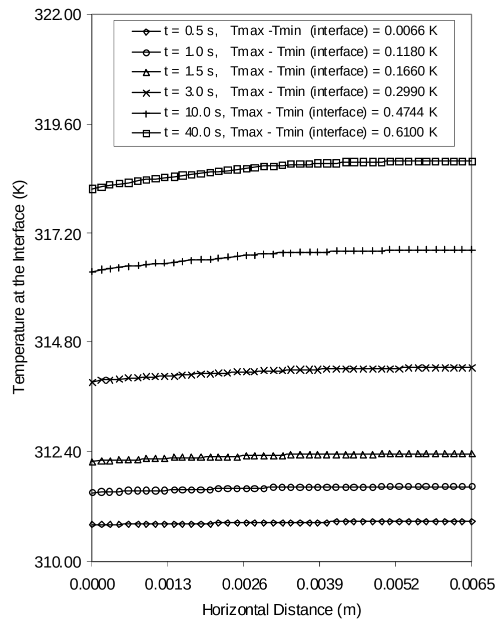

Fig. 2 Local temperature distribution at the interface (Re = 550, T_{jet} = 310 K, b = 0.005 m, H_n = 0.0085 m, Copper plate, q_w = 63 kW/m)

The variation of maximum temperature in the solid, maximum temperature at the interface, and the maximum-to-minimum temperature difference at the interface during the transient process are presented in Fig. 3 for two different Reynolds number. As expected, the temperature increases with time starting from the initial isothermal condition. A rapid increment is seen at the earlier part of the transient and it levels off as the thermal storage capacity of the solid diminishes and becomes zero at the steady state condition. It is noticed that the time required to reach the steady state is lower at higher Reynolds number. It changes from 40 s at Re = 550 to 33 s at Re = 880, because the higher velocity of the fluid helps to enhance the convective heat transfer process.

The average heat transfer coefficient and average Nusselt number variation with time for two different Reynolds numbers is presented in Fig. 4. It is noticed that both average heat transfer coefficient and average Nusselt number decrease with time and reach their

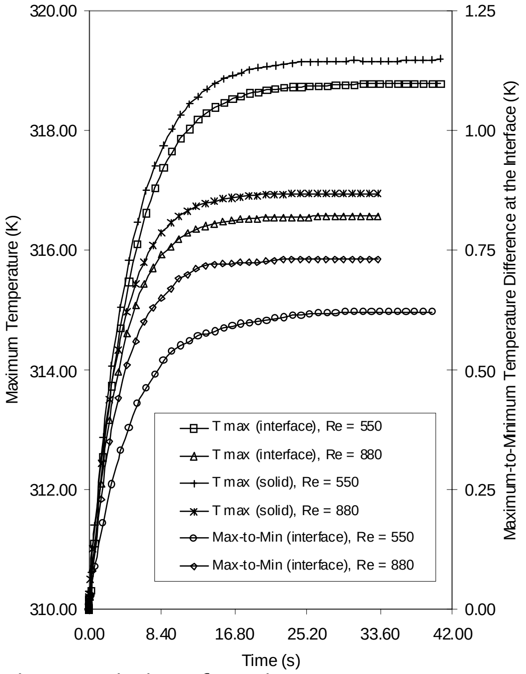

Fig. 3 Variation of maximum temperature at the interface, maximum temperature inside the solid, and maximum-to-minimum temperature difference at the interface with time for different Reynolds numbers (T_{jet} = 310 K, b = 0.005 m, H_n = 0.0085 m, Copper plate, q_w = 63 kW/m)

steady state values as an exponential decay function. Fig. 5 presents the time required to reach steady state as a function of Reynolds number. The duration of the transient decreases as the Reynolds number increases.

Another important factor that controls the transient heat transfer process is the thickness of the target plate. Its effect on the maximum temperature in the solid, maximum temperature at the interface, and maximum-to-minimum temperature difference at the interface can be observed in Fig. 6. As the thickness increases, the time required to reach steady state increases. This is expected because the thermal storage capacity of the plate is directly proportional to its thickness. Fig. 7 presents the average heat transfer coefficient and average Nusselt number variation with time for two different plate thicknesses. At early part of the transient process, the average heat transfer coefficient is larger for smaller thickness; as the transient phenomenon progresses, the average heat transfer coefficient becomes almost equal for both cases. A smaller thickness presents a lower thermal resistance to heat flux, and allows it to reach the interface faster, while in a thicker plate the heat flux reaches the interface in a more distributed form. The variation of the average Nusselt number with time is similar to that for the

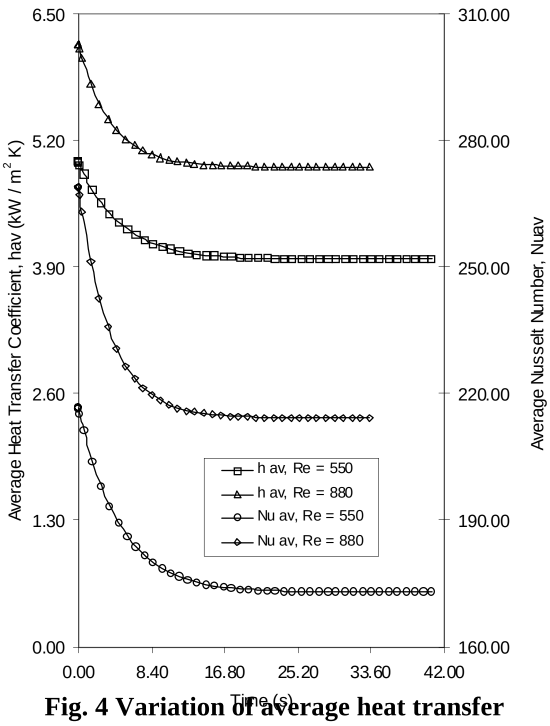

Fig. 4 Variation of average heat transfer coefficient and average Nusselt Number with time (T_{jet} = 310 K, b = 0.005 m, H_n = 0.0085 m, Copper plate, q_w = 63 kW/m)

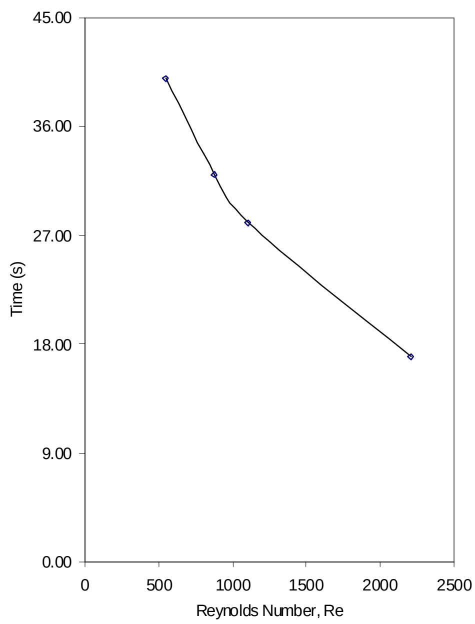

Fig. 5 Variation of time required to reach steady state with Reynolds number (T_{jet} = 310 K, b = 0.005 m, H_n = 0.0085 m, Copper plate, q_w = 63 kW/m)

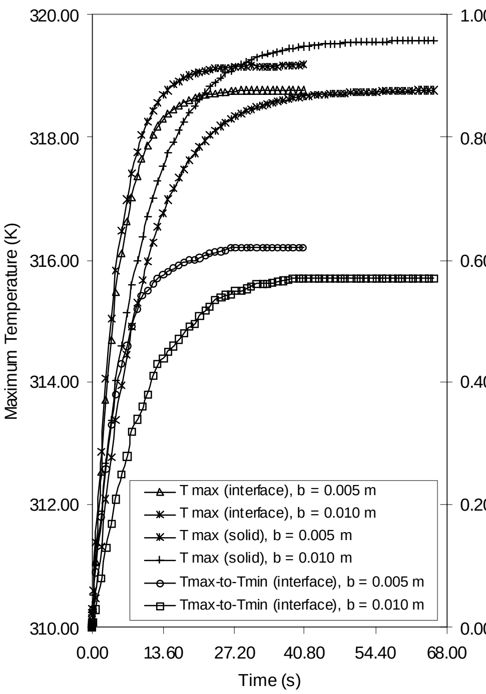

Fig. 6 Variation of maximum temperature at the interface, maximum-to-minimum temperature difference at the interface, and maximum temperature inside the solid with time for different plate thicknesses (Re = 550, T_{jet} = 310 K,H_n = 0.0085 m, Copper plate, q_w = 63 kW/m)

average heat transfer coefficient.

The maximum temperature in the solid, the maximum temperature at the interface, and maximum-to-minimum temperature difference at the interface for different materials is presented in Fig. 8. It can be noticed that the material with the highest thermal diffusivity reaches the steady state faster. Out of the materials considered in our study, diamond has much higher thermal diffusivity compared to silicon or copper. At 293 K, the values are $\alpha_{diamond}$ = 0.00129 m^2/s, α_{copper} = 0.000112 m^2/s, $\alpha_{silicon}$ = 0.0000984 m^2/s. It can be also noticed that even though copper has a slightly higher thermal diffusivity than silicon, it reaches the steady state somewhat slower. Comparing the thermal storage capacity of these materials, it can be noticed that $(\rho\ c_p)_{copper} = 3.43 \times 10^6\ J / K\ m^3$ is quite larger than $(\rho\ c_p)_{silicon} = 1.66 \times 10^6\ J / K\ m^3$. Therefore, at similar values of thermal diffusivity, the thermal storage capacity determines the amount of energy that can be stored in the solid, this capacity to store energy creates transient thermal resistance and slows down the heat conduction through the plate.

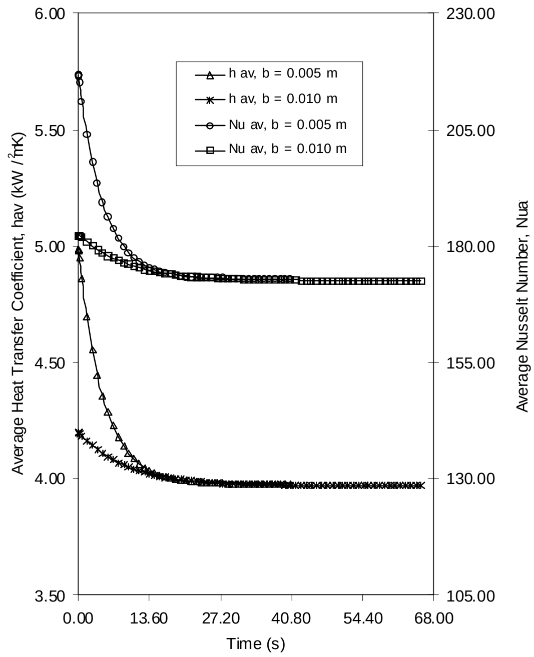

Fig. 7 Average heat transfer coefficient and average Nusselt number variation with time for two plate thicknesses (Re = 550, T_{jet} = 310 K, H_n = 0.0085 m, Copper plate, q_w = 63 kW/m)

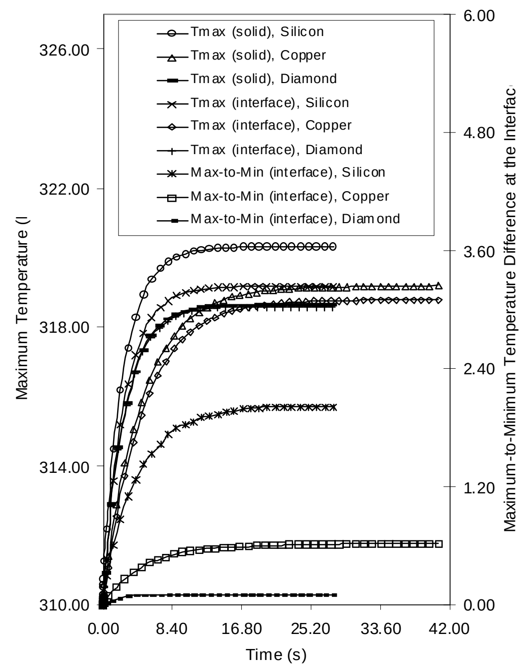

Fig. 8 Variation of maximum temperature at the interface, maximum temperature inside the solid, and maximum-to-minimum temperature difference at the interface with time for different materials (Re = 550, T_{jet} = 310 K, b = 0.005 m, H_n = 0.0085 m, q_w = 63 kW/m)

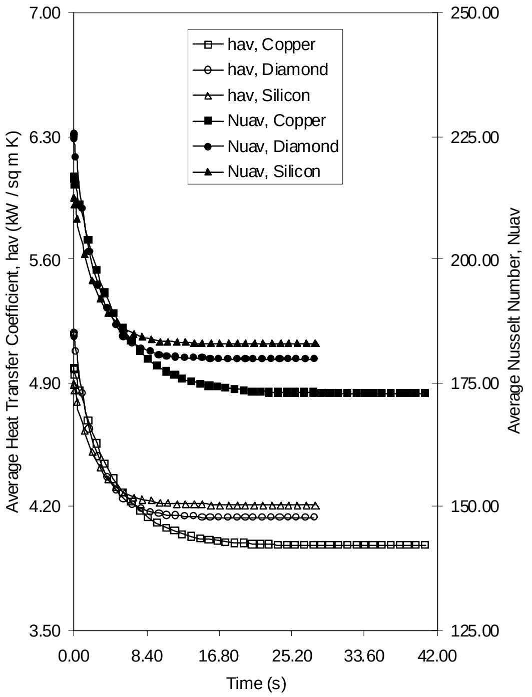

Fig. 9 Variation of average heat transfer coefficient and average Nusselt number with time for different materials (Re = 550, T_{jet} = 310 K, b = 0.005 m, H_n = 0.0085 m, q_w = 63 kW/m)

Fig. 9 presents the variation of the average heat transfer coefficient and average Nusselt number with time for different materials considered in this study. It is noticed that materials with the ability to transport the heat faster through them (diamond, silicon) have higher values at early stages of the transient process. The time required to reach steady state for different materials and different plate thicknesses is presented in Fig. 10. As expected, as the thickness increases, the time to reach steady state increases.

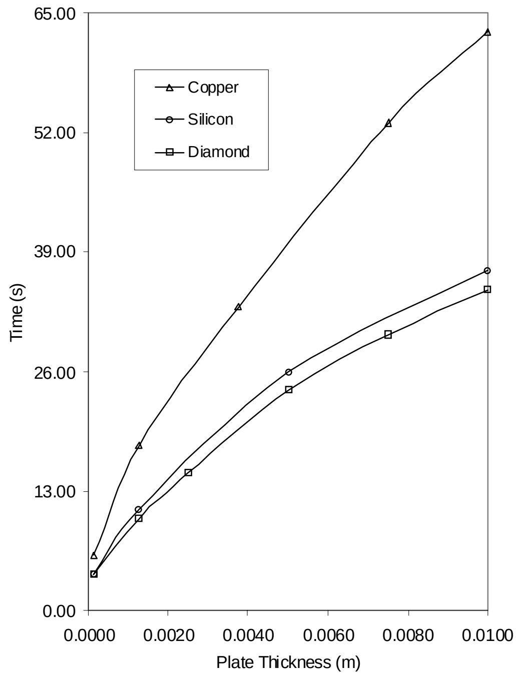

Fig. 10 Time required to reach steady state for different materials and plate thicknesses (Re = 550, T_{jet} = 310 K, H_n = 0.0085 m, q_w = 63 kW/m)

CONCLUSIONS

The following conclusions can be made regarding different high-heat-flux active thermal management techniques.

1. Using single phase flow, a microchannel heat exchanger is capable of removing a large heat flux, particularly if that appears over a small area. The pressure drop is high and is usually the limiting factor. Evaporation or boiling in a microchannel is still an area for active research and has the potential to give high thermal performance with low pressure drop.
2. During single phase liquid jet impingement, the maximum heat transfer coefficient is obtained at the impingement location and it decreases significantly downstream. Therefore, the cooling performance of the system is directly associated with the size of the heater. Multiple jets or radial impingement may be used to cover a larger area if needed.
3. Jet impingement with evaporation or boiling provides significantly higher heat transfer rate compared to other techniques. The critical heat flux can be enhanced by increasing the jet velocity, nozzle hydraulic diameter, or liquid subcooling.
4. The spray cooling provides much more uniform distribution of heat transfer coefficient compared to other cooling options. The best result is obtained when spray impact area just inscribes the heated surface. The nozzle to surface distance also has a very strong effect on heat transfer coefficient.
5. Unlike other techniques, a refrigeration system is capable of lowering the junction temperature below the ambient temperature, which is desirable in many applications. The technology still need to be improved to be competitive in terms of size and weight.

The following conclusions can be made regarding conjugate heat transfer during impingement of a slot jet.

1. The variation of average heat transfer coefficient and average Nusselt number with time present an exponential decay behavior over the duration of the transient.
2. The velocity of the jet impinging on the plate is an important parameter that affects the transient process. As the jet Reynolds number increases, the time required to reach steady state decreases.

3. The maximum temperature at the interface and the maximum temperature inside the solid decrease when Reynolds number increases, while the maximum-to-minimum temperature difference at the interface increases with Reynolds number.

4. The thickness of the plate plays an important role in the transient phenomenon; the transient time increases with thickness because the thermal storage capacity of the plate is directly proportional to its thickness. At the earlier part of the transient, the average heat transfer coefficient and average Nusselt number, are larger for smaller thickness.

5. For the materials considered (copper, diamond, silicon), it was found that the thermal storage capacity $\left(\rho\ c_p\right)$ controls the duration of the transient process, increasing the time required to reach steady state as the thermal storage capacity increases.

ACKNOWLEDGEMENT

This research was sponsored by Air Force Research Laboratory, Wright Patterson AFB, Ohio, under contracts F 33615-98-1-2923 and GS-35F-5431H (Propulsion Directorate Summer Faculty Research Program).

REFERENCES

Alkam, M. and Butler, P.B., "Transient Conjugate Heat Transfer Between a Laminar Stagnation Zone and a Solid Disk," *Journal of Thermophysics and Heat Transfer*, **8**, No. 4, 664-669, (1994).

Al-Sanea, S., "A Numerical Study of the Flow and Heat Transfer Characteristics of an Impinging Laminar Slot Jet Including Crossflow Effects", *International Journal of Heat and Mass Transfer*, **35**, No. 10, 2501-2513, (1991).

Ambatipudi, K.K. and Rahman, M.M., "Analysis of Conjugate Heat Transfer in Microchannel Heat Sinks," *Numerical Heat Transfer*, Part A, **37**, No.7, 711-731, (2000).

Arthur, J., "Experience with Microchannel and Pin-Post Water Cooling of Silicon Monochromator Crystals," *Optical Engineering*, **34**, No.2, 441-444, (1995).

Arthur, J., Tompkins, W.H., Troxel, C., Contolini, R.J., Schmitt, E., Bilderback, D.H., Henderson, C., White, J., and Settersten, T., "Microchannel Water Cooling of Silicon X-Ray Monochromator Crystals," *Review of Scientific Instrumentation*, **63**, No.1, 433-436, (1992).

Ashforth-Frost, S., Jambunatham, K., and Whitney, C.F., "Velocity and Turbulence Characteristics of a Semiconfined Orthogonally Impinging Slot Jet," *Experimental Thermal and Fluid Science*, **14**, 60-67, (1997).

Beach, R., Benett, W.J., Freitas, B.L., Mundinger, D.C., Comaskey, B.J., Solarz, R.W., and Emanuel, M. A., "Modular Microchannel Cooled Heat Sinks for High Average Power Laser Diode Arrays," *IEEE Journal of Quantum Electronics*, **28**, No. 4, 966-976, (1992).

Bernhardt, A.F., Barfknect, A.T., Contolini, R.J., Malba, V., Mayer, S. T., Raley, N.F., and Tuckerman, D.B., "Multichip Packaging for Very-High-Speed Digital Systems," *Applied Surface Science*, **46**, 121-130, (1990).

Besserman, D. L., Ramadhyani, S., and Incropera, F. P., "Numerical Simulation of Laminar Flow and Heat Transfer for Liquid Jet Impingement Cooling of a Circular Heat Source with Annular Collection of the Spent Fluid," *Numerical Heat Transfer*, Part A, **20**, 263-278, (1991).

Bowers, M.B. and Mudawar, I., "High Heat Flux Boiling in Low Flow Rate, Low Pressure Drop Mini-Channel and Micro-Channel Heat Sinks," *International Journal of Heat and Mass Transfer*, **37**, No.2, 321-332, (1994).

Bula, A.J., Rahman, M.M., and Leland, J.E., "Axial Steady Jet Impinging over a Flat Disk with Discrete Heat Sources," *International Journal of Heat and Fluid Flow*, **21**, 11-21, (2000a).

Bula, A.J., Rahman, M.M., and Leland, J.E., "Numerical Modeling of Conjugate Heat Transfer During Impingement of Free Liquid Jet Issuing from a Slot Nozzle," *Numerical Heat Transfer*, Part A, **38**, No.1, 45-66, (2000b).

Chou, Y.J. and Hung, Y.H., "Impingement Cooling of an Isothermally Heated Surface with a Confined Slot Jet," *Journal of Heat Transfer*, **116**, 479-482, (1994a).

Chou, Y.J. and Hung, Y.H., "Fluid Flow and Heat Transfer of an Extended Slot Jet Impingement," *Journal of Thermophysics and Heat Transfer*, **8**, No. 3, 538-545, (1994b).

Cziesla, T., Tandogan, E., and Mitra, N.K., "Large Eddie Simulations of Heat Transfer from Impinging Slot Jets," *Numerical Heat Transfer*, Part A, **32**, 1-17, (1997).

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, 1207-1216, (1994).

Elwell, R.J., Garman, R.W., and Doyle, M., "Thermal Management Techniques for an Advanced Linear Motor in an Electric Aircraft Recovery System," *IEEE Transactions on Magnetics*, **37**, No.1, 476-479, (2001).

Estes, K.A. and Mudawar, I., "Comparison of Two-Phase Electronic Cooling Using Free Jets and Sprays," *Journal of Electronic Packaging*, **117**, 323-332, (1995a).

Estes, K.A. and Mudawar, I., "Correlation of Sauter Mean Diameter and CHF for Spray Cooling of Small Surfaces," *International Journal of Heat and Mass Transfer*, **38**, 2985-2996, (1995b).

Fedorov, A.G. and Viskanta, R., "Three-Dimensional Conjugate Heat Transfer in the Microchannel Heat Sink for Electronic Packaging," *International Journal of Heat and Mass Transfer*, **43**, 399-415, 2000.

Garg, V.K. and Jayaraj, S., "Boundary Layer Analysis for Two-dimensional Slot Jet Impingement on Inclined Plates," *Journal of Heat Transfer*, **110**, 577-582, (1988).

Garimella, S.V. and Nenaydykh, B., "Nozzle-Geometry Effects in Liquid Jet Impingement Heat Transfer," *International Journal of Heat and Mass Transfer*, **39**, No.14, 2915-2923, (1996).

Garimella, S.V. and Rice R.A., "Confined and Submerged Liquid Jet Impingement Heat Transfer," *Journal of Heat Transfer*, **117**, 871-877, (1995).

Glauert, M.B., "The Wall Jet," *Journal of Fluid Mechanics*, **1**, No. 6, 625-643, (1956).

Goodson, K. E., Kurabayashi, K., and Pease, R. F., "Improved Heat Sinking for Laser-Diode Arrays Using Microchannels in CVD Diamond," *IEEE Transactions on Components, Packaging, and Manufacturing Technology*, Part B, **20**, No. 1, 104-109, (1997).

Harms, T. M., "Heat Transfer and Fluid Flow in Deep Rectangular Liquid-Cooled Microchannels Etched in a (110) Silicon Substrate," M.S. Thesis, University of Cincinnati, Cincinnati, Ohio, (1997).

Harms, T.M., Kazmierczak, M., Gerner, F.M., Hölke, A., Henderson, H.T., Pilchowski, J., and Baker, K., "Experimental Investigation of Heat Transfer and Pressure Drop through Deep Microchannels in a (110) Silicon Substrate," Proc. ASME Heat Transfer Division, Vol. 1, ASME / HTD-Vol. 351, pp. 347 – 357, (1997).

Hassaneen, A. and Rahman, M.M. "Analysis and Visualization of Free Surface Flow Characteristics of a Radial Impinging Free Liquid Jet," Proceedings of the Millennium 9th International Symposium on Flow Visualization, Edinburgh, Scotland, U.K., CD-ROM, pp. 125.1-125.7, (2000).

Hassaneen, A. and Rahman, M.M. "Thermal Characteristics of a Free Radial Liquid Jet Impinging on a Hot Surface," to appear at the Proceedings of 2001 ASME International Mechanical Engineering Congress and Exposition, New York, N.Y., November 11-16, (2001).

Hrycak, P., "Heat Transfer from Round Impinging Jets to a Flat Disk," *International Journal of Heat and Mass Transfer*, **26**, No. 12, 1857-1865, (1983).

Inoue, A., Ui, A., Yamazaki, Y., Matsusita. H., and Lee, S.R., "Studies on Cooling of High Heat Flux Surface in Fusion Reactor by Impinging Planar Jet Flow," *Fusion Engineering and Design*, **51-52**, 781-787, (2000a).

Inoue, A., Ui, A., Yamazaki, Y., and Lee, S., "Studies on Cooling by Two-Dimensional Confined Jet Flow of High Heat Flux Surface in Fusion Reactor," *Nuclear Engineering and Design*, **200**, 317-329, (2000b).

Jiang, L., Koo, J.M., Zeng, S., Mikkelsen, J.C., Zhang, L., Zhou, P., Santiago, J.G., Kenny, T.W., Goodson, K.E., Maveety, J.G., and Quan, A.T., "Two-Phase Microchannel Heat Sinks for an Electrokinetic VLSI Chip Cooling System," Proceedings of the Seventeenth IEEE Semi-Therm Symposium, San Jose, California, pp. 153-157, (2001).

Kim, S.J. and Kim, D., "Forced Convection in Microstructures for Electronic Equipment Cooling," *Journal of Heat Transfer*, **121**, 639-645, (1999).

Knight, R.W., Gooding, J.S., and Hall, D.J., "Optimal Thermal Design of Forced Convection Heat Sinks - Analytical," *Journal of Electronic Packaging*, **113**, No. 3, 313 – 321, (1991).

Kumagai, S., Suzuki, S., Sano, Y., and Kawazoe, M., "Transient Cooling of a Hot Metal Slab by Impinging Jet with Boiling Heat Transfer," ASME/JSME Thermal Engineering Conference, Vol. 2, pp. 347-352, (1995).

Laschefski, H., Cziesla, T., Biswas, G., and Mitra, N.K., "Numerical Investigation of Heat Transfer by Rows of Rectangular Impinging Jets," *Numerical Heat Transfer*, Part A, **30**, 87-101, (1996).

Lee, D.Y. and Vafai. K., "Comparative Analysis of Jet Impingement and Microchannel Cooling for High Heat Flux Applications," *International Journal of Heat and Mass Transfer*, **42**, 1555-1568, (1999).

Lee, X.C., Ma, C.F., Zheng, Q., Zhuang, Y., and Tian, Y.Q., "Numerical Study of Recovery Effect and Impingement Heat Transfer with Submerged Circular Jets of Large Prandtl Number Liquid," *International Journal of Heat and Mass Transfer*, **40**, No. 11, 2647-2653, (1997).

Leland, J.E. and Pais M.R., "Free Jet Impingement Heat Transfer of a High Prandtl Number Fluid Under Conditions of Highly Varying Properties," *Journal of Heat Transfer*, **121**, 592-597, (1999).

Lin, Z.H., Chou, Y.J., and Hung, Y.H., "Heat Transfer Behaviors of a Confined Slot Jet Impingement," *International Journal of Heat and Mass Transfer*, **40**, No. 5, 1095-1107, (1997).

Liu, X. and Lienhard, J.H., "Liquid Jet Impingement Heat Transfer on a Uniform Flux Surface," Heat Transfer Phenomena in Radiation, Combustion, and Fires, American Society of Mechanical Engineers, HTD/Vol. 106, pp. 523-530, (1989).

Liu, X, Lienhard, J.H., and Lombara, J.S., "Convective Heat Transfer by Impingement of Circular Liquid Jets," *Journal of Heat Transfer*, **113**, 571-582, (1991).

Ma, C.F., Zheng, Q., Sun, H., Wu. K., Gomi, T., and Webb, B.W., "Local Characteristics of Impingement Heat Transfer with Oblique Round Free Surface Jets of Large Prandtl Number Liquid," *International Journal of Heat and Mass Transfer*, **40**, No. 10, 2249-2259, (1997a).

Ma, C.F., Zheng, Q., Lee, S.C., and Gomi, T., "Impingement Heat Transfer and Recovery Effect with Submerged Jets of Large Prandtl Number Liquid – I. Unconfined Circular Jets," *International Journal of Heat and Mass Transfer*, **40**, No. 6, 1481-1490, (1997b).

Ma, C.F., Zhuang, Y., Lee, S.C., and Gomi, T., "Impingement Heat Transfer and Recovery Effect with Submerged Jets of Large Prandtl number Liquid – II. Initially Laminar Confined Slot Jets," *International Journal of Heat and Mass Transfer*, **40**, No. 6, 1491-1500, (1997c).

Ma, C. F., Zheng, Q., and Ko, S.Y., "Local Heat Transfer and Recovery Factor with Impinging Free Surface Circular Jets of Transformer Oil," *International Journal of Heat and Mass Transfer*, **40**, No. 18, 4295-4308, (1997d).

Maddox, D.E. and Bar-Cohen, A., "Thermofluid Design of Single-Phase Submerged-Jet Impingement Cooling for Electronic Components," *Journal of Electronic Packaging*, **116**, 237-240, (1994).

Metzger, D.E., Cummings, K.N., and Ruby, W.A., "Effects of Prandtl Number on Heat Transfer Characteristics of Impinging Liquid Jets," Proceedings of the 5th International Heat Transfer Conference, Heat Transfer 1974, Japan Society of Mechanical Engineers, Vol. 2, pp. 20-24, (1974).

Missaggia, L.J. and Walpole, J.N., "A Microchannel Heat Sink with Alternating Directions of Water Flow in Adjacent Channels," Proc. SPIE 1991, Vol. 1582, pp. 106 – 111, (1991).

Monde, M., "Critical Heat Flux in Saturated Forced Convection Boiling on a Heated Disk with an Impinging Jet," *Journal of Heat Transfer*, **109**, 991-996, (1987).

Monde, M. and Katto, Y., "Burnout in a High Heat Flux Boiling System with an Impinging Jet," *International Journal of Heat and Mass Transfer*, **21**, 295-305, (1978).

Mudawar, I., "Assessment of High-Heat-Flux Thermal Management Schemes," Proceedings of ITHERM 2000, Institute of Electrical and Electronics Engineers, Piscataway, New Jersey, Vol. 1, pp. 1-20, (2000).

Mudawar, I. and Estes, K.A., "Optimizing and Predicting CHF in Spray Cooling of a Square Surface," *Journal of Heat Transfer*, **118**, 672-679, (1996).

Mundinger, D., Beach, R., Benett, W., Solarz, R., Krupke, W., Staver, R., and Tuckerman, D., "Demonstration of High-Performance Silicon Microchannel Heat Exchangers for Laser Diode Array Cooling," *Applied Physics Letters*, **53**, No.12, 1030-1032, (1988).

Nakoryakov, V. E., Pokusaev, B. G., and Troyan, E. N., "Impingement of an Axisymmetric Liquid Jet on a Barrier," *International Journal of Heat and Mass Transfer*, **21**, 1175-1184, (1978).

Nayak, D., Hwang, L. T., Turlik, I., and Reisman, A., "A High-Performance Thermal Module for Computer Packaging," *Journal of Electronic Materials*, **16**, No. 5, 357-364, (1987).

Oliphant, K., Webb, B.W., and McQuay, M.Q., "An Experimental Comparison of Liquid Jet Array and Spray Impingement Cooling in the Non-Boiling Regime," *Experimental Thermal and Fluid Science*, **18**, 1-10, (1998).

Olson, R.G. and Turkdogan, E. T., "Radial Spread of a Liquid Stream on a Horizontal Plate," *Nature*, **221**, No. 5051, 813-816, (1966).

Ozmat, B., Korman, C.S., McConnelee, P., Kheraluwala, M., Delgado, E., and Fillion, R., "A New Power Module Packaging Technology for Enhanced Thermal Performance," Proceedings of ITHERM 2000, Institute of Electrical and Electronics Engineers, Piscataway, New Jersey, Vol. 2, pp. 287-296, (2000).

Pan, Y., Stevens, J., and Webb, B. W., "Effect of Nozzle Configuration on Transport in the Stagnation Zone of Axisymmetric, Impinging Free-Surface Liquid Jets : Part 2 – Local Heat Transfer," *Journal of Heat Transfer*, **114**, 880-886, (1992).

Peng, X.F. and Peterson, G.P., "Convective Heat Transfer and Flow Friction for Water Flow in Microchannel Structures," *International Journal of Heat and Mass Transfer*, **39**, No. 12, 2599-2608, (1996).

Phelan, P.E., Chiriac, V., and Lee, T., "Current and Future Miniature Refrigeration Cooling Technologies for High Power Microelectronics," Proceedings of the Seventeenth IEEE Semi-Therm Symposium, pp. 158-167, San Jose, California, (2001).

Phillips, R. J., "Forced-Convection, Liquid-Cooled Microchannel Heat Sinks," M.S. Thesis, Massachusetts Institute of Technology, Cambridge, Massachusetts, (1987).

Polat, S., Mujumdar, A.S., and Douglas, W.J.M., "Impingement Heat Transfer Under a Confined Slot Jet. Part I: Effect of Surface Throughflow," *Canadian Journal of Chemical Engineering*, **69**, 266-273, (1991a).

Polat, S., Mujumdar, A.S., and Douglas, W.J.M., "Impingement Heat Transfer Under a Confined Slot Jet. Part II: Effect of Surface Motion and Throughflow," *Canadian Journal of Chemical Engineering*, **69**, 274-280, (1991b).

Rahman, M.M., "Measurements of Heat Transfer in Microchannel Heat Sinks," *International Communications in Heat and Mass Transfer*, **27**, No.4, 495-506, (2000).

Rahman, M.M., Bula, A.J., and Leland, J.E., "Conjugate Heat Transfer During Free Jet Impingement of a High Prandtl Number Fluid," *Numerical Heat Transfer*, Part B, **36**, 139-162, (1999).

Rahman, M.M., Bula, A.J., and Leland, J.E., "Analysis of Transient Conjugate Heat Transfer to a Free Impinging Jet," *Journal of Thermophysics and Heat Transfer*, **14**, No.3, 330-339, (2000).

Rahman, M.M. and Gui, F., "Design, Fabrication, and Testing of Microchannel Heat Sinks for Aircraft Avionics Cooling," Proceedings 28th Intersociety Energy Conversion Engineering Conference, Atlanta, Georgia, Vol.1, pp. 1.1-1.6, (1993).

Rice, R.A. and Garimella, S.V., "Heat Transfer from Discrete Sources Using an Axisymmetric, Submerged and Confined Liquid Jet," Proceedings of the 10th International Heat Transfer Conference, Brighton, U.K., Vol. 3, pp. 89-94, (1994).

Rittmann, J.G. and Tuckerman, D.B., "High Performance Cooling for Wafer-Scale Integrated Circuits," Lawrence Livermore National Laboratory Annual Report of Status, pp. 41-43, (1983).

Roy, S.K. and Avanic, B.L., "A Very High Heat Flux Microchannel Heat Exchanger for Cooling of Semiconductor Laser Diode Arrays," *IEEE Transactions on Components, Packaging, and Manufacturing Technology*, Part B, **19**, No.2, 444-451, (1996).

Rujano, J.R. and Rahman, M.M., "Transient Response of Microchannel Heat Sinks in a Silicon Wafer," *Journal of Electronic Packaging*, **119**, No.4, 239-246, (1997).

Samson, J.R. and Cutting, F.M., "Thermal Management of High Performance Computing in Spaceborne Applications," Proceedings of ITHERM 2000, Institute of Electrical and Electronics Engineers, Piscataway, New Jersey, Vol. 1, pp. 247-254, (2000).

Schaffer D., Incropera, F.P., and Ramadhyani, S., "Planar Liquid Jet Impingement Cooling of Multiple Discrete Heat Sources," *Journal of Electronic Packaging*, **113**, 359-366, (1991).

Seyedein, S.H., Hasan, M., and Mujumdar, A.S., "Modeling of a Single Confined Turbulent Slot Jet Impingement Using Various Turbulence Models," *Applied Mathematical Modeling*, **18**, 526-537, (1994).

Seyedein, S.H., Hasan, M., and Mujumdar, A.S., "Turbulent Flow and Heat Transfer from Confined Multiple Impinging Slot Jets," *Numerical Heat Transfer*, Part A, **27**, 35-51, (1995).

Sehmbey, M.S., Chow, L.C., Hahn, O.J., and Pais, M.R., "Spray Cooling of Power Electronics at Cryogenic Temperatures," *Journal of Thermophysics and Heat Transfer*, **9**, No.1, pp. 123-128, (1995).

Stevens, J. and Webb, B.W., "Local Heat Transfer Coefficients Under an Axisymmetric, Single-Phase Liquid Jet," *Journal of Heat Transfer*, **113**, 71-78, (1991).

Tso, C.P. and Mahulikar, S.P., "Combined Evaporating Meniscus-Driven Convection and Radiation in Annular Microchannels for Electronics Cooling Application," *International Journal of Heat and Mass Transfer*, **43**, 2245-2256, (2000).

Tuckerman, D.B. and Pease, R.F., "High Performance Heat Sinking for VLSI," *IEEE Electron Device Lett.* EDL-2, 126-129 (1981).

Tuckerman, D.B. and Pease, R. F., "Optimized Convective Cooling Using Micromachined Structures," *Electrochemical Society Extended Abstracts*, **82**, 197-198, (1982).

Tuckerman, D.B. and Pease, R. F., "Microcapillary Thermal Interface Technology for VLSI Packaging," Symposium on VLSI Technology, Digest of Technical Papers, pp. 60–61, (1983).

Vader, D.T., Incropera, F.P., and Viskanta, R., "Local Convective Heat Transfer from a Heated Surface to an Impinging, Planar Jet of Water," *International Journal of Heat and Mass Transfer*, **34**, No. 3, 611-623, (1991).

Vidmar, R.J. and Barker, R.J., "Microchannel Cooling for High-Energy Particle Transmission Window, an RF Transmission Window, and VLSI Heat Dissipation," *IEEE Transactions on Plasma Science*, **26**, No.3, 1031-1043, (1998).

Wadsworth, D.C. and Mudawar, I., "Cooling of a Multichip Electronic Module by Means of Confined Two-Dimensional Jets of Dielectric Liquid," *Journal of Heat Transfer*, **112**, 891-898, (1990).

Walpole, J.N., Begley, D., Ettenberg, M., Roychoudhuri, C., Scifres, D., Simmons, W.W. "Panel Discussion on Two-Dimensional Laser Array Technology Comparison: Stack-and-Rack versus Monolithic," Proc SPIE 1988, Vol. 893, pp. 131 – 134, (1988).

Wang, B.X. and Peng, X.F., "Experimental Investigation on Liquid Forced Convection Heat Transfer Through Microchannels," *International Journal of Heat and Mass Transfer*, **37**, 73-82, (1994).

Wang, X.S., Dagan, Z., and Jiji, L.M., "Heat Transfer Between a Circular Free Impinging Jet and a Solid Surface with Non-Uniform Wall Temperature or Wall Heat Flux – 1. Solution for the Stagnation Region," *International Journal of Heat and Mass Transfer*, **32**, No. 7, 1351-1360, (1989a).

Wang, X.S., Dagan, Z., and Jiji, L.M., "Heat Transfer Between a Circular Free Impinging Jet and a Solid Surface with Non-Uniform Wall Temperature or Wall Heat Flux – 2. Solution for the Boundary Layer Region," *International Journal of Heat and Mass Transfer*, **32**, No. 7, 1361-1371, (1989b).

Wang, X.S., Dagan, Z., and Jiji, L.M., "Conjugate Heat Transfer between a Laminar Impinging Liquid Jet and a Solid Disk," *International Journal of Heat and Mass Transfer*, **32**, No. 11, 2189-2197, (1989c).

Wang, X. S., Dagan, Z., and Jiji, L. M., "Prediction of Surface Temperature and Heat Flux of a Microelectronic Chip with Jet Impingement Cooling, *Journal of Electronic Packaging*, **112**, 57-62, (1990).

Weisberg, A., Bau, H. H., and Zemel, J.N., "Analysis of Microchannels for Integrated Cooling," *International Journal of Heat and Mass Transfer*, **35**, No. 10, 2465-2474, (1992).

Wolf, D.H., Viskanta, R., and Incropera, F.P., "Local Convective Heat Transfer from a Heated Surface to a Planar Jet of Water with Nonuniform Velocity Profile," *Journal of Heat Transfer*, **112**, 899-905, (1990).

Womac, D.J., Ramadhyani, S., and Incropera, F.P., "Correlating Equations for Impingement Cooling of Small Heat Sources with Single Circular Liquid Jets," *Journal of Heat Transfer*, **115**, 106-115, (1993).

Womac, D.J., Incropera, F.P., Ramadhyani S., "Correlating Equations for Impingement Cooling of Small Heat Sources with Multiple Liquid Jets," *Journal of Heat Transfer*, **116**, 482-486, (1994).