Thermal Management Using Synthetic Jet Ejectors

Raghav Mahalingam, Nicolas Rumigny, and Ari Glezer

***Abstract*—The utility of a synthetic jet ejector for thermal management at low flow rates is discussed. A synthetic jet ejector typically consists of a primary "zero-mass-flux" unsteady jet driving a secondary airflow through a low profile, high aspect ratio channel. A simple configuration of a nominally two-dimensional jet ejector in a rectangular channel is used to investigate the effects of channel width on the induced flow rate, power dissipated, heat transfer coefficient and thermal efficiency. An active heat sink for high power microprocessors is developed using the jet ejector concept and its performance is discussed.**

***Index Terms*—Active heat sink, channel cooling, jet ejector, synthetic jets.**

I. Introduction

THE RISE in heat flux levels of integrated circuits accompanied by a shrinking thermal budget and device size has placed steep requirements on cooling solutions. Thermal issues exist over a wide range of power dissipation levels, from hand-held devices that dissipate a few watts to high-performance microprocessors dissipating over 100 W. The International Technology Roadmap for Semiconductors, 2001 [1] predicts a junction-ambient thermal resistance requirement of 0.33 °C/W for cost performance computers by 2010, and 0.18 °C/W for high performance computers. These steep cooling requirements have prompted the development of advanced two-phase and pumped liquid cooling techniques. However, as noted by Bar-Cohen [2], consumer-oriented systems still focus on air cooling approaches due to their simplicity and relative ease of implementation. Forced convection cooling using air is typically based on the use of various configurations of fans and blowers that can fit in medium and large scale enclosures and are used both for global air circulation and for local augmentation (e.g., in conjunction with heat sinks). In order to achieve increased local power dissipation levels with fan-heat-sink configurations, designers have used both copper heat sinks or aluminum heat sinks with copper inlays as well as larger fans driving higher flow rates. It is noteworthy that although fans can supply ample volume flow rates they are limited by relatively low pressure drop and, in addition, they are hindered by noise, long-term reliability, and low thermal effectiveness.

Work at Georgia Tech over the past several years has demonstrated the utility of synthetic jet actuators for highly efficient localized cooling of integrated circuits. These jets are formed by time-periodic, alternate suction and ejection of fluid through an orifice bounding a small cavity, by the time periodic motion of a diaphragm that is built into one of the walls of the cavity. Unlike conventional jets, synthetic are "zero-mass-flux" in nature and produce fluid flow with finite momentum with no mass addition to the system and without the need for complex plumbing (Smith and Glezer, [3]). Because of their ability to direct airflow along heated surfaces in confined environment and induce small-scale mixing, these jets are ideally suited for cooling applications at the package and heat sink levels. While there is extensive literature on cooling with steady and unsteady conventional jets (e.g., Jambunathan *et al.* [4]), the concept of using synthetic jets for heat transfer is relatively new. It was first implemented by Thompson *et al.* [5] who demonstrated a 250% increase in power dissipation over natural convection for direct normal impingement cooling of a 49-element multichip module using a single synthetic jet having a diameter of 1.6 mm. In a later investigation, Russell [6] showed that the inherent coupling between a local synthetic jet and global air flow (driven by a conventional fan) can be exploited for enhanced heat transfer at the package level at substantially reduced global air flow. Recently, Mahalingam and Glezer [7] developed an integrated active heat sink based on synthetic jets for heat dissipation at power levels over 100 W at flow rates of about 3–4 CFM. Mahalingam and Glezer [8] also demonstrated a low-volume active heat sink based on an impinging synthetic jet for moderate power dissipation requirements.

The present paper reports an investigation of heat removal by synthetic jet ejectors in relatively high aspect ratio channels. Section II describes the jet ejector principle and experiments performed in a channel cooled using this concept. Section III presents a cooling module for high power microprocessors developed using an integrated synthetic jet. Finally, conclusions are discussed in Section IV.

II. Cooling With Synthetic Jet Ejectors

A. Experimental Setup

The principle of jet ejectors or pumps [9], [10] has been known for several decades. In conventional jet ejectors, the primary jet is formed by ducting net mass flow from a continuous jet into the entry region of a channel. The low pressure created by the primary jet as it discharges into the channel results in the entrainment of ambient fluid, thus creating an increase in overall flow rate through the channel. The present jet ejector

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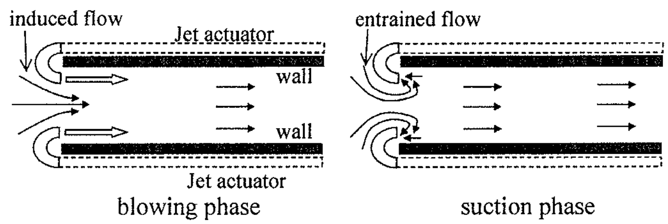

Fig. 1. Basic principle of operation of a synthetic jet ejector. Note that the flow some distance downstream of channel entrance is always from left to right.

consists of a primary, high momentum "zero-mass-flux" synthetic jet driving a secondary airflow through a channel. The use of synthetic jets as the primary jet is an attractive option since the only input to the primary jet is electrical, requiring no plumbing and flow sources. Similar to conventional jet ejectors, a low-pressure region is formed at the inlet region of the channel during the blowing stroke of the synthetic jet resulting in the entrainment of ambient fluid. However, considerably larger secondary flow entrainment is achieved during the suction stroke of the synthetic jet. A schematic of the flow induced by a synthetic jet ejector within a channel between two fins of a heat sink is shown in Fig. 1. The flow is measured using particle image velocimetry (PIV).

In the figure, the primary jets issue along the channel walls (fins) from contoured nozzles that form the inlet to the channel. The secondary flow is entrained through the channel between the fins. As will presented later in this paper, a cooling module integrated with a jet ejector was designed using an array of such channels, so that each fin is straddled by a pair of synthetic jets thereby creating a jet ejector that entrains cool ambient air upstream of the heat sink and discharges it into the channels between the fins. Each primary jet is comprised of discrete vortices that presumably break up the thermal boundary layer and enhance the heat transfer from the surface of the fins.

The effects of the channel geometry on the induced flow rate, power dissipated, resultant heat transfer coefficients and thermal efficiency are investigated in a prototypical modular channel that is fabricated of two heated walls, having a variable distance in order to change the channel width and aspect ratio (Fig. 2). There are two synthetic jets, each alongside a wall of the channel as illustrated in the top view and the jets blow into the channel parallel to the channel walls. Each jet spans the entire height of the channel and its orifice width is 0.5 mm. The operating frequency of the jets in the present paper is 250 Hz.

Each heated surface is made of an aluminum plate heated with a flat foil heater that provides a constant heat flux along the wall. The wall length and height are fixed resulting in varying channel width and aspect ratio. The backside of the heated surfaces as well as the narrow top and bottom surfaces are fabricated out of high temperature machinable glass ceramic with embedded air pockets. It is estimated that about 8% of the input power is lost through the insulation under forced convection. The temperature at several points in the setup is monitored using T-type thermocouples. The air temperature is measured using a thermocouple that can be traversed along the length of the channel. The thermocouple readings are digitized using a PC-based A/D board where cold-junction compensation is performed using an IC-sensor embedded in the shielded I/O connector block, which is used as the interface between the A/D board and the thermo-

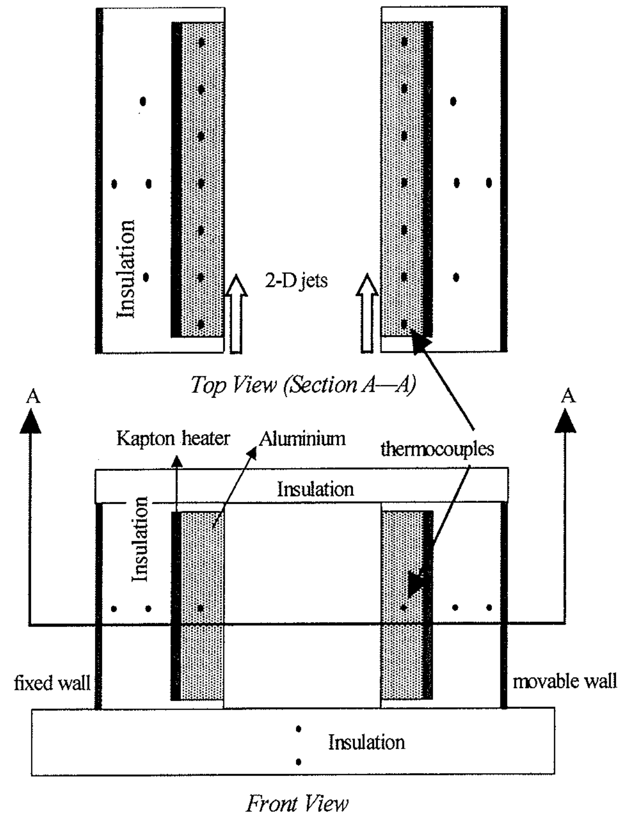

Fig. 2. Schematic diagram of the setup for measurements of jet ejector channel cooling.

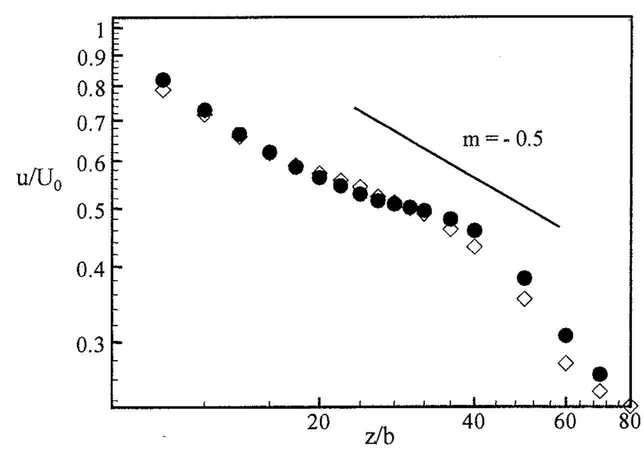

Fig. 3. Streamwise variation of the free jet centerline velocity ($U_0 = 11$ m/s, $b = 0.5$ mm). [(◇) Jet 1, (•) Jet 2.]

couple wires. The measurements have a resolution of 0.024% of full scale, resulting in a temperature resolution of 0.03 °C, (the temperature limits are set to 120 °C). The maximum error for each temperature reading is no greater than ±1.5 °C, which corresponds to an error of ±1.5–6% for the temperature range of the present data. The flow at the exit of the channel is measured using a miniature total pressure probe having an outer diameter of 0.8 mm (A maximum of 1% flow blockage based on the exit area of each channel). The pressure is measured using a 0.2-Torr pressure transducer, which yields a resolution of 0.1 m/s.

B. Results

Fig. 3 shows the streamwise decay of the time-averaged (normalized) centerline velocity for the free synthetic jet actuators

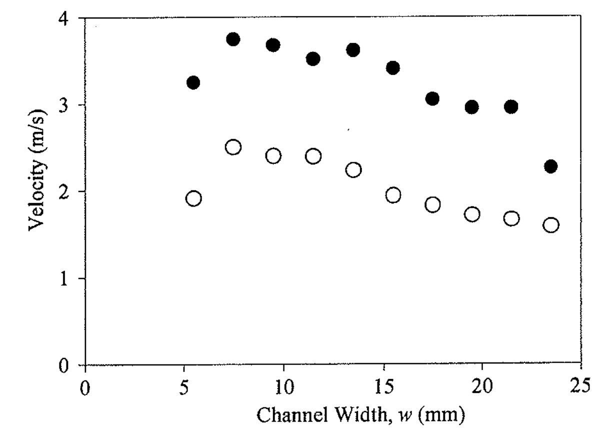

Fig. 4. Variation of channel centreline (•) and average (o) velocities.

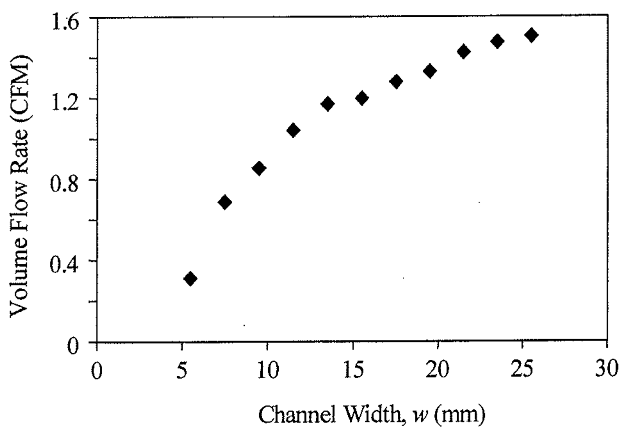

Fig. 5. Variation in induced volume flow rate.

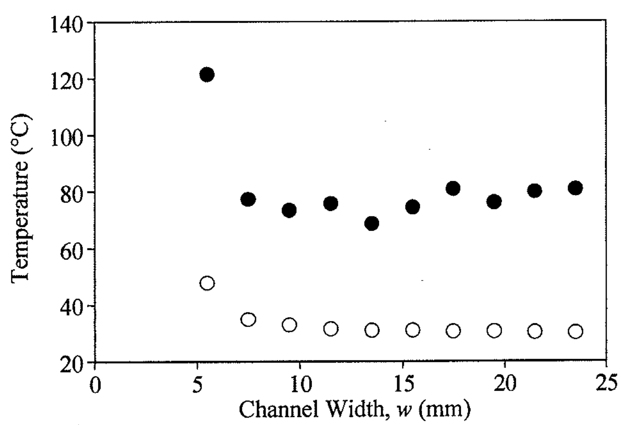

Fig. 6. Variation in average wall (•) and air (o) temperatures, $Q_{\mathrm{wall}} = 4$ W.

(i.e., prior to their insertion into the channel). These data show that the jets (which are almost identical) decay rapidly from about 10 to 3 m/s within 40 mm from the jet exit. (The decay rate for conventional 2-D jet is shown for reference.)

The effect of channel width, w on the flow characteristics and heat transfer is shown in Figs. 4–12 below. The variations of the centerline and average (over the exit plane) velocities are shown in Fig. 4. Note that since each jet is attached to one of the channel walls, the channel inlet is effectively blocked when the $w = 3.43$ mm. For the given jet operating conditions, the centerline and average velocities have a maximum when $w \approx 10$ mm, which indicates that this width is optimal for achieving the highest secondary flow momentum.

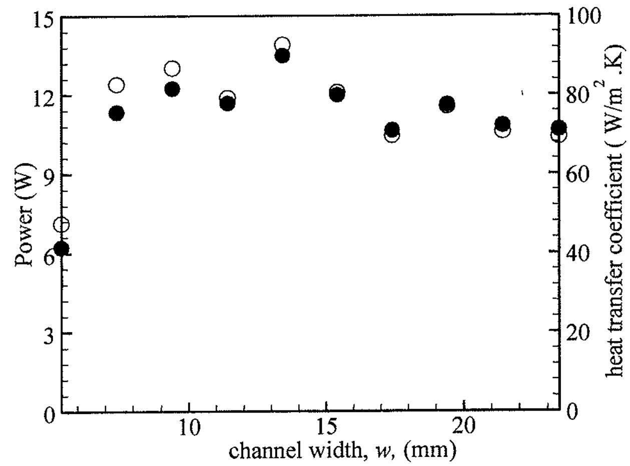

Fig. 7. Variation in power dissipated (•) and heat transfer coefficient (o), $T_{\mathrm{wall}} = 100\,^{\circ}$C.

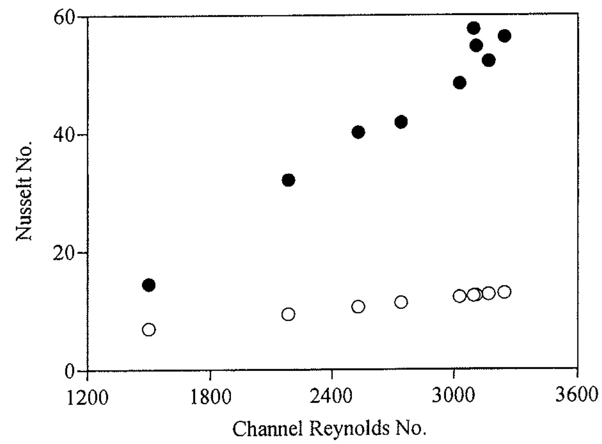

Fig. 8. Dependence of channel Nusselt number on Re: (o) empirical fully developed turbulent channel flow correlation, (•) synthetic jet ejector.

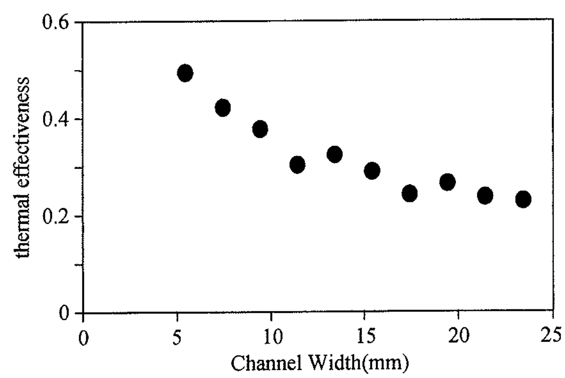

Fig. 9. Thermal effectiveness.

It is interesting to note that although the secondary (induced) flow has a distinct maximum for a 10-mm-wide channel, the volume flow rate through the channel (computed from integration of the streamwise velocity at the exit plane) increases monotonically with w (Fig. 5). As expected, the channel flow rate initially increases rapidly and then tends to an asymptotic level for

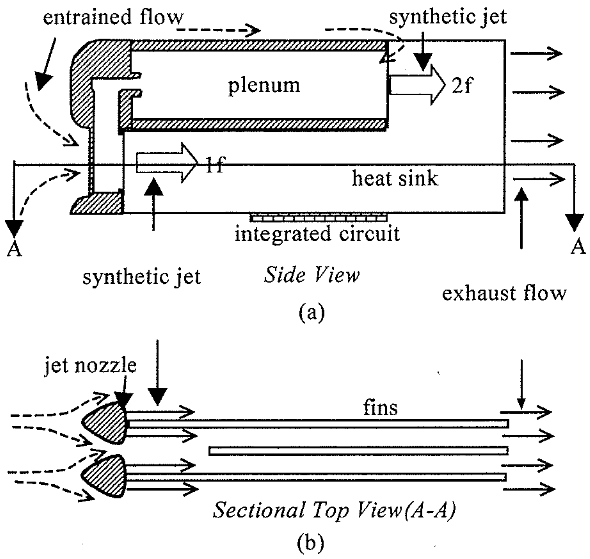

Fig. 10. Schematic diagram of a jet ejector heat sink cooling module (a) side view and (b) top view.

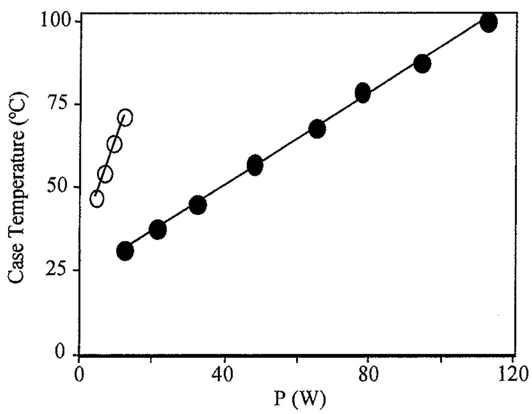

Fig. 11. Thermal performance of the integrated jet ejector module (•). The thermal performance under natural convection is shown for reference (o).

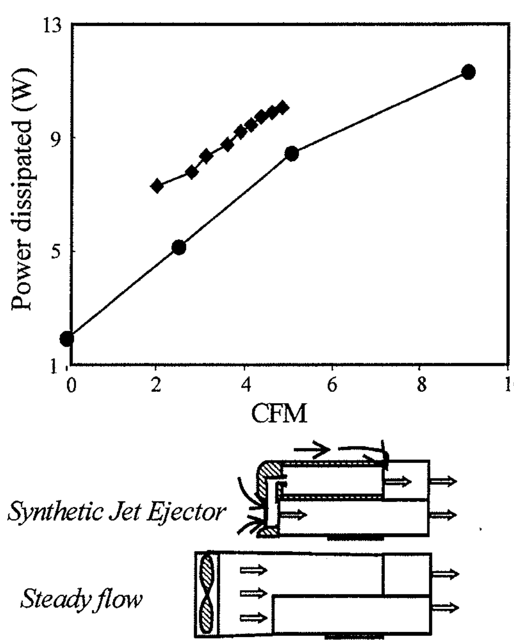

Fig. 12. Comparison of a synthetic jet ejector (♦) and steady fan driven flow (•) on the same heat sink.

$w > 25$ mm. The volume flow rate for a channel width of 5 mm is about 0.3 CFM.

The effect of the channel flow on heat transfer is investigated at a heater power of 4 W. Fig. 6 shows the variation of the average wall and exit air temperatures with w. As expected, the difference between wall and exit air temperatures decreases with increasing w until the optimal channel width is reached. Beyond the optimal width, the wall and air temperatures, which increase and decrease, respectively and ultimately become almost invariant with channel width. The decrease in air temperature past the optimal channel width indicates lower mixing with the heated fluid along the surfaces. In fact, for channel width over 15 mm, there is essentially no increase in heat addition to the air. This fact is confirmed by the corresponding variations in the power dissipated (Fig. 7), which increases rapidly up to $w \approx 10$ mm and thereafter is essentially invariant. These results suggest that the increased mass flow rate with channel width does not necessarily lead to an increase in total power dissipation, which is clearly limited by the heat transfer coefficient at the wall. The corresponding heat transfer coefficient is calculated based on the heat transported by the air flow in the channel and the average temperature difference between the local airflow and the wall (cf. Fig. 6). As shown in Fig. 7, the heat transfer coefficient drops from approximately 90 to 70 W/m^2K at $w \approx 10$ and 25 mm, respectively. Nevertheless, the overall heat transfer coefficients are much higher than what is achieved in steady fully developed flows at similar Reynolds numbers as will be seen in the following paragraphs.

The dependence of the jet ejector flow Nusselt number on the channel Reynolds number (based on the measured centerline velocity and the hydraulic diameter of the channel) is calculated based on the estimated heat transfer coefficients and compared to the empirical correlations that determine Nusselt number as a function of channel Reynolds number for duct flows (Incropera and Dewitt [11]). The results are shown in Fig. 8. Within the Reynolds number range of the present experiments, the Nusselt numbers of the synthetic jet ejector are about six times higher than in corresponding conventional turbulent duct flows. This is presumably due to enhancement of local mixing by the synthetic jets near the duct walls.

The thermal effectiveness of the synthetic jet ejector cooling is calculated as the ratio between the ideal and actual temperature rise of the airflow through the channel. As shown in Fig. 9, the effectiveness is 0.5 for smaller channel widths, but drops rapidly to about 0.25 for a channel width of 25 mm. This is related to the amount of mixing between the hotter fluid near the wall and the mean flow. It is clear that an increase in channel width reduces mixing thus results in lower overall thermal effectiveness.

III. Synthetic Jet Ejector Cooling Module

The concept of synthetic jet ejector cooling in a channel was used to design a cooling module for high power (>100 W) microprocessors. A schematic diagram of the cooling module (which measures nominally 115 × 70 × 30 mm) is shown in Fig. 10(a) and (b). Fig. 10(a) shows a sectional side view of the cooling module which consists of an aluminum plate-fin heat sink integrated with a synthetic jet actuator (plenum and driver) which straddles the L-shaped fins. Two arrays of synthetic jets are formed as a result of the time periodic pressure changes within the plenum. The first array (marked 1f in the figure) is directed through the lower portion of the fins via nozzles that are connected to the plenum while the second array (marked 2f in the figure) is directed through the fin section that is adjacent to the plenum. Fig. 10(b) shows a top (sectional) view where the jets flow along the surfaces of the fins similar to what is shown in Fig. 2. A thermal test die with an area of 1.61 cm^2 encased in a copper heat spreader of 7.38-cm^2 area is used to assess the thermal performance. The case temperature is monitored using a T-type thermocouple having a 0.25-mm bead that is embedded in the centre of the package heat spreader. It is estimated that less than 5% of the total input power to the die is transferred to its (insulated) substrate circuit board.

The thermal performance (i.e., power dissipation) of the module is shown in Fig. 11 over a range of case temperatures. When the jet ejector is operating, the system dissipates 110 W at 100 °C, resulting in a case-to-ambient thermal resistance of 0.69 °C/W (3.14 °C/W under natural convection). It is noteworthy that this resistance is achieved with a heat sink that is fabricated out of Al 6061 which effectively means that the resistance can be substantially reduced by using a different alloy for the present heat sink (e.g., a reduction of $\sim$10% with Al 6063 and 25% with Cu) and perhaps further reduced by improving the thermal interface between the test die package and the heat sink.

The effectiveness of synthetic jet ejectors is compared to a steady flow produced by a conventional fan (Fig. 12), where the power dissipation is measured for different flow rates through the heat sink. Note that the flow generated by the fan is forced to pass through the heat sink by placing a duct and a shroud over the heat sink. These data show that for flow rate within the range of 3–5 CFM and for the same configuration, the jet ejector dissipates about 40% more power than the fan.

IV. Conclusion

The operation of a synthetic jet based ejector and its utility for electronic cooling at relatively low flow rates are discussed. A simple configuration of a 2-D synthetic jet ejector in a rectangular channel is used to study the effect of channel width on the induced flow rate, power dissipated, resultant heat transfer coefficients and thermal efficiency. The induced flow rate increases rapidly for small widths but tends to asymptote above a channel width of 25 mm. The peak centerline velocity and the average heat transfer coefficient occur at an optimal channel width of 10 mm above which the power dissipated is essentially invariant. The thermal effectiveness of the jet ejector induced cooling in the channel decreases with increasing channel width and this is manifest as an increase in the temperature difference between the wall and the exit air. The Nusselt number for synthetic jet ejector flows is six to eight times higher than for comparable conventional turbulent flow. Thus the jet ejector yields higher heat transfer at low flow rates. An active heat sink for high power microprocessors is developed using the ejector concept. The device enables a dissipation of about 110 W at 100 °C resulting in a case to ambient thermal resistance of 0.69 °C/W (using Al 6061) that is 350% higher than natural convection. In a direct comparison with a fan driven flow through the same heat sink, an ejector module having a flow rate within the range 3–5 CFM dissipates 40% more power than the fan at the same flow rate.

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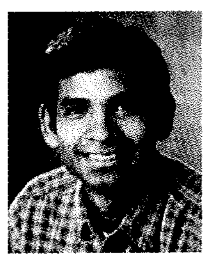

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