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# Enhanced Cooling of Electronic Components by Flow Oscillation

Kaveh Azar*
*AT&T Bell Laboratories, North Andover, Massachusetts 01845*

**An experimental investigation was conducted to examine the effect of forced oscillation of the fluid (air) entering an electronic circuit pack channel on component cooling. A realistic air-cooled channel made of two vertically mounted circuit packs containing nine components each was setup. Each component was individually powered and its temperature monitored. To induce controlled oscillation of the incoming air, a low-aspect ratio blade attached to a mechanical shaker was placed at the inlet of the channel. Several parameters including component power dissipation, air velocity (natural to moderate forced convection $Re_D$ = 3000), blade angle, and channel width were varied. The component-flow exposure was enhanced by setting the blade frequency to the frequency of the free-shear layer present in the groove formed by the components. This resulted in significant cooling of the components. The results showed that for blade frequencies varying from 1 to 3 Hz up to 25% cooling can be attained in electronic components.**

## I. Introduction

ENHANCEMENTS to the cooling of electronic components remains an active area of investigation. The issue of enhancement becomes even more attractive when air is used as the cooling fluid. Several schemes such as turbulators or vortex generators have been examined as methods for cooling enhancements. Oscillation of incoming fluid at the entrance of the circuit pack is another method for improving cooling capability without making major architectural changes to the system.

In this article, the result of an experimental investigation of forced oscillation of the incoming fluid on enhanced cooling of electronic components is reported. The study looks at a realistic setup in which the components and channel resemble an actual system. Observed departures from steady-flow cooling measurements were the effects of alteration of flow structure and component-flow exposure as the result of induced oscillation. The term "flow exposure" refers to the time when the hotter fluid engulfing a component is removed as the result of alteration to the flow structure.

The study first focused on showing that forced oscillation of flow can decrease component temperature. Then, several parameters affecting performance were explored and their contributions reported here. The method, herein called oscillatory cooling, showed that component temperature can decrease up to 25% for low frequencies of oscillation in forced convection. It was also observed that the enhanced cooling appeared to be nonlocalized, and the entire channel can benefit from forced oscillation. Natural convection results showed the opposite effect in which component temperature increased as the result of induced oscillation.

The concept of flow oscillation and hydrodynamic stability associated with this phenomenon is briefly discussed in the following section. The experimental setup assembled to conduct the study is described in Sec. III. The results and the significance of parameters that could influence the cooling mechanism are discussed in Sec. IV. This article is concluded by highlighting the areas that require further investigation to make oscillatory cooling a more readily adaptable technique in various system configurations.

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*Member of Technical Staff, 1600 Osgood St.

## II. Flow Phenomenon

Cooling of electronic components with either liquid or gas creates a unique and complex problem of flow over grooves. The periodic flow geometry (due to the spacing between electronic components on the printed wiring board) in turn creates a flow which is oscillatory in nature (Fig. 1). As shown in Fig. 1, the groove is formed by two components in the streamwise direction and the circuit board. The groove, which is three-dimensional, is open on three sides and is surrounded by a fluid at higher velocity. This is not a typical groove flow that has traditionally been considered in other cases.[1,2]

The flow structure in the groove is dominated by the two standing free-shear layers. The slow rate of interaction (exchange) of these vortices with the surrounding fluid is caused by the oscillatory nature of the flow. The fluid forming the shear layers is in close contact with the board and is partially originated from the stagnation point of the streamwise block of components. Since the fluid is in close contact with the board and the board temperature is significantly above ambient, the fluid inside the groove tends to be at a higher temperature.

The presence of a slowly oscillating free-shear layer in the groove is also disadvantageous from the heat transfer point of view. Since the groove flow can effectively be stagnant, a high-flow impedance area is created which prevents the cooler fluid in the midstream from reaching the component (Fig. 2). Therefore, the presence of the groove flow and its slow rate

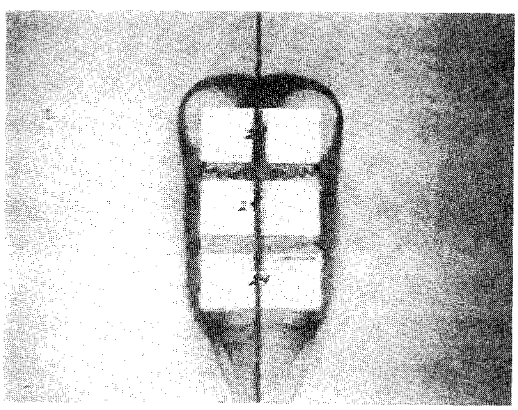

**Fig. 1 Flow visualization around a block of components, at $Re_D$ = 2250 showing the flow inside the grooves.**

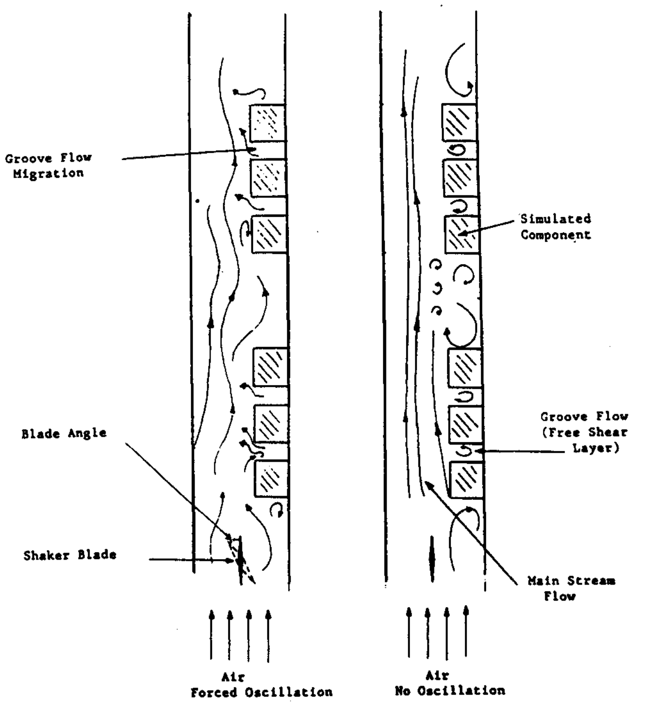


**Fig. 2 Schematic drawing of the flow patterns with and without oscillation for the experimental channel.**

of interaction with the mainstream increases component temperature. However, by forcing alteration in the groove-flow structure (i.e., removal of the stagnant air and enhancing the component-flow exposure), its temperature may significantly decrease.

The periodic nature of the groove flow and its frequency of oscillation are the elements that link it dynamically to the mainstream flow. The natural frequency of oscillation of the groove is governed by many parameters such as channel inlet velocity, component spacing and geometry, and channel height. In addition to these parameters, groove-wall roughness will also impact the frequency and lends itself to spatially varying instabilities. Because of the nature of the groove-flow oscillations, the hydrodynamic instabilities created by these oscillations are propagated into the mainstream, which may cause a self-sustained oscillation of the boundary-layer flow. This results in a direct interaction between the groove and the mainstream flow that may be used to enhance flow exposure of a given array of components.

To induce disturbance or migration of the free-shear layer of the groove, its frequency of oscillation has to match the one for the mainstream flow.[3] This will cause significant mixing at the component level which increases the heat transfer coefficient. The enhanced mixing may also create secondary flows which assist in further reduction of component temperature. Nevertheless, to create such mixing, one must know the frequency of oscillation of the free-shear layer.

Conversely, the induced mixing can create complex flow structures and increase the turbulence level of the channel flow. These effects (which may be more pronounced with large aspect ratio components) may potentially be detrimental from a heat transfer point of view. These regions are usually in the wake of large protrusions (components) with significant flow reversals.[4]

The flow phenomenon described above is quite complex for analytical or computational treatment. However, in a periodic-groove geometry and parallel-flow condition the unstable modes, thereby, groove frequency, can be determined from linear stability theory (Orr-Summerfeld equations[5]). But in other geometries, especially if the groove length in the streamwise direction is long, the flow dynamics are significantly different and large disturbances are present. In that case, the groove frequency would have to be determined by nonlinear stability theory.[6] Further complication in the flow structure, such as pulsation, may also occur at high Reynolds numbers, $Re_D > 1200$, that will make the analytical approach even more difficult. No commercially available computational fluid dynamics (CFD) software exists that can handle problems of this magnitude—especially for Reynolds numbers that characterize transient and pulsating flows. Therefore, an experimental approach is the only alternative for addressing the feasibility of oscillatory cooling and its application to geometries such as channels created by circuit packs.

## III. Experimental Setup

Several experiments were performed to study the concept of oscillatory cooling in an electronic enclosure and to determine its effectiveness in reducing component temperature. The experimental setup is depicted in Fig. 3a and consists of the following: 1) a channel made of two glass-epoxy boards, each containing nine simulated components, 2) minimechanical shaker, 3) function generator, 4) power amplifier, 5) au-

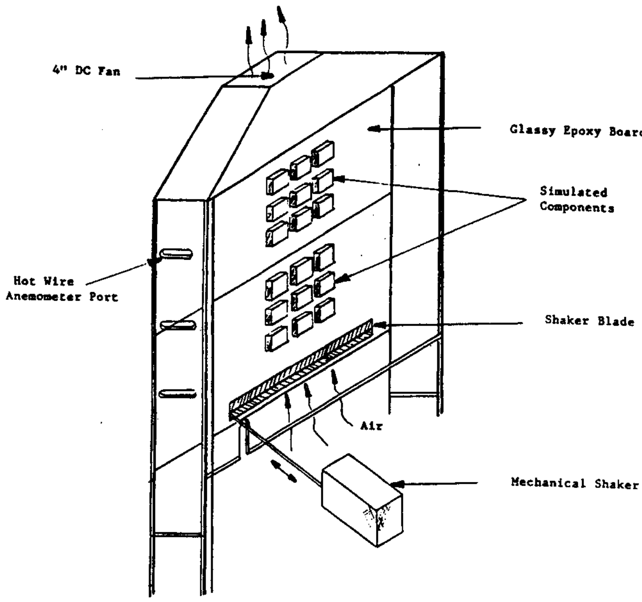


**Fig. 3a Wind-tunnel test setup for the oscillatory cooling experiment.**

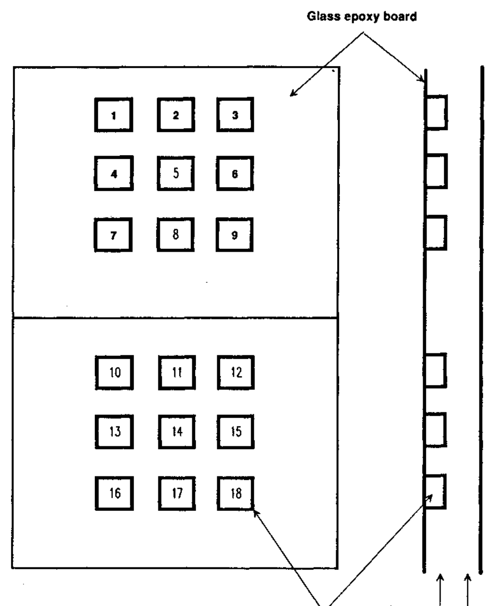


**Fig. 3b Component layout and numbering scheme for the oscillatory cooling experiment.**

tomated data acquisition/control unit, 6) smoke generator, 7) Strobotac and laser illuminator, and 8) hot-wire anemometer.

### A. Test Section

A channel (35.6-cm long) was set up to simulate an electronic enclosure (Fig. 3a). It consisted of two glass epoxy boards (33 × 17.8 cm) that were mounted on top of each other simulating an electronic channel. To study the impact of channel depth (wall-to-wall spacing), the tunnel was designed to be adjustable in depth. Also, the side walls and the front panel were made of clear Plexiglas® to observe the flow structure when using smoke for flow visualization.

The forced flow oscillation was done by installing a minishaker and a blade assembly at a distance half way between the first row of components and the inlet of the channel (Fig. 3a). To minimize the flow disturbance that can be generated by the presence of the blade, it was designed in a flattened diamond shape with a low profile having a thickness to height ratio of 0.2. Furthermore, to generate a uniform oscillation in the flowfield, the blade length, perpendicular to flow, was 17.8 cm extending well beyond the first and last columns of components. The blade was positioned so that its height was parallel with the flow (Fig. 2).

This particular configuration was suitable for uniform flowfield oscillations at low-shaker frequencies. There were some nonuniformities observed at higher frequencies. These nonuniformities were attributed to the cantilever beam design of the blade. One end of the blade was rigidly supported to the shaker arm while the other end was free. This could be one of the reasons for nonsymmetric cooling of the channel at higher frequencies.

### B. Component Design

To simulate an electronic component, 18 aluminum blocks each 1.9 × 1.9 × 0.64 cm with a small cavity 1.9 × 1.27 × 0.25 cm in the bottom were prepared. The cavity housed a thick-film resistor that provided the heat source in the simulated component and also provided passage for the fluid as seen in an actual component. Each resistor was epoxied to the aluminum block with a thermally conductive epoxy. Also, to simulate the thermal performance of the component and provide an adequate heat conduction path to the board, thermally conductive grease was applied at the board and the aluminum block interface. Components were uniformly laid out on the boards with 0.64-cm spacing in the stream and spanwise directions.

The component temperature was measured by a thermocouple epoxied to its surface. The thermocouples were flush-mounted to the surface of the component, and neither the junction nor the wire protruded into the flowfield. To eliminate any effect that may have been caused by nonuniform power distribution on board, each component was powered equally. The power was supplied by means of a power board to ensure uniformity and constancy in power dissipation.

### C. Experimental Procedure

The objective of the experiment was to observe the effect of the mainstream oscillation on the cooling of the components. This is achieved by attempting to match the frequency of the groove flow with that of the mainstream. The mechanical shaker forced the incoming air to oscillate at a prescribed frequency. The frequency of oscillation was set by the experimenter by way of the function generator. Then, by observing selected groove temperatures, component temperature, and airflow patterns, it was determined whether free-shear layer has migrated out of the groove. Smoke-flow visualization was performed on several occasions to gain some insight into the mixing pattern at the component level and to determine the potential range of frequencies for the shaker (oscillator).

The groove frequency can be obtained by using smoke-flow visualization and a strobe light or by direct observation of the groove temperature fluctuations. The groove flow, made visible by smoke, will appear stagnant when its frequency is matched by the frequency of the strobe light, therefore, yielding the desired frequency of oscillation. Another effective method for determination of the frequency is monitoring the groove temperature by a thin thermocouple and a digital thermometer (cold-wire anemometer). By inducing oscillation in the mainstream and observing the groove temperature, one can determine whether the groove flow has indeed been altered. The groove temperature, once reaching steady state, will decrease if the groove-flow frequency matches the forced frequency of the main stream.

Since there were 18 components in the experimental channel, the grooves associated with the hottest components under no oscillation were chosen to be monitored. Smoke-flow visualization revealed a frequency range of 0.5–12.5 Hz at several channel inlet air velocities. For frequencies above 12.5 Hz, some fanning by the blade was observed. The frequency of oscillation of the mechanical shaker was varied from 0.5 to 12.5 Hz with increments of 0.5 Hz. In this range, there was no noise generated by the shaker. Furthermore, to ensure that no fanning was present as a result of the forced oscillation, velocity measurements were performed inside the channel by a hot-wire anemometer.

### D. Data Collection Process

In this experiment, inlet air velocity, channel height, frequency of oscillation of the shaker arm and angle of the blade were varied. All component temperatures and the three groove temperatures were observed for inlet air velocities. Once the steady-state temperature was reached (a time constant of approximately 8 min for forced convection and 45 min for natural convection), the shaker was set into motion. This implied that the data at $\tau = 0$ corresponded to steady-state temperatures with no oscillation. The frequency of the shaker was increased by 0.5 Hz, while monitoring selected groove temperatures. If no appreciable fluctuations over $\Delta T < 3°C$ were observed and the temperatures reflected a steady-state condition (usually a time constant of approximately 4 min), the frequency of the shaker was increased by 0.5 Hz. This process continued until the frequency range determined by smoke-flow visualization was scanned. The shaker frequency was changed approximately every 10 min, allowing two and half times the system time constant for it to reach steady state. The data was collected at 2-min intervals for every change of frequency.

## IV. Results

The experimental study conducted had two objectives: 1) to verify that oscillatory cooling is a viable technique in a realistic circuit pack channel; and 2) to show the impact of some of the parameters that may affect the performance of this cooling process. The parameters considered in this study were as follows: 1) air velocity at the channel centerline; 2) channel depth; and 3) angle of shaker blade $\alpha$.

In this experiment, the velocity was varied from no-forced flow natural convection, to 162.5 cm/s with increments of 40.6 cm/s. Two channel depths ($H$ = 1.27 and 2.22 cm) and blade angles of 30 and 45 deg were considered. The basis for selection of these parameters was the practical configurations seen in electronic equipment.

The data in Figs. 4–10, depicts the nondimensional temperature

$$\theta = \frac{T_{w,o} - T_{\mathrm{amb}}}{T_{n,o} - T_{\mathrm{amb}}}$$

vs $\tau$ defined as

$$\tau = \omega \cdot t$$

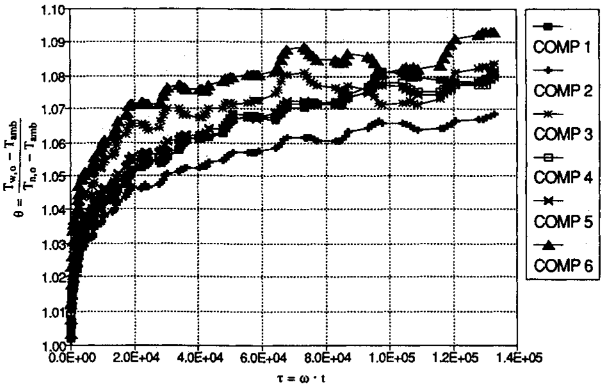


**Fig. 4a** $\theta$ **vs** $\tau$ **for components 1–6 in a natural convection mode** $H$ **= 1.27 cm.**

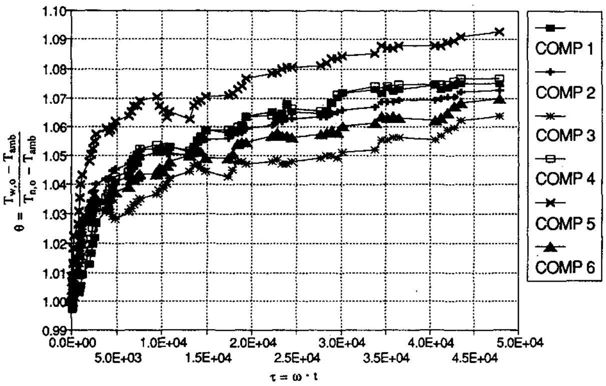


**Fig. 5a** $\theta$ **vs** $\tau$ **for components 1–6 in a natural convection mode** $H$ **= 2.22 cm.**

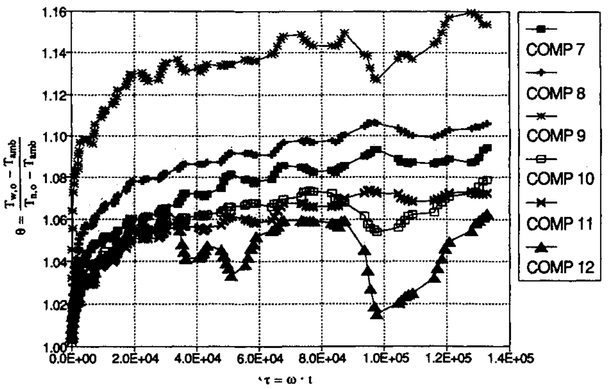


**Fig. 4b** $\theta$ **vs** $\tau$ **for components 7–12 in a natural convection mode** $H$ **= 1.27 cm.**

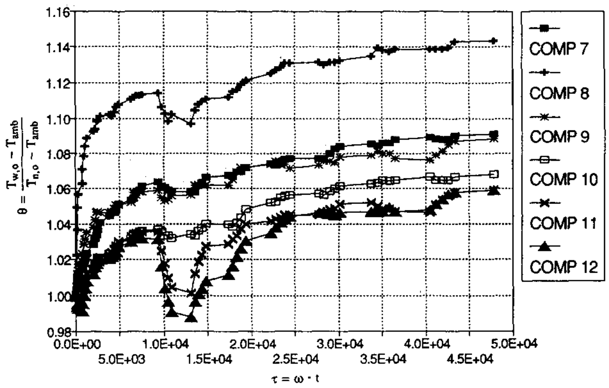


**Fig. 5b** $\theta$ **vs** $\tau$ **for components 7–12 in a natural convection mode** $H$ **= 2.22 cm.**

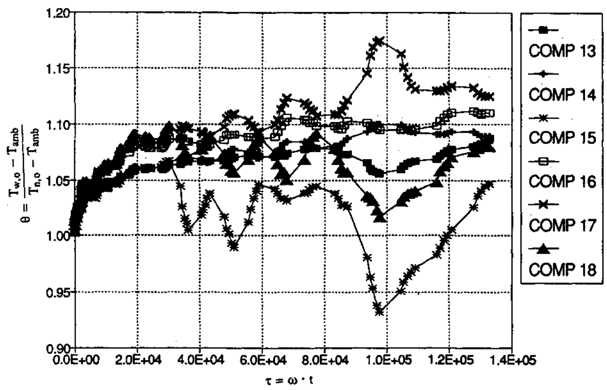


**Fig. 4c** $\theta$ **vs** $\tau$ **for components 13–18 in a natural convection mode** $H$ **= 1.27 cm.**

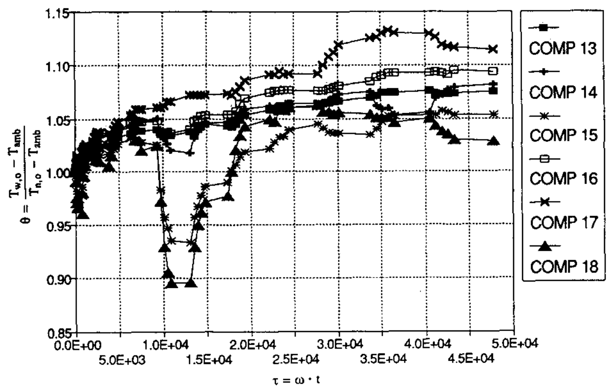


**Fig. 5c** $\Theta$ **vs** $\tau$ **for components 13–18 in a natural convection mode** $H$ **= 2.22 cm.**

for different tests and component numbering scheme shown in Fig. 3b. Where $T_{w,o}$ represents the component temperature with forced oscillation, $T_{n,o}$ is the component temperature with no oscillation, $T_{\text{amb}}$ is the ambient temperature measured at the channel inlet, $\omega$ is the frequency of the shaker, and $t$ is the time of duration of the shaker frequency.

The data were chosen to be presented in the $\theta$ vs $\tau$ domain instead of $\theta$ vs Strouhal or frequency number, $N_f$, since generally accepted characteristic length and velocity have not been established in electronic cooling. Also, $\tau$ was selected as the independent parameter since it is nondimensional and combines the changes in time and frequency of the oscillator. Therefore, direct comparison appeared to be the most suitable method for showing the process of oscillatory cooling.

Figures 4a–4c show the effect of oscillation for the channel in the absence of forced convection. The channel depth for this case was 1.27 cm and the blade was parallel with the flow. The data showed that the component temperature actually increased as the result of oscillation. The increase in temperature can be attributed to the partial reversal of flow in the channel when oscillation was introduced. These figures also show that for $H$ = 1.27 cm no fanning effect took place as the result of the oscillation in the range of frequencies examined.

To ensure that induced oscillation is independent of the channel depth, $H$ = 2.22 cm was also considered. The motivation behind this consideration was to verify that viscous dissipation at the channel walls did not cause the induced oscillation to damp out, especially at smaller channel depth. Figures 5a–c show similar results as in the $H$ = 1.27-cm case. The induced oscillation does not cause any additional cooling in the natural convection mode and appears to be independent of the channel depth.

The natural convection results (Figs. 4a–5c) clearly showed that no fanning occurred as the result of the oscillation at the inlet. These results corroborated the velocity measurement

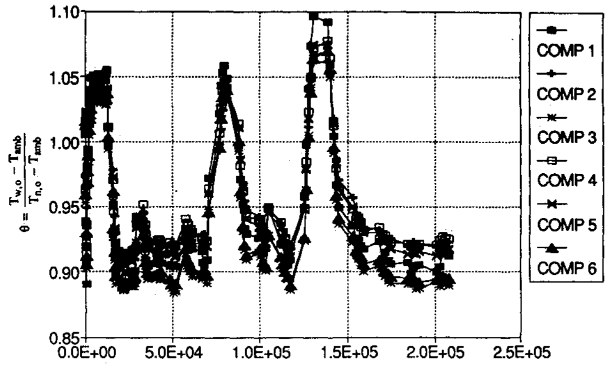


**Fig. 6a θ vs $\tau$ for components 1–6, $Re_D$ = 2712.2, $H$ = 2.22 cm, and $\omega$ 0.5–12.5.**

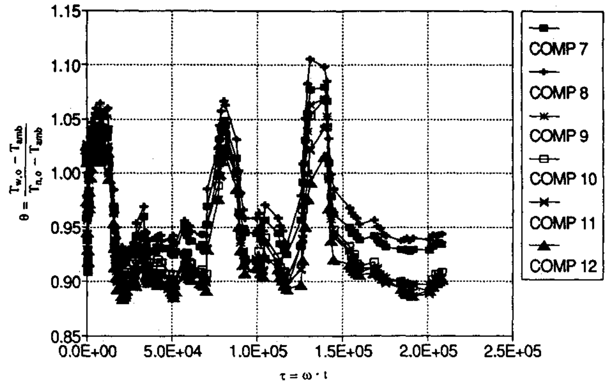


**Fig. 6b θ vs $\tau$ for components 7–12, $Re_D$ = 2712.2, $H$ = 2.22 cm, and $\omega$ = 0.5–12.5.**

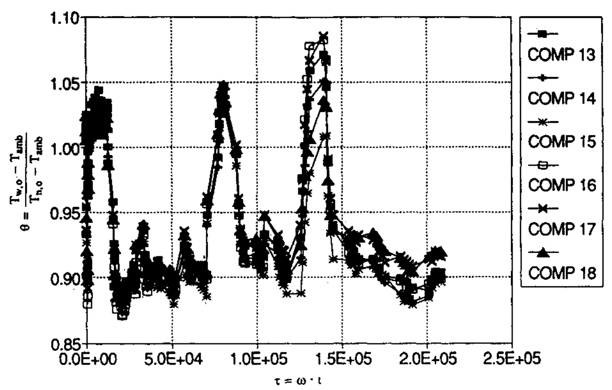


**Fig. 6c θ vs $\tau$ for components 13–18, $Re_D$ = 2712.2, $H$ = 2.22 cm, and $\omega$ = 0.5–12.5.**

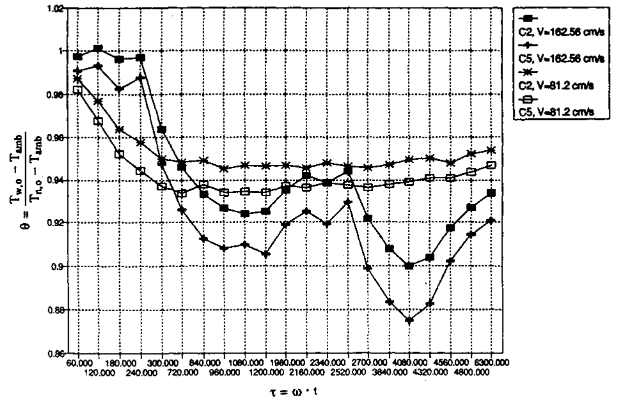


**Fig. 7a Effect of inlet velocity θ vs $\tau$ for components 2 and 5 when $Re_D$ = 2712.2 and 1356.2, and $H$ = 1.27 cm.**

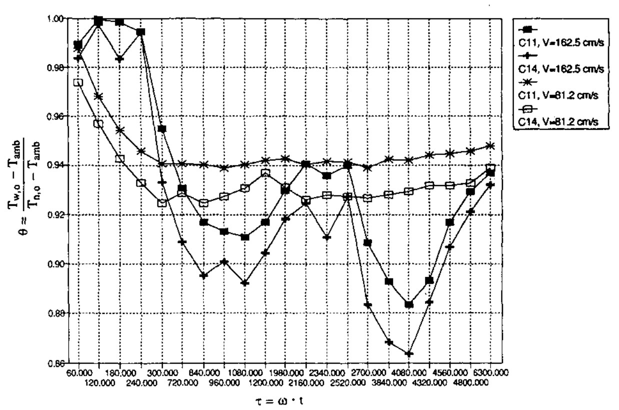


**Fig. 7b Effect of inlet velocity θ vs $\tau$ for components 11 and 14 when $Re_D$ = 2712.2 and 1356.2, and $H$ = 1.27 cm.**

data, even at higher frequencies, $\omega$ = 10–12.5 Hz. However, some components (12, 15, and 18, on Figs. 4b, 4c, and 5c) showed moderate cooling as the result of the oscillation. This perhaps is attributed to the forced convection behavior of the channel. Velocity measurement showed formation of viscous boundary layers on both walls of the channel. Because of this, the channel behaved like a forced convection—one which caused the formation of the free-shear layer in the groove associated with those components. It is likely that the groove's flow frequency matched that of the oscillator and resulted in further enhancement of the component flow exposure. However, the effect appeared to be localized and it cannot be used as an enhancement to cooling in the natural convection mode.

Figures 6a–c show the effect of inlet-flow oscillation on enhancing component cooling. The data show the case when the channel depth was 2.22 cm, the inlet air velocity of 162.5 cm/s, and a power dissipation of 1 W per component. These figures show the effect of oscillation for the full range of frequencies (0.5–12.5 Hz) examined. For this test condition, component temperature is reduced by approximately 15% and the magnitude of cooling is a function of component position on board. Also, increases of 10% are possible at certain frequencies.

Figures 6a–c also show an interesting trend in response to the forced oscillation. Although the magnitude of cooling varies, all components are responding to the oscillation with a similar trend on both boards. One possible explanation for this is that because the flow is three-dimensional, many periodic secondary flows are also formed. Some of the cooling (or the trend) can be caused from these secondary flows that respond favorably to the forced oscillation. In a spatially two-dimensional case when the groove walls occupy the entire width of the channel (see Ref. 3), secondary flows do not have a dominant presence. Therefore, a single oscillatory free-shear layer is more likely to occur than simultaneous coexistence of several flow structures. This implies that in a spatially two-dimensional case, one may potentially determine a single-shaker frequency to which the groove will respond.[3] In a three-dimensional situation, this may not be the case. However, the areas where more cooling occur are attributed to the strongest coupling between the groove shear-layer oscillation and the forced oscillation of the mainstream flow.

Figures 4a–6c clearly show that forced oscillation of air entering a channel will affect the thermal response of the components. What follows is the characterization of the parameters that were thought to be significant in performance of oscillatory cooling. Since showing the data for all components tends to be congested, only the results for the hottest components (2, 5, 11, and 14) are reported here. To further minimize crowding of the plots, only a small percentage of $\tau$ is shown.

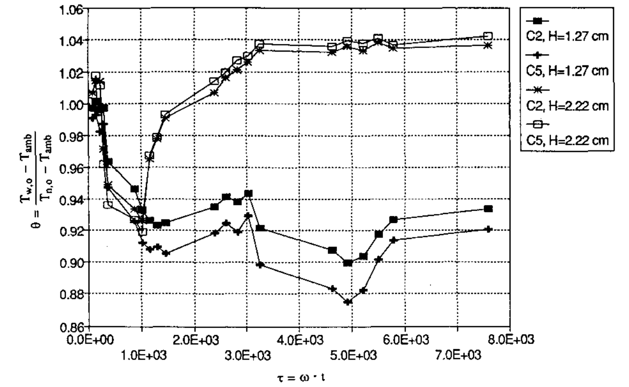


**Fig. 8a Effect of channel depth, $\theta$ vs $\tau$ for components 2 and 5 when $Re_D$ = 2712.2.**

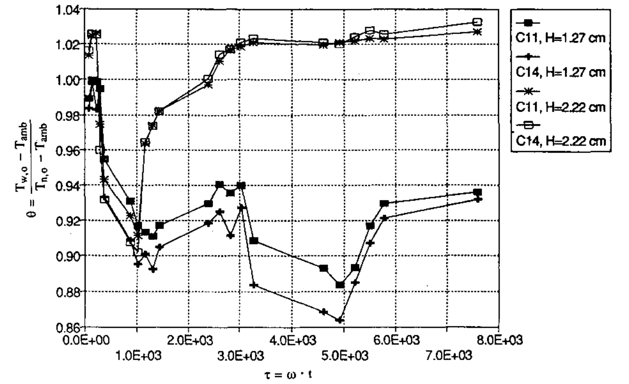


**Fig. 8b Effect of channel depth, $\theta$ vs $\tau$ for components 11 and 14 when $Re_D$ = 2712.2.**

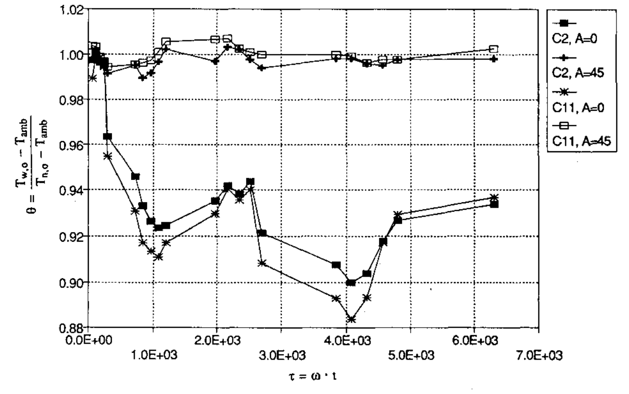


**Fig. 9 Effect of shaker blade angle, $\alpha$ = 0 and 45 deg, $\theta$ vs $\tau$ for components 2 and 11 when $Re_D$ = 2712.2 and $H$ = 1.27 cm.**

Figures 7a and 7b show the effect of inlet velocity on the cooling of components for the same range of $\tau$. These correspond to velocities 81.2 and 162.5 cm/s measured at the centerline of the channel inlet for $H$ = 1.27 cm. The corresponding $Re$ numbers, based on the channel hydraulic diameter, are 1356 and 2712, respectively. Although fully developed theory assumptions in most cases do not apply to electronic cooling its premises can be used for comparison. Consequently, the first $Re$ number shows a clear laminar flow region and the next is the turbulent flow or the onset of one.

These figures show the sensitivity of the component thermal response to velocity and oscillation frequency. At lower values of $\tau < 840$, both $Re_D$ conditions respond similarly to forced oscillation and result in approximately an 8% enhanced cooling. At higher $\tau$, however, $Re_D$ = 2712 appears to respond more favorably than the $Re_D$ = 1356. This can be expected since the turbulence intensity level is higher at the larger $Re_D$ and the flow tends to be more periodic in nature. In addition,

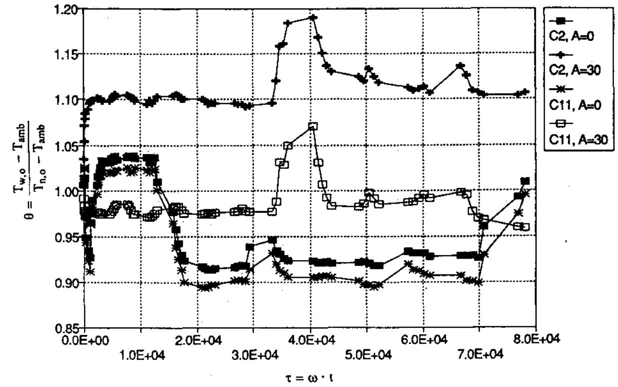


**Fig. 10 Effect of shaker blade angle, $\alpha$ = 0 and 30 deg, $\theta$ vs $\tau$ for components 2 and 11 when $Re_D$ = 2712.2 and $H$ = 2.22.**

although components 11 and 14 exhibit slightly higher cooling than components 2 and 5, the improvement is not confined to the board or to the component nearest the oscillator. The results also suggest that the induced oscillation is transported throughout the channel and minimal viscous damping occurs. In addition, altering the flow structure of the first board could further magnify or sustain the oscillatory nature of the mainstream flow. This is a major advantage for enhanced cooling of electronic systems since limited areas are available for inducing forced oscillation.

Figures 8a and 8b show the effect of channel depth $H$, on thermal response of the hottest components in the channel. The inlet velocity was maintained at 162.5 cm/s, and the channel depth was varied from $H$ = 1.27–2.22 cm. It is worth noting that in complex channel structures such as the ones made of electronic circuit packs, centerline velocity, and velocity in place of pressure drop, is the most experimentally repeatable quantity. Therefore, in this case, velocity is kept constant and the channel depth is changed to the values mentioned above.

These figures show that at low values of $\tau$, thermal response of the components with smaller depth tend to decrease rapidly. The thermal response appears to be independent of the channel depth until $\tau$ becomes larger than 1000. At this junction, the component thermal response becomes sensitive to $H$ and it shows a major departure from each other. Larger $H$ shows a slight increase in temperature even at higher values of $\tau$, whereas lower $H$ shows significant cooling at the same ranges of $\tau$. Therefore, impact on component cooling is both a function of channel depth and frequency of the oscillator.

The effects of shaker blade angle on performance of oscillatory cooling are shown in Figs. 9a, 9b, and 10 for $H$ = 1.27 and 2.22 cm and inlet air velocity of 162.5 cm/s. The blade angle is defined as the angle that the shaker blade makes with the flow. If the blade is parallel with the flow, $\alpha$ is equal to zero. The angles considered in this study were 30 and 45 deg. The motivation behind variation in $\alpha$ is to quantify the impact of the shape of the generated disturbance on oscillatory cooling. Figure 9 shows that $\theta$ of the hot components is adversely affected when $\alpha$ is changed from 0 to 45 deg. Except for very low values of $\tau$, changing the orientation of the blade causes no additional cooling in the channel.

Figure 10 compares the two hot components from each board for $\alpha$ = 0 and 30 deg, when $H$ = 2.22 cm and inlet air velocity is 162.5 cm/s. Comparison of these cases shows a clear increase in temperature of the hot components as the result of the change in the blade angle. Although the channel depth associated with this data is larger than the case shown in Fig. 9 and the blade angle is smaller, the thermal response of the components is unfavorable. This suggests that the best results are obtained, independent of channel depth, when the shaker blade is parallel with the direction of the flow or less than 30 deg. Nevertheless, the results show that the blade angle can significantly affect the performance of oscillatory cooling, and it suggests further investigation to optimize the blade angle.

## V. Summary

The results of the investigation reported here clearly indicate that forced oscillation at the channel inlet can be utilized to enhance component cooling. It has been shown that the concept of oscillatory cooling is a useful method for channels made of several vertically mounted circuit packs. No significant damping of the forced oscillation was observed as the result of the first circuit pack. The imapct of oscillation is either transported throughout the channel or magnified by the first rows of components. It was shown that the channel depth is a critical parameter in the performance of oscillatory cooling and points toward an optimum geometrical configuration for maximum cooling.

The result of this study suggests that there exists a critical-groove frequency where maximum cooling occurs. Therefore, an analytical method is required to determine $\omega$ as a function of system configuration. Also, further work is required to understand the flow dynamics in the channel since the thermal response of the components appears to be harmonious.

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