Piezoelectric translational agitation for enhancing forced-convection channel-flow heat transfer

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ABSTRACT

Piezoelectric translational agitator (PTA) is proposed and its heat transfer performance is demonstrated by experiments in a narrow channel. For the present study, this channel simulates a portion of an air-cooled heat sink. An oval loop shell structure successfully generates millimeter-range translational displacement to a blade attached to the shell. A PTA operating at 961 Hz as its second resonance mode with a 1.4 mm displacement under 60 Liters Per Minute (LPM) of cross flow achieved about 55% improvement in heat transfer coefficient compared to the non-agitated state. Pressure drop in the test section and the corresponding power required to drive the flow through the channel against the oscillating blade were measured. In addition, the PTAs were tested in the open channel without through-flow to examine the sole effect of agitation and to investigate the possibility of using it as a stand-alone cooling device. A total Reynolds number was defined to characterize the combined effects of cross flow and agitation. The Stanton number developed from the relationship between the total Reynolds number and heat transfer coefficients enables predicting operating performance.

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1. Introduction

High demand for fast processing speed and powerful computational capability of electronics has yielded remarkable development of integrated circuits. The integrated circuit, a vital component of almost all the electronics, consists of numerous tiny transistors on a very small area of substrate. The rapid development of micro and nanoscale fabrication technologies enabled the evolution of the integrated circuit. This progress is even surpassing the rate of the Moore's law, which describes the trend of electronic device capability. However, advanced integrated circuits generate enormous thermal dissipation that potentially degrades the functionality of electronic devices. As a result, there is a strong need for effective electronic cooling systems. A variety of cooling techniques such as air cooling, liquid cooling, direct sprays, boiling heat transfer, etc., have been developed to remove heat from electronic components. The liquid cooling methods, including the direct spray and boiling methods, can provide fairly large cooling capability compared to air cooling [1–5]. However, they have a potential reliability problem regarding leakage and require complex components such as pumping [6,7], spraying, and nozzle systems [8,9]. These make them unattractive for application to electronics cooling. Therefore, air cooling remains the most general and popular method for cooling electronics due to its reliability and environmentally favorable characteristics. As a passive approach for air cooling, natural convection on different types of heat sinks was widely investigated experimentally and numerically. Most focused on design optimization of heat sinks with rectangular fins [10–13], radial fins [14], and circular pin fins [15]. Harahap and Setio [16] proposed several correlations based upon experimental studies on variously-arranged finned arrays. However, the cooling capability of natural convective heat sink is far from the desired performance for modern power electronics due to the intrinsic limitations of natural convection. To enhance performance, a fan or a blower, which can generate strong air flow through finned arrays, is adopted, together with optimized heat sinks. This scheme has become the most widely adopted method for electronic cooling. Researchers have recently focused on developing and optimizing different kinds of heat sinks with forced convection [17–19]. In spite of the addition of forced convection with a fan or blower, thermal management through air cooling continues to require more powerful cooling capability as integrated circuit fabrication technologies continue to evolve.

As a result, more active cooling components that can enhance, or replace, the fan or blower are presently of interests. Piezoelectricity, electro-mechanical charge stored in non-symmetric crystals or ceramic materials, is central to the present development. Piezoelectric materials physically deform when the applied electric field changes. These materials can possibly reduce reliance on
fan or blower power while improving reliability of the systems. Recently, piezoelectric flappers or fans attracted much attention as a thermal management solution since Toda and Osaka [20] proposed them as cooling or ventilation devices. The piezoelectric fan consists of either bimorph or monomorph piezoelectric ceramic patches attached to highly elastic metal or plastic thin layers. When contracting and expanding piezoelectric patches caused by alternating voltage generate flapping movement of the elastic shim at the natural frequency of the system, strong air currents take place that can be utilized for cooling. Yoo et al. [21] performed experimental and theoretical studies with different types of piezoelectric fans made with three different shim materials; phosphor bronze, brass, and aluminum. Their fans generated the largest displacement of 34.7 mm with an operating frequencies of between 10 Hz and 60 Hz. They also investigated the relationship between the tip displacement and wind velocity that can be used in the optimization process. Acikalin et al. [22] performed experimental studies for determining optimum positions of the fan against a heat sink at four different positions as well as flow visualization studies of the piezoelectric fan operating at 20 Hz with a 1.5 cm tip deflection at an applied voltage of 40 V to better understand their performance for heat transfer augmentation. In their report, feasibility of cooling a laptop was investigated with three different fan configurations of wide, narrow transverse, and transverse shapes. Wait et al. [23] studied both numerically and experimentally, with flow visualization, the performance of piezoelectric fans operating at higher resonance modes. They evaluated and compared the performance at different resonance modes with electromechanical coupling factors that show the effectiveness of converting electrical energy to mechanical energy. Kimber et al. [24] investigated localized cooling capability of piezoelectric fans positioned closed to an electrically heated stainless steel foil. Temperatures were observed by an infrared camera. Their piezoelectric fan operated at four different vibrational amplitudes; 6.35 mm, 7.5 mm, 8.5 mm, and 10 mm, with three different distances from the heated steel foil. They proposed several correlations for predicting thermal performance of piezoelectric fans which used specifically-defined Reynolds numbers and Nusselt numbers. Additional characterization and optimization processes of piezoelectric fans in terms of tip amplitude, fan geometry, operating frequency, and fan arrangement were reported in the literature [25,26]. For the both reports, the operating frequencies of the piezoelectric fans were either around 100 Hz or less than 60 Hz. Petroski et al. [27] proposed optimized heat sink designs specially adapted to the oscillating piezoelectric fans. The combined system of optimized heat sink and embedded piezoelectric fan achieved a thermal resistance of 1 °C/W.

As reviewed at the previous sections, piezoelectric flapping fans were developed and investigated by several research groups as an active cooling device for electronic components and their many advantages were displayed. However, there are some weak points. First, piezoelectric fans have relatively low operating frequencies of around 100 Hz or less that can limit cooling capacity, though attractive from noise and power consumption view points. Second, the maximized cooling effect from the fan tip is restricted in a local area [24]. Thus, application to larger heated areas is difficult. Third, previous studies covered piezoelectric fans as only a stand-alone cooling devices. Therefore, in most of cases, cooling performance of piezoelectric fans was compared to natural convection cases. There is no known report that shows enhancement by flapping fans in forced convection systems with fans or blowers. In fact, it is unrealistic to put piezoelectric fans in heated coolant flow as thermal interaction between heated coolant and piezoelectric ceramics possibly degrades piezoelectric fans.

A synthetic jet is an alternative method for electronics cooling. Extensive studies about hydrodynamic characteristics of the synthetic jet have been performed [28,29]. Moreover, significant enhancement of thermal performance by a synthetic jet from a single orifice was achieved for various applications, such as a micro channel [30] and a flat surface [31,32]. A multiple-orifice synthetic jet cooler with a radial heat sink was proposed and applied to a 3D stacked chip cooler [33,34]. None of the above was considered for addition to the fan-cooled heat sink system.

In the present study, we investigate a high-frequency translational agitation cooling mechanism driven by piezoelectricity that eliminates many of the inherent weaknesses of the piezoelectric flapping fan or the synthetic jet. The piezoelectric translational agitator developed here normally operates at relatively higher frequencies, above 500 Hz up to more than 1 kHz, with sufficiently large amplitudes. Due to the translational movement of a blade, the heated area that falls within the domain of a blade can be provided with consistent cooling from agitator-driven air. In order to
investigate cooling performance of the piezoelectric translational agitator either as a stand-alone device or a supplemental device to the fan, heat transfer experiments were performed with the agitator inside a channel that simulates a heat sink channel. For comparison, tests were run also with the agitator and with fan assistance or with no fan assistance.

2. Piezoelectric translational agitator

2.1. Working principles

An oval loop shell structure and a PZT stack actuator were utilized in the current study to achieve high-frequency translational movement with oscillating amplitudes sufficiently large to provide cooling capability to the heat source. Yeom et al. [35] performed detailed studies on the vibrational characteristics of the oval loop shell amplifier. A schematic of the piezoelectric translational agitator (PTA) that consists of the PZT stack actuator, the oval loop shell, and a carbon fiber blade structure is shown in Fig. 1. The PZT stack actuator normally generates very small translational displacement of less than a micron, which is around 0.1% of the strain range of its own actuator. The oval loop shell structure significantly amplifies small displacements of the PZT stack actuator using resonance energy of the structure. The PZT stack actuator generates displacement of “**da**” (Fig. 1) in the horizontal direction and the oval loop shell amplifies the movement in the vertical direction. When the operating frequency of the PZT stack actuator approaches one of the natural frequencies of the oval loop shell structure, highly amplified translational movement (“**da**”, arrow in Fig. 1) is generated in the vertical direction and transferred to the blade structure attached to the oval loop shell, providing strong air current that can be used for cooling purposes. At resonance, either the upper or the lower beam of the shell is excited vigorously. There are two appropriate resonance modes of the PTA within a 2 kHz frequency range that can provide amplified translational motion. For the first resonance mode, the bottom beam of the shell is mainly excited, moving the entire agitator body up and down. For the second resonance mode, only the upper beam is excited conveying translational movement to the blade. The PTA operating at the two resonance modes is shown in Figs. 2(a) and (b) show the PTA operating at the first resonance mode. In Figs. 2(b), it is confirmed that the lower beam is flapping and the movement is seen as a blur formed along the entire agitator body, except for the center area of the lower beam. Figs. 2(c) and (d) show the PTA operating in the second resonance mode. In Fig. 2(d), it is clearly seen that the upper beam is strongly excited, providing large translational motion to the blade while the remainder of the agitator is stable. Three design parameters, “**a**”, “**b**”, and “**t**” shown in Fig. 1 characterize the operating performance of the shell. The oval loop shell and connector pieces were cut from 5160 spring steel by the wire EDM (electrical discharge machining) process. A shot peening process was added to increase fatigue strength of the shell structure. A carbon fiber composite plate was used for the blade structure (Fig. 3) due to its very high stiffness-to-weight ratio compared to metals. The weight of the carbon fiber blade structure is about 2 g. The blade mounted at the top of the oval loop shell produces strong air currents adjacent to the heated surface by generating high-frequency translational oscillation. Four different PTAs were designed and fabricated for heat transfer tests. The design parameters and dimensions of the PTAs are summarized in Table 1.

2.2. Vibration characteristics of the piezoelectric translational agitator

Vibration characteristics of the PTAs were investigated with the PSV-400 laser Doppler vibrometer from Polytec. A schematic of the vibration test arrangement is illustrated in Fig. 3. A mechanical vice is used to clamp the PTA on the optical table and a vertically-mounted laser vibrometer scanning head is shooting a laser on the surface of the blade. In order to account for air damping effects of the actuating agitator when it is inside the channel, measurements were taken inside a transparent channel made of clear plastic plates, shown in Fig. 3. The VF-500 linear piezoelectric voltage amplifier (Dynamic Structures & Materials, LLC) was used to provide high voltage to the PTAs. Alternative current signals were generated by the AFG3102 function generator from Tektronix, Inc. As shown in Fig. 4, mean-to-peak displacements per unit of applied voltage of the PTAs were measured in the frequency domain over the range 0 Hz to 2 kHz. For each PTA, two resonance peaks were found in the frequency range. The first mode, which peaks at relatively lower frequencies of less than 500 Hz, exhibits smaller displacements than those of the second mode peaks. The second mode peaks appear at higher frequencies of above 500 Hz and up to 1.1 kHz. In addition, the second mode provides a much larger displacement than the first mode. The resonance frequencies of each PTA are listed in Table 2. Generally, the oval loop shell becomes stiffer as the design parameter “**a**” decreases and the thickness “**t**” increases. Therefore, the PTA-4 actuator has the largest resonance frequencies, but produces the smallest displacement with the same applied voltage. On the other hand, the PTA-1 actuator has the smallest resonance frequencies and the largest displacements among the four PTAs. As the second mode operation provides larger displacement and higher frequency than the first mode, the second mode is regarded as the more appropriate for this cooling application. For effective comparisons of heat transfer test results using different PTAs, seven displacements were selected and the corresponding voltages for the various PTAs were measured for the second mode so that the displacement is consistent for each heat transfer test. However, the intrinsic capability of each PTA toward generating dynamic flexural displacement is different due to different stiffnesses of the oval loop shell structures. Therefore, the largest displacements used for the heat transfer test-

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**Fig. 1.** Schematic of piezoelectric translational agitator.
The displacements of each PTA used for heat transfer testing and their corresponding applied voltages are listed in Table 3. Peak velocities of the PTAs for each displacement value were also measured, as shown in Fig. 5. Overall, the peak velocities increase almost linearly with increasing displacement. For all the cases, the peak velocities range from 0.7 m/s to 4.6 m/s within the ranges of displacements and frequencies. The largest velocity is from PTA-3, with a displacement of 1.6 mm.
Table 2
Resonance frequencies of PTAs.

<table>
<thead>
<tr>
<th>Resonance frequency (Hz)</th>
<th>The first mode</th>
<th>The second mode</th>
</tr>
</thead>
<tbody>
<tr>
<td>PTA-1</td>
<td>178</td>
<td>596</td>
</tr>
<tr>
<td>PTA-2</td>
<td>203</td>
<td>673</td>
</tr>
<tr>
<td>PTA-3</td>
<td>256</td>
<td>921</td>
</tr>
<tr>
<td>PTA-4</td>
<td>316</td>
<td>1080</td>
</tr>
</tbody>
</table>

Table 3
Dynamic displacements of PTAs for heat transfer testing and their corresponding applied voltages.

<table>
<thead>
<tr>
<th>Displacement (mm)</th>
<th>0.4</th>
<th>0.6</th>
<th>0.8</th>
<th>1.0</th>
<th>1.2</th>
<th>1.4</th>
<th>1.8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Voltage (V)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>PTA-1</td>
<td>11.6</td>
<td>22.0</td>
<td>35.0</td>
<td>49.0</td>
<td>64.0</td>
<td>80.0</td>
<td>126.0</td>
</tr>
<tr>
<td>PTA-2</td>
<td>16.0</td>
<td>30.0</td>
<td>50.0</td>
<td>68.0</td>
<td>90.0</td>
<td>116.0</td>
<td>180.0</td>
</tr>
<tr>
<td>PTA-3</td>
<td>25.8</td>
<td>48.0</td>
<td>72.0</td>
<td>98.0</td>
<td>122.0</td>
<td>152.0</td>
<td></td>
</tr>
<tr>
<td>PTA-4</td>
<td>45.0</td>
<td>86.0</td>
<td>188.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 4
A-weighted acoustic level of PTA-1 and PTA-3 at different displacements.

<table>
<thead>
<tr>
<th>Displacement (mm)</th>
<th>0.4</th>
<th>0.6</th>
<th>0.8</th>
<th>1.0</th>
<th>1.2</th>
<th>1.4</th>
<th>1.8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Noise level (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>PTA-1</td>
<td>67.5</td>
<td>72.3</td>
<td>75</td>
<td>78.3</td>
<td>80.3</td>
<td>81.8</td>
<td>84.4</td>
</tr>
<tr>
<td>PTA-3</td>
<td>85.3</td>
<td>87.1</td>
<td>91</td>
<td>94.5</td>
<td>97.3</td>
<td>99</td>
<td></td>
</tr>
</tbody>
</table>

2.3. Noise level

Noise of a cooling device could be an important issue in many engineering applications, including electronics cooling. Therefore, a brief discussion about the noise aspect of the PTAs is provided in this section. The PTA creates high-frequency and large-displacement oscillating motion to the beam elements of the device; thereby, it generates strong acoustic pressure waves to the ambient air resulting in substantial noise. A 12.5 mm (1/2 inch) Free-field microphone (Type 40AE) from GRAS Sound and Vibration was used to measure the noise. The microphone was placed at one meter away from the PTAs. The A-weighted noise levels for each of PTA-1 and PTA-3 at different displacements are listed in Table 4. The background noise level is around 46 dB, which is in the range of typical noise level in an office environment. The PTA-3 actuator, operating at 921 Hz, generates about 20% larger noise level in dB than the PTA-1 actuator with an operating frequency of 596 Hz for cases of various displacements. The noise level also tends to increase with increasing displacement at a fixed frequency. If the PTA operation raises a noise problem, solutions, such as using a muffler [36,37], can be found. It is found that a muffler used with PTA actuators similar to those discussed above could lower the noise level 40 dB.

2.4. Scaled-up system

The present investigation of the PTA with a single blade focuses on detailed studies in a single channel over a variety of operating conditions. This allows assessing operating conditions for the PTA to be used in the proposed cooling scheme, a multiple-channel heat sink system. Generally, a heat sink for electronics cooling would possess numerous fins and channels to maximize convective heat transfer area. The complexity of such system raises challenges to making it active. However, the proposed heat sink concept which utilizes the present research employs a coarser fin structure and capitalizes on much more effective heat transfer, as induced by agitation. The single blade PTAs tested herein are being extended to a multiple-blade system driven by one actuator. This is done by adding more blade plates to the frame, which can be done while adding little mass to the drive system, allowing high-frequency and large-displacement dynamic operation. An 11 g prototype of a multiple-blade frame made by a carbon fiber composite is shown in Fig. 6. The operating frequency and peak-to-peak displacement of this system are 520 Hz and 1.4 mm when operating with 200 V. Initial work on this system and corresponding results are presented by Yu et al. [38].

3. Heat transfer experiments

3.1. Experimental set up

Heat transfer experiments were conducted at various operating conditions for four PTAs to demonstrate the cooling performance of the PTA. The heat transfer facility is illustrated in Fig. 7. The test section simulates a portion of a heat sink channel with a heated surface on one side. A long, rectangular copper block constitutes the convection surface. A cartridge heater was inserted at the end of the copper block and the energy conducts from the heater to the exposed surface inside the channel. The copper block is surrounded by thick Styrofoam insulation blocks to ensure that nearly all the energy from the heater is conducted to the channel wall and is available for convection. A narrow and long channel is formed along the exposed surface of the copper block using transparent plastic plates. The PTA plate oscillates adjacent to the exposed surface. A blower generates flow through the channel. A pressure tap at the outlet of the channel gives access to the exit static pressure. A u-shape water manometer is used to measure the pressure drop from the upstream ambient pressure (the upstream stagnation pressure) to the downstream static pressure, giving the pressure loss through the test section. The function generator provides AC voltage and the piezoelectric voltage amplifier provides amplified voltage to the PTA. A flow meter between the test section and the vacuum pump measures volume flow rate of the channel flow. More detailed illustrations of the heat transfer experiment test section are shown in Fig. 8. Fig. 8(a) shows the copper block surrounded by insulation and the position of the heater at one end of the block. The PTA is positioned facing the convection surface and frames supporting the carbon fiber blade inside the channel go through the transparent channel wall opposite to the heated surface. The holes for the supporting frames on the channel wall were completely sealed with flexible latex to make sure there is
no air leakage to the channel through the holes. At the upstream and downstream edges of the heated area, there are adiabatic extended sections of the channel to allow for the flow to become fully developed. Dimensions in Fig. 8: channel height (h1), width (w), upstream length (L1), heated section length (L2), and downstream length (L3), are 20 mm, 4.3 mm, 32 mm, 50 mm, and 32 mm, respectively. Two thermocouples are used to measure air inlet temperature ($T_{in}$) at the channel entrance and one thermocouple is used to measure outlet temperature ($T_{out}$), as shown in Fig. 8(c). Ten thermocouples, denoted as $T_{sub,i}$, are used to measure temperatures 1 mm into the copper block above the convection surface. From these temperatures, the surface temperature ($T_{surface,i}$) can be found by extrapolation to the convection surface, as follows:

$$T_{surface,i} = T_{sub,i} + \frac{q \cdot l}{k \cdot A_c} \quad (1)$$

where, $q$, $l$, $k$, and $A_c$ express heat input from the heater, distance of thermocouples from the convection surface, copper thermal conductivity, and cross sectional area of the copper block, respectively.

The average temperature of the exposed copper surface, $T_{surface}$, assumed to be isothermal, is computed as:

$$T_{surface} = \frac{\sum_{i=1}^{N} T_{surface,i}}{N}, \quad N = 10 \quad (2)$$

The manometer measures the static pressure ($P_{static}$) at the discharge adaptor in which the pressure tap is located. The static pressure is expressed as:

$$P_{static} = 2yg \cdot (\rho_{water} - \rho_{air}) \quad (3)$$

where $y$, $g$, $\rho_{water}$, and $\rho_{air}$ are manometer reading, standard gravity, water density, and air density, respectively. The reference pressure is the ambient (also the upstream stagnation pressure). The discharge adaptor at the outlet is connected to the blower through a contraction section and a flexible hose. The pressure tap is located slightly downstream of this contraction section and, thus, is measuring the static pressure consistent with the velocity in that section. The exit total pressure ($P_{total}$) is:
where \( V_{\text{contract,exit}} \) is the average velocity at the location of the pressure tap. In order to get the pressure loss \( (\Delta P_{\text{test}}) \) across the heated section where the PTA operates, the total pressure losses of the inlet \( (\Delta P_{\text{inlet}}) \), upstream \( (\Delta P_{\text{L1}}) \), and downstream \( (\Delta P_{\text{L2}}) \) were subtracted from the overall total pressure drop \( (\Delta P_{\text{total}} = P_{\text{ambient}} - P_{\text{total, eq}}) \) as:

\[
\Delta P_{\text{test}} = \Delta P_{\text{total}} - \Delta P_{\text{inlet}} - \Delta P_{\text{L1}} - \Delta P_{\text{L2}}
\]

(5)

Each loss term can be calculated by using the equations shown as:

\[
\Delta P_{\text{inlet}} = K_{\text{loss,inlet}} \frac{1}{2} \rho_{\text{air}} V_{\text{in}}^2
\]

\[
\Delta P_{\text{L1,L2}} = f \frac{L}{2D_h} \rho_{\text{air}} V_{\text{L1,L2}}^2
\]

(6)

where, \( K_{\text{loss,inlet}} \) and \( f \) are the inlet loss factor and Darcy friction coefficients at sections L1 and L2, respectively. In addition, \( D_h, V_{\text{in}}, V_{\text{L1}}, \) and \( V_{\text{L2}} \) represent the hydraulic diameter of the channel, channel inlet velocity, velocity at section L1, and velocity at section L2, respectively. Heat transfer experimental results and measured pressure drops for each case are presented in the following sections.

### 3.2. Experiment results

For the heat transfer experiments, four different channel flow rates of 10 LPM, 20 LPM, 40 LPM, and 60 LPM were used to investigate the effects of different channel flow rates on heat transfer performance. The corresponding inlet average velocities and Reynolds numbers are listed in Table 5. Upstream of the heated surface, the flow rates of 10 LPM and 20 LPM fall into the laminar regime, based on computed Reynolds numbers, and transitional Reynolds numbers taken from the literature and the other two flow rates fall in the turbulent regime, in a non-agitated channel. For all the test conditions, 5 W of heater input was supplied. The cooling performance of the PTAs was mainly investigated by calculating heat transfer coefficients using the log mean temperature difference (LMTD) method with the following equation:

\[
h = q/A \cdot \Delta T_{\text{LMTD}}
\]

(7)

where \( q, A, \) and \( \Delta T_{\text{LMTD}} \) represent the heater input power, the heated surface area, and the log mean temperature difference, respectively. Herein, \( \Delta T_{\text{LMTD}} \) can be computed by:

\[
\Delta T_{\text{LMTD}} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)},
\]

\[
\Delta T_1 = T_{\text{Surface}} - T_{\text{In}},
\]

\[
\Delta T_2 = T_{\text{Surface}} - T_{\text{Out}}
\]

(8)

An uncertainty analysis was performed to estimate uncertainties in heat transfer coefficients. The heater input power, \( q \), is known to 10% uncertainty according to the power supply manufacturer's manual. Temperature measurements carry a 0.5 °C uncertainty. The uncertainty for the convection area was ignored as it is very small compared to the other terms. The total uncertainty of heat transfer coefficients propagated from each uncertainty listed above is 10%. Thus, the total uncertainty is dominated by the uncertainty of the heater input power to the heater. The carbon fiber blade of the PTA is located in the center of the channel. Therefore, the clearance between the blade surface and the heated copper surface is 1.4 mm when the PTA is at the middle of its oscillation cycle. When PTA-1 operates at its maximum peak-to-peak displacement of 1.8 mm, the minimum clearance between the blade and the heated surface becomes 0.5 mm. Fig. 9 present the comparison of heat transfer coefficients of four PTAs operating at different displacements and channel flow rates. Overall, heat transfer coefficients increase with increasing displacement, frequency, and flow rate as anticipated. From Fig. 9, the peak-to-peak displacement of 0 mm.
represents a condition in which the PTA does not operate. For PTA-1 and PTA-2, the heat transfer coefficients are almost the same when the displacement increases from 0 mm to 0.4 mm and the flow rates are in the upper range of 40 LPM and 60 LPM. This implies that the lower frequency of translational agitation below 700 Hz has no effect on heat transfer under this high channel flow velocity condition. On the other hand, there is slight enhancement about 5%, above the idle state condition in heat transfer coefficient when PTA-3 and PTA-4 operate with 0.4 mm displacement at flow rates of 40 LPM and 60 LPM. At the lower flow rates of 10 LPM and 20 LPM, all the PTAs provided great improvement in heat transfer coefficient once the PTAs are turned on with displacements of 0.4 mm or larger. Considering 10% uncertainty in the heat transfer coefficient, one can say that PTA-1 and PTA-2 present almost equivalent cooling performance even though the performance of PTA-2 appears to be slightly better than PTA-1. Agitators PTA-1 and PTA-2 show the best heat transfer enhancement of 158% and 178% compared to no-agitation cases at the lowest flow rates of 10 LPM and with a peak-to-peak displacement of 1.8 mm. Agitator PTA-3 provided the best cooling performance among the four PTAs. The performance of PTA-4 cannot be compared with other cases in a fair way since its available displacement is limited at 0.8 mm. Agitator PTA-3 generated the largest heat transfer coefficient enhancement, even with the smaller maximum displacement than those of PTA-1 and PTA-2. When PTA-3 operated with 1.4 mm peak-to-peak displacement, the heat transfer coefficient of 283 W/m²K was achieved under a channel flow of 60 LPM. This is a 55% improvement in heat transfer coefficient compared with the non-agitated state. In the laminar flow regimes, heat transfer augmentation by agitation is more significant than the agitation enhancement in the turbulent flow regime.

Fig. 10 represents pressure drop through the heated section (section ‘L2’ in Fig. 8(c)), and the corresponding flow power required to drive the flow through that section at each flow rate with and without PTA operation. The flow power is computed by multiplying the volume flow rate and the pressure drop within the test section. On the whole, pressure drop and flow power follow the trends of heat transfer coefficient enhancement. Higher frequency, displacement, and flow rate result in larger pressure drop and flow power. In addition, pressure drop due to increasing flow rate at a fixed PTA displacement overwhelms the pressure drop from increasing the PTA displacement at a fixed flow rate. When the PTA is idle, the pressure drop increases 1500% purely due to the raise of a flow rate from 10 LPM to 60 LPM. For all the PTAs, as the flow rate rises and the displacement remains unchanged, the increasing rate of the pressure drop is similar to that observed with channel flow-only. Similarly, the effect of increasing PTA displacement on pressure drop is insignificant, compared to the effect of the flow rate on pressure drop. Agitator PTA-2 showed the largest increase of pressure drop of 150% above the zero displacement case when the agitator is operated with 1.8 mm displacement at 10 LPM of flow rate. As the flow rate rises, increases of the pressure drop due to displacement decrease to around 40% at the 60 LPM flow rate. The pressure drop ranges between 20 Pa and 520 Pa, according to operating conditions of the PTA and channel flow rate. The flow power ranges between 0.003 W and 0.35 W under the various experimental conditions. The pressure drop and flow power presented here can provide an estimate of fan power needed when the translational agitator is coupled with the external fan to provide channel flow and agitation within the heat sink system.

In addition, PTA-1 and PTA-2 were tested with the channel ends open and without the flow forced by the fan, to study the sole per-
formance of the PTA as a stand-alone cooling device. For the experiment, the cover plate in Fig. 8(d) was removed to make the channel open to the ambient air. Then, the PTA is placed in the same position as it was in the channel flow tests. For the open channel test, the heat transfer coefficient of the heated surface was calculated as:

\[
h = \frac{q}{A \cdot (T_{\text{surface}} - T_{\text{air}})}
\]

(9)

where \(T_{\text{air}}\) is the averaged ambient air temperature measured by three thermocouples adjacent to the heated surface area. Fig. 11 compares the heat transfer coefficients of the heated surface cooled by PTA-1 and PTA-2 in the open channel to the cases of the PTAs and a cross flow of 10 LPM. In the open channel, the heat transfer coefficient at a displacement of 0.0 mm shows rather higher levels considering it is a natural convection condition. It is recorded that the ambient air currents were strong in the test room. Similar currents may be seen in a forced-air vented electronics cabinet. The results show that the PTA alone can provide fairly large cooling performance compared to those of the combined system of a PTA and channel flow. When PTA-1 and PTA-2 are operating at 1.8 mm of displacement in the open channel, 174 W/m² K and 195 W/m² K of heat transfer coefficients were achieved which are 83% and 86% of the heat transfer coefficients from the cases with PTAs and 10 LPM of channel flow.

4. Total Reynolds number and Stanton number analysis

For more detailed analysis of the heat transfer experimental results, a total Reynolds number (\(Re_{\text{tot}}\)) was proposed to characterize the combined effects of the translational agitation and channel flow. The channel flow Reynolds number (\(Re_c\)) is defined as:

\[
Re_c = \frac{V_{\text{in}} D_h}{\nu}
\]

(10)

where \(V_{\text{in}}\), \(D_h\), and \(\nu\) are the channel flow bulk velocity, hydraulic diameter of the channel, and the kinematic viscosity of air, respectively. The Reynolds number (\(Re_{\text{pta}}\)) for the PTA is defined with the maximum blade velocity and the hydraulic diameter of the blade as:

\[
Re_{\text{pta}} = \frac{\omega A D_b}{\nu}
\]

(11)
where $\omega$, $A$, and $D_b$ are the angular frequency in rad/s, the maximum mean-to-peak displacement, and the hydraulic diameter of the blade, respectively. The blade hydraulic diameter is defined as:

$$D_b = 4P_b/A_b$$

where $P_b$ and $A_b$ are the perimeter and area of the blade, respectively. From the two Reynolds numbers shown above, the total Reynolds number can be defined with the second power relation as follows:

$$Re_{tot} = \sqrt{Re^2 + Re_{pr}^2} = \sqrt{\left(V_{in} D_b\right)^2 + (\omega A D_b)^2}$$  \quad (12)

In Fig. 12, the heat transfer coefficients of all the cases shown in Fig. 9 were plotted with respect to this total Reynolds number. The heat transfer coefficients are distributed in a linear relation with total Reynolds number, as displayed in the figure. From the two Reynolds numbers shown above, the total Reynolds number can be defined with the second power relation as follows:

$$Re_{tot} = \sqrt{Re^2 + Re_{pr}^2} = \sqrt{\left(V_{in} D_b\right)^2 + (\omega A D_b)^2}$$  \quad (12)

In the equation:

$$Re_{tot} = \frac{(h - \Pi) D_b}{k_i} = CRe_{tot}Pr, \quad C = \frac{\Omega}{Pr} \frac{D_b}{k_i}$$  \quad (13)

The Stanton number is defined as:

$$St = \frac{(h - \Pi) D_b}{k_i} = \frac{\sqrt{\left(V_{in} D_b\right)^2 + (\omega A D_b)^2}}{k_i} \frac{\rho CV}{(\omega A)^2} \left(\cdot Pr = \frac{\rho CV}{k_i}\right)$$  \quad (14)

with the further arrangement, the equation becomes the form of Stanton number ($St$):

$$St = \frac{(h - \Pi)}{\rho CV (V_{in})^2 + (\omega A)^2 (A_j)^2} = C$$  \quad (15)

which contains the effective velocity defined as:

$$V_{eff} = \sqrt{\left(V_{in}\right)^2 + (\omega A)^2 \left(\frac{D_b}{D_b}\right)^2}$$  \quad (16)

The electrical power consumption of the PTAs were measured to investigate their feasibility as electronics cooling devices; results are shown in Fig. 15. The power consumption was measured based on the following equation.

$$P = \frac{1}{T} \int_0^T V(t) \cdot I(t) \, dt$$  \quad (18)
where $T$ is the data-taking period, $V(t)$ and $I(t)$ are voltage and current to the piezo stack actuator. The actuator PTA-3, which provided the best cooling performance among the PTAs, consumes about 9 W when it is operating at 1.4 mm displacement at the second resonance mode. Additionally, the coefficient of performance (COP) values of the PTAs were evaluated, defined as:

$$\text{COP} = \frac{Q_{30-80}}{P_{in}}$$

(19)

$Q_{30-80}$ is the converted heat transfer rate when channel inlet and convection surface temperatures are assumed to be 30 °C and 80 °C, respectively. This is generally a desired operating condition of chips or CPUs in electronic devices. Therefore, $Q_{30-80}$ are computed from the measured heat transfer coefficients at different operating conditions shown in Fig. 9, assuming that inlet and surface temperatures are 30 °C and 80 °C. The power, $P_{in}$ includes the electrical power input to the PTA (Fig. 15) and flow power required to drive the flow through the test section (Fig. 10). Fig. 16 shows the calculated COPs in a log-scale with respect to the total Reynolds number defined previously. Overall, COP tends to decrease with increasing total Reynolds number. The PTA with a lower operating frequency shows a higher COP at a fixed displacement and channel flow velocity. In addition, PTAs are more efficient at a lower channel flow velocity and smaller oscillation displacement. The COP analysis indicates that the PTA system is providing good cooling performance even at a less active operating condition with a lower power consumption. This reflects a positive side in terms of the noise aspect as the PTA with a smaller displacement and lower frequency generates much less noise.

5. Conclusions

Four piezoelectrically-driven in-channel flow agitator designs were fabricated and tested as potential electronics cooling devices. In order to create a high-frequency and large-displacement translational motion, an oval loop shell structure was made of spring steel. It successfully amplified a few microns of piezoelectric stack range to the millimeter displacement ranges needed for the agitators. The oval loop shell mainly uses the second resonant mode of the structure so that the agitator is excited and the piezoelectric stack is not. The operating frequencies of the four PTAs range from 596 Hz to 1080 Hz with amplified displacements of up to 1.8 mm. The cooling performance of each PTA was tested in a channel flow situation with one side heated and through-flow rates of between 10 LPM and 60 LPM, representing laminar to turbulent flows. The best cooling performance was provided by a system operated with 1.4 mm displacement and 60 LPM. This was a 55% improvement in heat transfer coefficient compared to the non-agitated state with the same channel flow. In addition, the pressure drop in the test section and the corresponding required flow power to drive the flow through the channel against the oscillating blade were measured. The pressure drop ranges between 20 Pa and 400 Pa according to the different operating conditions of the PTA and the rates of channel flow. The flow power ranges between 0.003 W and 0.274 W at different experimental conditions. The PTAs were also tested in the open channel to examine the sole effect of the PTA and to demonstrate the possibility of using it for stand-alone cooling.

It was shown that the proposed agitators demonstrated excellent capability to enhance the traditional fan-cooled heat sink or to operate as a stand-alone cooling device. For collapsing the data, a PTA Reynolds number and a total Reynolds number were defined to characterize the combined effects of cross flow and agitation. Measured heat transfer coefficients from all the test conditions were fit with a linear relationship in terms of total Reynolds number. This relationship gave a Stanton number that allows predicting the operating performance of the PTA-activated cooling module.

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References
