Optimization of Piezoelectric Oscillating Fan-Cooled Heat Sinks for Electronics Cooling

James Petroski, Mehmet Arik, and Mustafa Gursoy

Abstract— **Piezoelectric fans have been investigated for electronics cooling over the last decade. The primary usage or method has been to place the vibrating fan near the surface to be cooled. The piezofan used in the current study is composed of a piezo actuator attached to a flexible metal beam. It is operated at up to 120-VAC and at 60 Hz. While most of the research in the literature focused on cooling bare surfaces, larger heat transfer rates are of interest in the present study. A system of piezoelectric fans and a heat sink is presented as a more efficient method of system cooling with these fans. In this paper, a heat sink and piezoelectric fan system demonstrated a cooling capability of 1 C/W over an area of about 75 cm^2 where electronic assemblies can be mounted. The heat sink not only provides surface area, but also flow shaping for the unusual 3-D flow field of the fans. A volumetric coefficient of performance (COP$_v$) is proposed, which allows a piezofan and heat sink system volume to be compared against the heat dissipating capacity of a similar heat sink of the same volume for natural convection. A piezofan system is shown to have a COP$_v$ of five times that of a typical natural-convection solution. The paper will further discuss the effect of nozzles in flow shaping obtained via experimental and computational studies. A 3-D flow field of the proposed cooling scheme with a piezofan is obtained via a flow visualization method. Velocities at the heat sink in the order of 1.5 m/s were achieved through this critical shaping. Finally, the overall system characterization to different heat loads and fan amplitudes will be discussed.**

Index Terms— **Electronics cooling, flow visualization, heat sink, piezoelectric fans, piezofans, volumetric coefficient of performance (COP$_v$).**

Nomenclature

A	Coded variable, input wattage.
B	Coded variable, piezofan amplitude.
COP_v	Volumetric coefficient of performance.
h	Heat transfer coefficienct [W/m2-K].
Q	Heat transfer [W].
R_{th}	System thermal resistance [C/W].
T	Temperature [C].

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J. Petroski is with the GE Lumination, Valley View, OH 44125-4635 USA (e-mail: jim.petroski@graftech.com).

M. Arik is with the GE Global Research Center, Niskayuna, NY 12309 USA (e-mail: arik@crd.ge.com).

M. Gursoy is with Pro Solutions, Troy, NY 12180 USA (e-mail: mustafa.gursoy@calikenerji.com).

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I. Introduction

PIEZOFANS (also called piezoelectric fans or piezoceramic fans), which were developed in the 1970s, have received some attention for use in cooling applications throughout the last few decades. Beginning with Toda's work [1], [2], over 25 years of development has occurred with analytical and experimental work, and numerous patents have been issued for piezofan applications.

A piezofan is a resonant device that uses a piezoceramic material to induce oscillations in a cantilever beam that is used to move air. Typically, a piezoelectric patch is bonded to a thin material or shim and an alternating current is applied at the same frequency as the shim's resonant frequency. The small displacements of the piezoelectric patch are amplified in the shim to produce oscillations, usually ranging from a few millimeters to 2.5 cm. Although the ac voltages are typically line voltages (e.g., 120-VAC), many electronic devices are stationary and plugged into the grid, providing a source for electricity (one good example would be LED lighting systems). Portable devices can use a small electrical conversion circuit to convert dc battery power to the required ac.

Figs. 1 and 2 show a piezoelectric fan assembly and an oscillating fan's range of motion. The area with the largest oscillations is the free end of the cantilever beam and does not have the piezoceramic material in that region.

Operating the fan at the resonance frequency is desirable. First, these fans do not require large amounts of power and typically operate in the range of 10 to 20 mW. Second, small deflections in the piezoelectric patch are amplified to large values at the tip of the fan to create the stirring leading to air movement. Most devices are driven at frequencies under 100 Hz so that the acoustic energy and noise levels are kept low (i.e., under 25 dBA).

The initial patents published through the 1980s and 1990s show that most attention was focused on how to move air with the intent to replace other air movers such as rotating fans [3]. While cooling was often mentioned as an application for the movement of air, the discussions were limited to the idea of replacing a rotary fan and moving air in the same fashion or patterns as the rotary unit. Fans were placed in ducts or conduits to create a semi-straight line flow to feed air flow across items to be cooled as presented in [3] and [4].

Several studies in the past have characterized much of the flow behavior and direct cooling abilities of a piezofan. Schmidt [5] investigated distance variations from the fans to the surface to be cooled and determined the variation in heat

Fig. 1. Typical infrasonic piezoelectric fan (piezoceramic patch on reverse side).

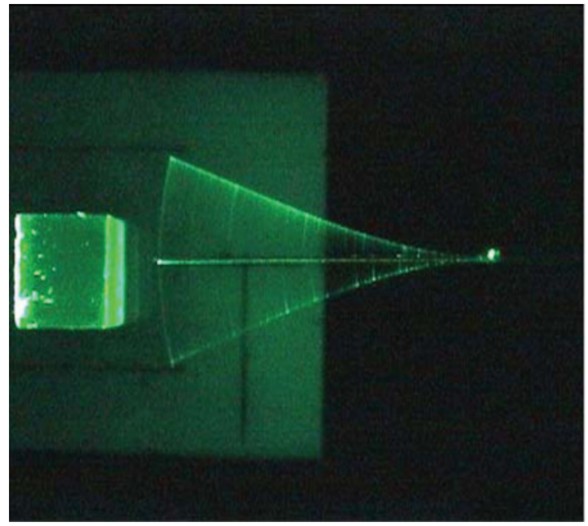

Fig. 2. Piezofan motion (used with permission from Suresh Garimella, Purdue University).

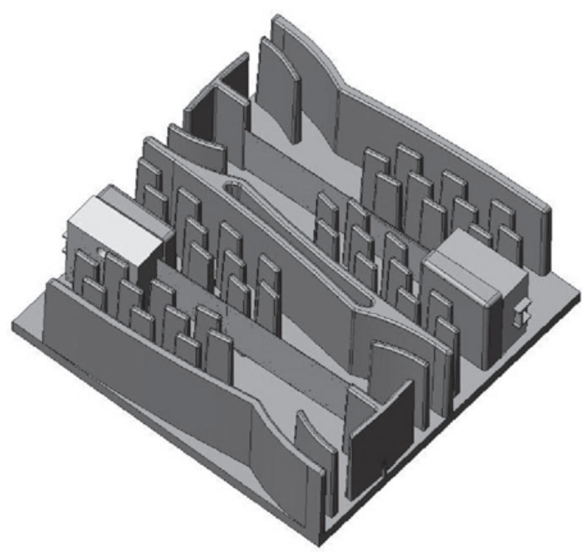

Fig. 3. Embedded piezofans in a heat sink.

transfer coefficients of the system. Açikalin *et al.* [5] and Wait *et al.* [6] investigated the use of piezofans in small-scale electronics cooling applications in cellphones and laptops. The unusual flow fields involving vortices and vortex shedding and recirculation patterns have been characterized by experiment and computational methods [7]–[10].

Heat transfer studies to understand the capabilities of the fans have been published in recent years. The predominant fan orientation has been direct impingement of the vortices from the fan tip on a heated component or surface [5]–[7], [11]. Other work has involved using the downstream vortex shedding to push through traditional heat sinks or other surface-area-enhancing-devices [11]. Some recent work has involved studying the coupling of multiple piezofans cooling the same surface for enhancement effects [12], [13].

Energy efficiency of low-form-factor cooling devices such as synthetic jets, muffin fans, and piezo fans have been studied by Arik *et al.* [14]. Their study showed that piezofans are a candidate cooling technology with or without heat sink integration due their low power consumption and low noise. Emerging thermal technologies are very promising, while development efforts are underway to make them feasible for many low-form-factor confined space electronics cooling challenges.

II. Enhancement of Piezofan-Cooling

The piezoelectric fans have primarily been studied as a 2-D flow object. This is a useful method and has led to appropriate use of the fans for cooling, but has not been optimized to take the full 3-D flow field advantage.

Additionally, decoupling of the fan from the object that is being cooled, along with the view that the airflow field is primarily 2-D, tends to limit the application of the fan in electronics cooling. In a few instances, a piezoelectric fan has been inserted directly into a heat sink for better cooling, such as in U.S. patent 4 923 000 and [14]. But these have been exceptions, and the heat sink designs are often the same used for natural and forced convection along the fins.

A preferred flow-optimized heat sink concept is proposed, which is a combination of a heat sink, a piezofan mount, and a 3-D flow-shaping/optimized design to significantly enhance the cooling. Fig. 3 is a depiction of such a system.

The moving piezofan blade creates and sheds vortices at the tip of the fan blade. The heat sink shown in Fig. 3 uses curved surfaces to shape the vortices and uses their kinetic energy to effectively couple the fan flow with finned surfaces with a minimal pressure penalty. This creates a vacuum in the area where the fan blade is oscillating and in turn draws in new cooler air. This leads to an efficient distribution of flow over surfaces in addition to effective cooling by the vortices at the fin surfaces.

The geometry of the heat sink is composed of two identical chambers, rotated 180° with respect to each other, as shown in Fig. 4. The heat sink is made up of a thermally conductive material such as aluminum, and on the flat underside the heat load is applied. This heat is then transferred to the heat sink fins and the areas where the piezofans are oscillating.

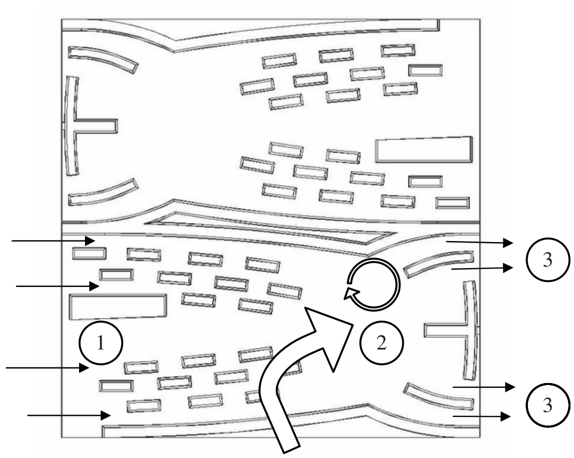

Fig. 4. Top view of the heat sink (fans removed for clarity).

As a piezofan oscillates, the vortices are created in region 2 of Fig. 4. The vortices are shaped by this chamber area to create linear flow in the outlet region, which is region 3 (there are two outlets for each fan).

Incoming cooler air is drawn into the system in two distinct places. First, incoming air enters the heat sink by the base of the piezofan and creates laminar flow across the fins shown in region 1 of Fig. 4. It then enters region 2 where the vortices are formed. Additional cooling air is drawn from the heat sink, above region 2, so there is fresh cooling air being entrained into the airflow while the vortices are being formed.

This design is created to take advantage of the unique 3-D airflow of the piezofans. As previously mentioned, there is a full 3-D flow field created by a piezofan operating in free air. Most work in the previous literature has confined the fans to some degree and attempted to reduce them to a 2-D flow pattern. This can cause a reduction in the actual airflow, as the fan is more constrained and the viscous effect of air near boundaries creates drag. In the laboratory, it was noted during different tests that placing flat planes near the oscillating end of the piezofan (for example, placing a cover over the heat sink of Fig. 1) actually reduces the amplitude of the fan noticeably and reduces cooling within the system.

Experiments in open ambient revealed that there is a low-pressure zone created in the plane of the fan blade, on either side of the blade, where most of the oscillation is occurring (this would be above region 2 in the open air or below it, next to the heat sink). Making use of this low-pressure zone allows additional cooling air to be brought into the airflow streams and enhances cooling. This zone was found through smoke flow visualization tests, and future work will attempt to quantify the amount of this pressure change. An interesting observation found during the design is the placement of a constriction point relative to the channel and the fan. Fig. 5 shows a schematic of an experimental study that was performed with the constriction points determined by the (x, y) coordinates. A simple full factorial (three-level) design of experiments (DOE) was conducted and the airflow velocities from the outlets recorded as the output variable. In general, the constriction placed closer to the moving end of the piezofan performed the best for outlet flow velocities, even

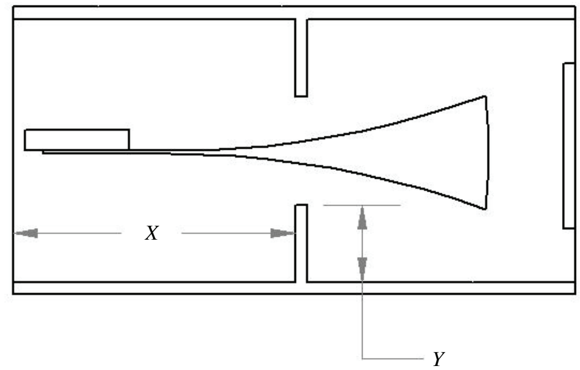

Fig. 5. Geometry and placement of the piezofan.

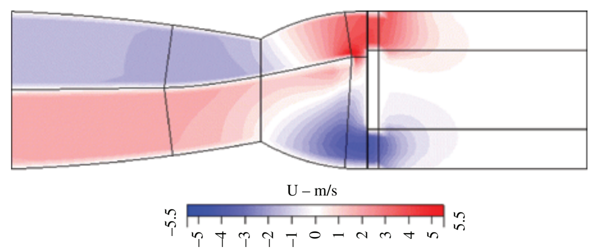

Fig. 6. Pressure zones caused by fan displacement.

though the y value had to be minimized due to the proximity of the moving blade. Actual results with performance values for some different cases are discussed at the end of this paper.

This particular heat sink design has several advantages. First, the fans and the heat sink are brought together in one assembly. The piezofan is not a separate air mover outside the heat sink but integral to it. This allows the unique airflow of the piezofan directing the convection of air off the various heat sink surfaces. These extended surfaces of the heat sink are designed to intrude into the various airstreams and maximize the heat transfer.

A. Computational Study

A computational study was carried out via commercially available computational fluid dynamic (CFD) software (CFD ACE+). Several 3-D models were created to examine the flow patterns and flow shaping of different heat sink designs. Figs. 6 and 7 show the calculated pressure zones and velocity patterns with a cut through the center of a fan for the heat sink of Fig. 3. The analysis was performed by a deforming mesh routine where the piezofan shape was entered into the CFD model as a function of time. The quasi-steady state figures shown below are representative of the results after five or more full cycles (to allow the flow field to develop in the transient analysis). Various grid densities were tested to ensure that these results are grid-independent.

B. Flow Visualization

Understanding the flow scheme in the heat sink is very critical. Therefore a laser Doppler anemometry (LDA) study was completed to validate the CFD models and to obtain the optimal design of the heat sink. Fig. 8 shows the experimental setup. A 400-mW Argon-ion laser provided the

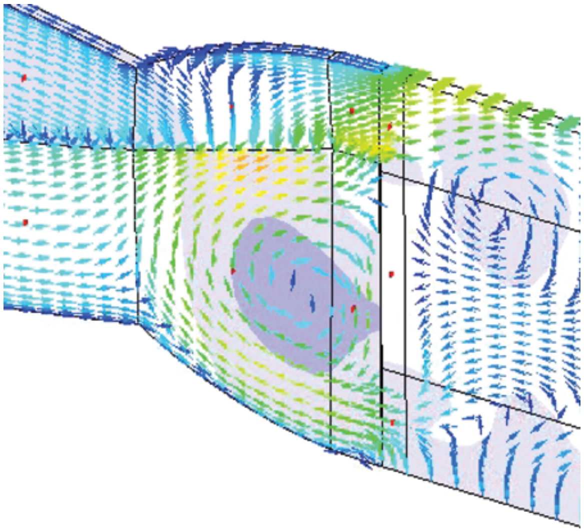

Fig. 7. Velocity vectors during operation.

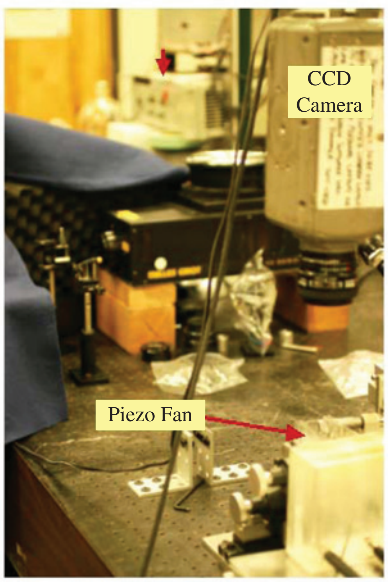

Fig. 8. Schematic of the LDA experimental setup.

light source, which was guided via appropriate lenses. A high-speed charge-coupled device (CCD) camera allowed capturing images within the low-frequency operation of the piezofan heat sink assembly. To avoid the external boundary effects, the heat sink structure is placed in a box, which is 10× from each side. Therefore, air drifts from room ambient were avoided. Later, the collected data were studied, and local flow conditions were identified.

A sample frame of one test is shown in Fig. 9, which depicts the same pattern of air ejection at the outlets with a low-pressure vortex in the wake of the fan. Note the laminar flow across the upstream cooling fins in the piezoceramic patch area (region 1 of Fig. 4).

System exit velocities were also measured just outside the heat sink nozzle exits. The highest velocities were measured with the largest piezofan amplitudes, and peak velocities of

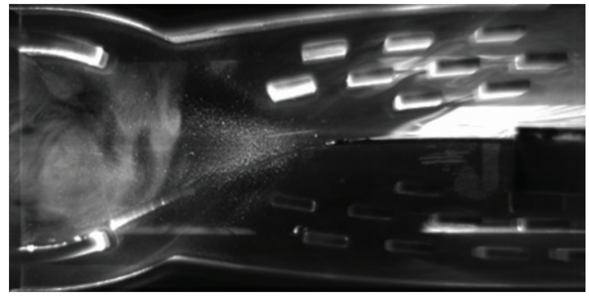

Fig. 9. Smoke pattern in the heat sink.

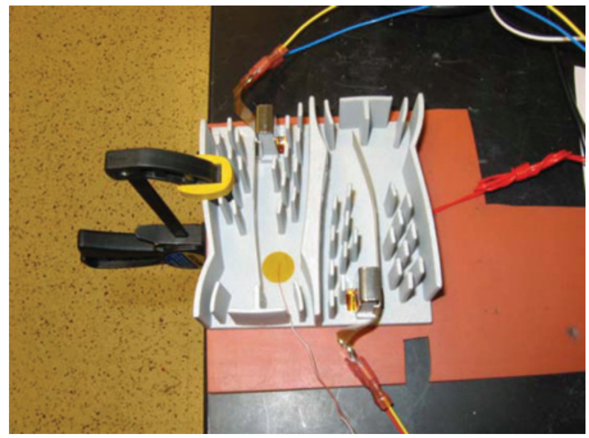

Fig. 10. Thermal test setup of piezofan system.

1.5 m/s were measured with a hot-wire anemometer. The heat sink fins closer to the fan's tip would be subjected to these high velocities and are responsible for most of the heat transfer along with the flat surface under the oscillating region.

III. System Performance

The piezofan assembly shown in Fig. 3 was built and tested in the laboratory. The heat sink was formed from cast aluminum via a rapid prototyping process. The base dimension of the heat sink is 85 mm × 85 mm. Power was applied via a flat resistance heater during testing, and the amplitude of the piezofans (peak-to-peak) was varied.

To characterize the system performance, a two-level three-factorial design-of-experiment matrix was composed. Power inputs of 20, 30, and 40 W were used with piezofan amplitudes of 19, 22, and 25 mm. Four test runs at the "zero point" (30 W and 22 mm) were made for error estimation. Measurements were made of the heat sink temperature near region 2 of Fig. 4. This test configuration is shown in Fig. 10.

Ambient and local heat sink temperatures were collected. The radiated power to the environment was calculated, as well as the system thermal resistance based on the measured temperatures. Table I shows the system thermal resistance versus the piezofan amplitude.

The experimental data was analyzed with Design-Expert 6 software, which is a commercially available code for design of experiments. One of the interesting plots is the surface plot of system thermal resistance versus input power and piezofan amplitude, as it shows the trends in the system. This plot is

TABLE I

SYSTEM THERMAL RESISTANCE (R_{TH})

Piezofan amplitude [mm]	R_{th} [°C/W]	σ [°C/W]
19	1.094	0.052
22	1.032	0.036
25	0.920	0.013

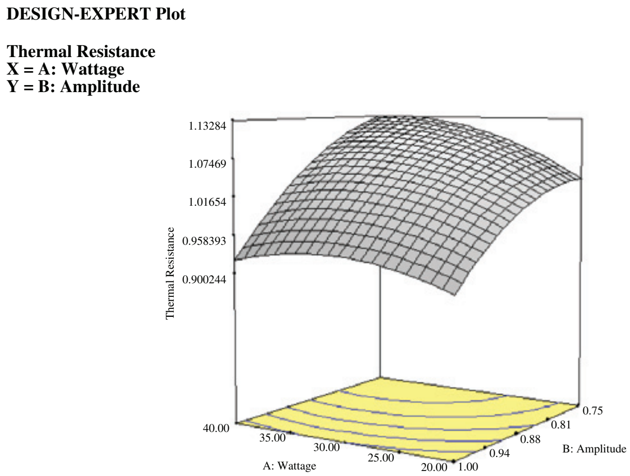

Fig. 11. Surface response for thermal resistance.

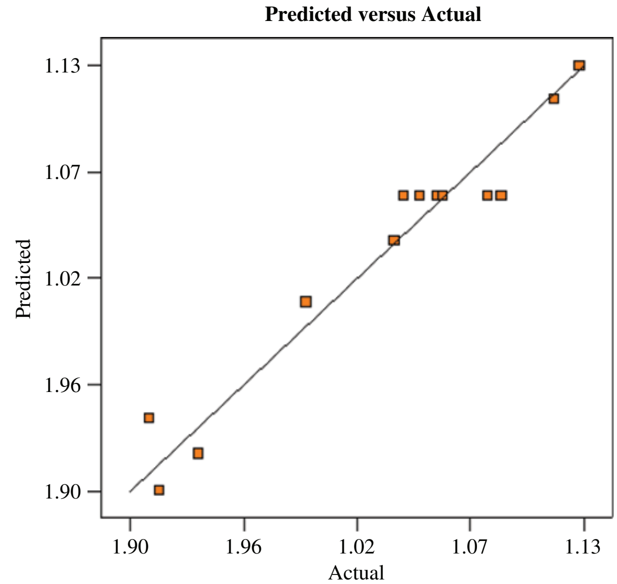

Fig. 12. Comparison of actual and predicted thermal resistances.

shown in Fig. 11. Note that the amplitude axis in the figure gives values in inches and not in millimeters.

Another point of interest from this response surface is the greater variation in the system thermal resistance when lower amplitudes are used for the piezofan. This implies that some level of natural convection is occurring in the system, and the most likely places for this is on the outside surfaces of the heat sink, where no fan air movement touches the surfaces, and in the lower flow regions of the heat sink inlets.

A transfer function (or equation) of thermal resistance based upon the independent variables of power and amplitude may be created. It is significant that the independent variables appear not only in linear form but also in quadratic and mixed terms, similar to the results reported previously [8]. The system is particularly sensitive to changes in piezofan amplitude. Equation (1) is written in terms of "coded values." These are values of $-1, 0$, and $+1$ used for the three levels of each input factor. For example, 20 W would be entered as -1, 30 W as zero, and 40 W as $+1$. With A being the power variable, and B the amplitude variable, this system transfer function (found using a regression calculation within the Design-Expert software) can be written as

$$R_{\mathrm{th}} = 1.06 + 0.029A - 0.087B - 0.029A^2 - 0.035B^2 - 0.019AB. \quad (1)$$

It is found that there is a reasonable statistical correlation, as the R^2 adjusted value is approximately 91%. The fit is graphically seen in a plot of actual thermal resistance values versus the predicted values of the transfer function (Fig. 12). The reader may note that several of the experimental points have a similar predicted value of 1.06 °C/W using the regression, while the actual values were from 1.04 to 1.09 °C/W; this line is more of an anomaly than a trend and is within a reasonable error for the regression.

IV. COMPARISON TO NATURAL CONVECTION

One area of interest in comparison to cooling systems is to compare system performance to other standard cooling methods (e.g., natural convection or other forced convection methods). As noted in [14], the piezofan is a cooling method that fits in between the regions of natural convection and the more powerful rotary fans for cooling. When some degree of enhancement over natural convection is needed, but not to the extent provided by typical forced convection systems, a comparison with natural convection is the better choice. In many systems, the overall thermal resistance is most frequently used to calculate the system efficiency for comparison purposes.

However, this method has some drawbacks when considering new methods and how they may be employed. In natural cooling systems, a more effective system can often be made but is done at the expense of size and weight (larger heat sinks). Since size is often as much of a design restriction as performance, one other method to measure a system's performance is to compare heat removal ability versus the volume of the cooling system. Here, we propose a metric termed the *volumetric* coefficient of performance (COP_v).

Such a metric was used to characterize this piezofan system. A similar size, shape, and volume extruded finned heat sink was used to evaluate natural convection performance in the same orientation (horizontal) as the piezofan system. Fig. 13 shows the heat sink used for comparison in this paper. The heat sink was evaluated with the heated (flat) surface facing down, on a similar insulated surface, so it would be in the same orientation as the piezofan system. Though not the most optimal natural convection solution, it is the most common

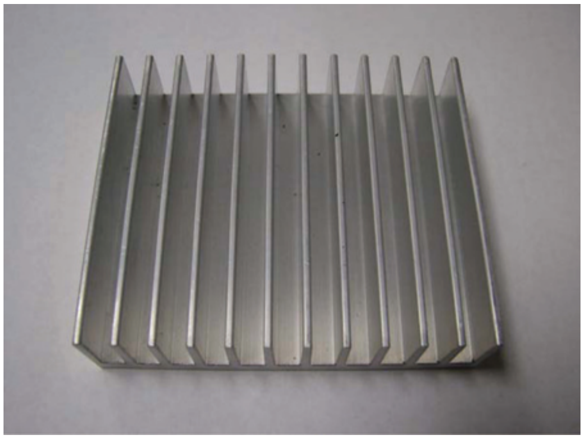

Fig. 13. Extruded heat sink for COP_v comparison.

TABLE II
VARIOUS RESULTS FOR DESIGN PARAMETERS

Variable	Design P1	Constrictive P2 design	Optimized final design
x (see Fig. 5)	60	48	64
y (see Fig. 5)	8	9	7
COP_v	1.7	2.0	5.0

design used in most applications and is fairly close to an optimal heat sink.

To determine the COP_v, both systems were heated with a flat resistance heater that covered most of the flat underside, and power was applied to achieve a base heat sink temperature of 60 °C in a 21 °C ambient condition. The heat sink underside with the heater was also insulated to ensure that nearly all the applied heat would be dissipated in the heat sink (see the orange insulating material shown in Fig. 10). The power required to achieve this temperature at steady state was recorded. With this data, the following equation was used to determine the COP_v:

$$COP_v = \frac{Q_{\text{pzf}}}{Q_{\text{nc}}} \tag{2}$$

where Q_{pzf} is the resistive heating power applied for the piezofan system and Q_{nc} the power applied for the natural convection system.

Several heat sink design iterations prior to the design of Fig. 3 were tested, and Fig. 3 is the final design after optimizations were applied. As the systems were built and tested, COP_v's of 1.5 to 2.7 were initially achieved. As the design matured to what is shown in Fig. 3, COP_v's of 5:1 were achieved—a significant result, as the volume of the heat removal system is now five times more efficient at than with natural convection. It is believed further refinements to the design will allow higher values of COP_v to be achieved. Future work will entail mapping the airflow in 3-D around the heat sinks to understand the airflow patterns noted in Fig. 4 and discussed earlier. Also, in the future a study may be undertaken to evaluate the system when the heated surface is in a vertical orientation rather than horizontal (one would expect lower COP_v's since the extruded heat sink will be more efficient in this orientation).

Table II tabulates some of the designs that were tested in the constriction point DOE and later built into thermal test units, and the results. This allowed the final optimized design to be arrived at with only a few design iterations, and improving the COP_v from 1.7 to 5.0 was largely due to this paper.

V. CONCLUSION

Natural convection has been used extensively for cooling electronics, but piezofan systems can be used in integrated designs to increase the efficiency of cooling for a given volume. Piezoelectric fans have been investigated for electronics cooling over the last decade. The piezofan used in the current study is composed of a piezo actuator attached to a flexible metal beam operated at 120-V and at 60 Hz. It is shown that those miniature fans can enhance heat sink surface heat transfer compared to cooling bare surfaces in prior published literature. A system that is capable of dissipating five times the power for the same volume has been demonstrated by optimizing a heat sink design for the unique 3-D flow of a piezofan. A heat sink and piezoelectric fan system demonstrated a capability of cooling (about 1 C/W) over an area of about 75 cm^2. The heat sink not only provided the surface area but also flow shaping for the unusual 3-D flow field of the fans. A novel flow shaping concept is obtained via experimental and computational studies. 3-D flow fields of the proposed cooling scheme with a piezofan is presented via LDA. Velocities at the heat sink in the order of 1.5 m/s were achieved through this critical shaping obtained via experimental and computatuional models.

A COP_v is proposed, which allows a piezofan and heat sink system volume to be compared against the heat dissipating capacity of a similar heat sink of the same volume for natural convection. A piezofan system is shown to have a COP_v of five times the typical natural convection solution. Proposed cooling schemes are capable of not just hot spot cooling, as previously published piezofan studies, but now can cool a larger heated area effectively. The performance of such a system fits well in between the regimes of typical natural convection and normal forced convection of rotary fan systems.

REFERENCES

[1] M. Toda, "Voltage-induced large amplitude bending device PVF2 bimorph its properties and applications," *Ferroelectrics*, vol. 32, no. 1, pp. 127–133, 1981.

[2] M. Toda, "Theory of air flow generation by a resonant type PVF2 bimorph cantilever vibrator," *Ferroelectrics*, vol. 22, no. 8, pp. 911–918, 1979.

[3] H. H. Kolm and E. A. Kolm, "Solid state blower," U.S. Patent 4 498 851, Feb. 12, 1985.

[4] Y. Yamada, K. Fujimoto, and J. Inoue, "Piezoelectric fan," U.S. Patent 4 780 062, Oct. 25, 1988.

[5] R. R. Schmidt, "Local and average transfer coefficients on a vertical surface due to convection from a piezoelectric fan," in *Proc. Intersociety Conf. Thermal Phenomena*, 1994, pp. 41–49.

[6] T. Açikalin, "Thermal and fluidic characterization of piezoelectric fans," Ph.D. dissertation, Purdue University, 2007.

[7] T. R. Walton, "Ducted oscillatory blade fan," U.S. Patent 4 834 619, May 30, 1989.

[8] T. Açikalin, S. Wait, S. V. Garimella, and A. Raman, "Experimental investigation of the thermal performance of piezoelectric fans," *Heat Transfer Eng.*, vol. 25, no. 1, pp. 4–14, 2004.
[9] T. Açikalin, S. V. Garimella, A. Raman, and J. Petroski, "Characterization and optimization of the thermal performance miniature piezoelectric fans," *Int. J. Heat Fluid Flow*, vol. 28, no. 4, pp. 806–820, 2007.
[10] T. Açikalin, A. Raman, and S. V. Garimella, "2-D streaming flows induced by resonating thin beams," *J. Acoustical Soc. Amer.*, vol. 114, no. 4, Pt. 1, pp. 1785–1795, 2003.
[11] M. Kimber and S. V. Garimella, "Local heat transfer characteristics of flows induced by multiple piezoelectrically actuated vibrating cantilevers," in *Proc. ASME-JSME Thermal Eng. Summer Heat Transfer Conf.*, Vancouver, Canada, Jul. 8–12, 2007, HT2007-32394.
[12] I. Sauciuc, "Piezo actuators for electronics cooling," *Electron. Cooling*, vol. 13, no. 1, pp. 12–17, Feb. 2007.
[13] M. Kimber, S. V. Garimella, and A. Raman, "Experimental mapping of local heat transfer coefficients under multiple piezoelectric fans," in *Proc. ASME Int. Mech. Eng. Congr. Expo.*, Chicago, IL, Nov. 2006, IMECE2006-13922.
[14] M. Arik, J. Petroski, A. Bar-Cohen, and M. Demiroglu, "Energy efficiency of low form factor cooling devices," in *Proc. ASME Int. Mech. Eng. Congr. Expo.*, Seattle, WA, Nov. 11–15, 2007, IMECE2007-41275.

James Petroski received the Bachelor's degree in engineering science and mechanics from Georgia Institute of Technology (Georgia Tech), Atlanta, GA, the M.S. degree in engineering mechanics from Cleveland State University, Cleveland, OH, and is currently pursuing the Ph.D. degree in mechanical engineering from Villanova University, Philadelphia, PA.

He is a Research Associate at GrafTech International in the Electronics Thermal Management group. He joined GrafTech in 2008 and is responsible for developing natural graphite materials for use in LED and other emerging technology applications. Prior to GrafTech, Jim worked at GE Lumination (formerly GELcore) for eight years and was responsible for mechanical and thermal design of various LED lighting systems used in residential, commercial, and industrial applications. He has been involved in thermal, shock and vibration management of electronics systems for DOD, NASA and commercial applications for over 25 years with a focus on passive cooling of electronics. He has authored several papers related to LED and electronics packaging and has over 15 patents either issued or pending pertaining to solid-state lighting.

Mehmet Arik received the B.Sc. degree in mechanical engineering from Istanbul Technical University, Istanbul, Turkey, the M.Sc. degree from the University of Miami, Miami, FL, focusing on single-phase forced convection turbulent flow in regular channels, in 1996, and the Ph.D. degree from the University of Minnesota, Minneapolis, focusing on the thermal management of high flux electronic components and microelectromechanical systems (MEMS), in February 2001.

Since December 2000, he has been with the General Electric Global Research Center in Niskayuna, NY, as a Senior Engineer and Project Leader on thermal management of electronics. He has expertise in air-cooled and liquid cooled power electronics, photonics packaging, and medical systems. He holds 20 U.S. patents and 15 more are pending. He has published over 60 papers in international journals and conferences in the fields of electronics cooling and MEMS.

Dr. Arik is a member of ASME, Sigma Xi and SPIE. He serves as an Associate Editor for IEEE TRANSACTIONS ON COMPONENTS AND PACKAGING TECHNOLOGY and ASME TSEA journals.

Mustafa Gursoy received the B.Sc. degree in mechanical engineering from Yildiz Technical University, Istanbul, Turkey, and the M.Sc. degree from Rensselaer Polytechnic Institute (RPI), Troy, NY, in 2006.

His research at RPI focused on natural convection on thin vertical rectangular surfaces. During his education at RPI, he was involved with the Thermal System Lab and the Performance Technology Lab, the GE Global Research Center, Niskayuna, NY. He is the Co-founder of Pro Solutions, Troy, NY, a base start up company. His research activities are focused on design and development of electronic cooling system technologies.