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Quantification of Piezoelectric Fan Flow Rate Performance and Experimental Identification of Installation Effects

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ABSTRACT

A piezoelectric fan is a flexible cantilever beam whose vibration is actuated by means of a piezoelectric material. Such fans have been employed for the enhancement of heat transfer by increasing the fluid circulation in regions which are otherwise stagnant. The main focus of past studies has been to describe the heat transfer achieved from these devices, as well as the flow field generated by vibrating cantilevers. In order to directly compare these fans with their traditional counterparts such as small axial fans, the present work casts the performance of piezofans in terms of a characteristic often used to represent conventional fans, namely the fan curve. The main thrust of this paper is to determine the relationship between the pressure and flow rate generated by miniature piezoelectric fans. Experimental measurements are obtained for two different fans with operating frequencies of 60 and 113 Hz. The maximum flow rate conditions yield nearly 30 l/min, while the greatest static pressure generated is found to be 6 Pa. The performance is highly dependent on both the vibration amplitude and frequency. Predictive relationships are developed to describe the experimental trends and provide insight into the sensitivity of pressure and flow rate to these operating parameters. A second thrust of this paper is to explore the effects of fan installation details on fan performance. The proximity of surrounding walls is considered through the use of three different enclosures within which the fan is mounted. Effective inlet areas from which the air enters the fan are also identified. This work provides a practical framework for determining the optimal placement and configuration for these fans in prototypical applications.

Keywords: electronics cooling, piezoelectric fans, fan curves, pressure, flow rate

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>A</td>
<td>Vibration amplitude</td>
</tr>
<tr>
<td>D</td>
<td>Piezoelectric fan width</td>
</tr>
<tr>
<td>d</td>
<td>Distance from fan tip to fan outlet</td>
</tr>
<tr>
<td>E</td>
<td>Enclosure within which fan is mounted</td>
</tr>
<tr>
<td>L₀</td>
<td>Overall piezoelectric fan length</td>
</tr>
<tr>
<td>L</td>
<td>Length of exposed portion of fan blade (see Figure 3)</td>
</tr>
<tr>
<td>P</td>
<td>Pressure (static)</td>
</tr>
<tr>
<td>P₀</td>
<td>Pressure (static) at zero flow rate condition</td>
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<tr>
<td>Q</td>
<td>Flow rate</td>
</tr>
<tr>
<td>Q₀</td>
<td>Flow rate at zero pressure condition</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>S</td>
<td>Length of inlet opening (see Figure 11)</td>
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Greek Symbols

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<tr>
<td>ω</td>
<td>Fan driving frequency</td>
</tr>
<tr>
<td>ν</td>
<td>Kinematic viscosity of fluid (air)</td>
</tr>
<tr>
<td>ρ</td>
<td>Density of fluid (air)</td>
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</table>

Subscripts

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<tr>
<td>1,2,3</td>
<td>Enclosure number</td>
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Superscripts

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<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>*</td>
<td>Dimensionless quantity</td>
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INTRODUCTION

Piezoelectric fans are cantilever beams whose actuation is made possible through a piezoelectric element typically mounted near the cantilever base. As the input signal frequency approaches the fundamental resonance frequency, large vibrations occur at the cantilever tip. This causes motion in the surrounding fluid, which has been shown to provide enhancements in heat transfer with minimal power requirements. Most previous studies have focused on the heat transfer and flow fields generated by these devices. Some of the earliest such efforts were reported by Toda [1,2] beginning...
in 1979, who discussed simplified models for the air flow and vibration as well as various possible applications for such devices. Açıkalın et al. [3] addressed the feasibility of their use in products by placing fans in an actual laptop and a simulated cell phone enclosure. Experimental results showed a notable increase in heat transfer in both cases. A comparative study between traditional cooling methods and piezoelectric fans was conducted by Açıkalın et al. [4]. They found that piezoelectric fans exhibit significant advantages over axial fans by requiring less power; they were also superior to natural convection heat sinks due to the reduction in volume of the cooling solution. A fundamental study of the heat transfer performance of piezofans, and its dependence on various operating parameters, was conducted by Kimber et al. [5]. The observed behavior was found to share a number of characteristics with impinging jets; correlations were therefore developed based on jet impingement parameters for piezoelectric fans and found to explain the important trends with good accuracy.

These recent studies have increased our understanding of these novel cooling devices. However, in order to include these fans as a viable option for the thermal engineer, a direct comparison to conventional fans is necessary. Their performance must be cast in terms of common metrics used to describe the behavior of traditional (axial flow) fans. One of the most often-used fan metrics is the pressure-flow rate relationship, or the fan curve. This is typically used to assess the suitability of fans for a given application. To the authors’ knowledge, no such data exists for piezofans, and the experimental results presented in this paper are the first of their kind. Because these results are obtained using standardized techniques [6], the performance of piezofans can thus be evaluated against any other traditional or non-traditional fan. Another important contribution of the current study is a determination of the influence of key piezofan parameters (vibration amplitude and frequency) on the overall attainable pressure and flow rate. For conventional fans such as axial fans, guidelines referred to as fan laws suggest how the performance of a certain fan might change with respect to operating variables such as fan diameter and rotation speed. Analogous relationships for piezofans are therefore proposed here, shedding light on the sensitivity of various geometric and operational parameters. In the remainder of this paper, the experimental setup is first explained, followed by a description of the parameters under investigation. The experimental results are then presented, followed by a discussion of the effects of fan installation details on fan performance.

**EXPERIMENTAL SETUP**

The piezoelectric fans used in this work are commercial products with a piezoelectric actuator bonded to a flexible cantilever beam, or fan blade. The driving frequency is tuned to the first resonance frequency of the fan to provide large oscillations. A typical fan is illustrated in Figure 1 depicting the important geometric parameters. The footprint of the device is described by the overall length (L₀) and fan width (D). The largest oscillations occur within the portion of the flexible blade not covered by the piezoelectric material. Therefore, the exposed blade length (L) is also of importance in describing the performance and behavior of these devices.

The experimental apparatus used to quantify the flow rate and pressure characteristics consists of an air-tight chamber and blower designed according to AMCA standard 210 [6]. This standard outlines different approaches for measuring the pressure and flow rate characteristics for fans of many different types. A schematic diagram of the setup is shown in Figure 2, in which the fan under investigation is positioned to direct airflow into the chamber. The static pressure generated by the fan (P) is measured with a pressure transducer (Novasina PascalVision with accuracy ± 0.05 Pa). The flow rate (Q) is determined from an additional pressure drop downstream (P_m) through a mass flow meter (accuracy ± 1 l/min). An auxiliary blower is employed downstream of the mass flow meter to regulate the flow rate through the system and overcome the pressure drop needed for reliable flow rate measurements.

The measurement procedure begins by closing the valve to capture the attainable pressure at a zero flow rate condition (P₀). Next, the valve is opened and the strength of the auxiliary blower is increased until the static pressure in the chamber (P) reaches zero. This point is the flow rate attainable at a zero pressure condition (Q₀). The strength of the blower is then incrementally decreased yielding non-zero pressure readings in the chamber and flow rates smaller than Q₀. Each new setting for the blower represents a point along the pressure-flow rate curve which starts at P₀ and ends at Q₀. For each experiment conducted in this work, a total of eight points are obtained (P₀, Q₀, and 6 intermediate points).

At this point, a brief discussion regarding differences between standard axial fans and piezofans is in order to understand the conditions which should be maintained for the piezofan in the experimental apparatus. An axial fan is typically contained within a casing where a certain gap is maintained between the rotating blade and casing. It is easily understood that the maximum attainable pressure would suffer dramatically if this gap were increased. In order to make a direct comparison to piezoelectric fans, the fan assembly is designed to maintain...
small gaps between the vibrating blade and the casing where the fan is mounted. However, while a single radial gap exists for the standard axial fan, casing-to-fan gaps in orthogonal directions are considered for piezofans, namely the vibration-parallel (dependent on vibration amplitude) and vibration-orthogonal (dependent only on fan width) directions. This is illustrated in Figure 3 where the fan is mounted in an assembly with side walls sufficiently far from the fan (nearly four times the fan width) to remove their possible flow limiting influence. These walls are attached to a cover plate with a small opening in the center. The side and outlet views shown in Figure 3 further illustrate the two outlet dimensions of significance. The piezoelectric fans considered in this work have a width of 12.7 mm. The outlet width for all experiments is 15 mm, which leaves a 1.15 mm gap between the vibrating fan and casing in the horizontal direction. The height of the outlet is slightly larger than twice the vibration amplitude (A), and therefore must change for different amplitudes, even when the same fan is under investigation. When considering a range of amplitudes, this calls for an outlet with a variable vertical dimension. To accomplish this, a single cover plate is used with an outlet size of 15 mm x 25 mm in the horizontal and vertical directions, respectively. The input voltage is adjusted to obtain the desired vibration amplitude, which is measured using a laser displacement sensor (Keyence LK-G150) to capture the vibration signal of the fan tip. For an outlet height of 25 mm, a gap larger than desired exists in the vertical direction, especially when considering small amplitudes. Therefore, thin strips of aluminum are then attached on the cover plate above and below the vibrating tip to provide less than 1 mm gap for the vertical blade-to-fan spacing. As the amplitude is changed, the process begins over and the strips are repositioned to maintain this vertical gap for all experiments.

Figure 2. Illustration of experimental apparatus used to acquire pressure-flow rate measurements.

The assembly is designed such that the fan tip protrudes 1 mm beyond the outlet surface. This protrusion distance was initially considered as an additional variable, but the configuration just described is found to provide the greatest pressure and flow rate. This optimum location is somewhat insensitive to protrusion distance within a tolerance of ± 2 mm, and is used for all the experimental results presented in this paper.

Figure 3. Illustrations of piezoelectric fan assembly used during experimentation. The outlet dimensions for each case are slightly larger than the fan width (D) and two times the vibration amplitude (A) in the horizontal and vertical dimensions, respectively.

Similar to an axial fan in the same experimental setup, the piezofan is allowed to pull air from all directions on the suction side. For implementation into an actual device, these conditions will undoubtedly change as inlet obstructions are introduced from the walls of the device. In order to address fan performance and implementation issues separately, the experiments are divided into two sections. The first section employs the fan assembly previously described with the intention of providing fundamental insight into characterizing the pressure and flow rate from vibrating cantilevers. The two fans considered in this first set of experiments have blades made from different materials (mylar and steel) and are shown in Figure 4. Due to the differences in length and material, the resonance frequency is also different for each fan. The fan dimensions and frequencies are listed in Table 1 along with the vibration amplitudes considered for each fan. Note that both fans have identical widths, and therefore the influence of this parameter cannot be gauged in the current work. The steel fan is shorter, has a higher frequency, and cannot achieve as high of vibration amplitudes when compared to the mylar fan. The maximum amplitude for each fan corresponds to the point where the input voltage is slightly below the depoling voltage for the piezoelectric element (approximately 120 and 30 V_{rms}
for mylar and steel fans, respectively). It is also emphasized that for each experiment, the fans are driven at their first resonance frequency, and claims made regarding this parameter are not necessarily applicable to higher resonance frequencies.

![Steel Blade](Image)

![Mylar Blade](Image)

**Figure 4.** Two piezoelectric fans (mylar and steel blades) tested to characterize the influence of operating parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Mylar Blade</th>
<th>Steel Blade</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall Length ($L_0$)</td>
<td>[mm]</td>
<td>69</td>
<td>47</td>
</tr>
<tr>
<td>Exposed Blade Length ($L$)</td>
<td>[mm]</td>
<td>36</td>
<td>23</td>
</tr>
<tr>
<td>Width ($D$)</td>
<td>[mm]</td>
<td>12.7</td>
<td>12.7</td>
</tr>
<tr>
<td>Frequency ($\omega$)</td>
<td>[Hz]</td>
<td>60</td>
<td>113</td>
</tr>
<tr>
<td>Amplitude ($A$)</td>
<td>[mm]</td>
<td>6.0, 7.5, 9.0, 10</td>
<td>3.5, 4.5, 5.5, 6.5</td>
</tr>
</tbody>
</table>

The second set of experiments addresses installation effects by mounting the fan in three different enclosures as shown in Figure 5. The width and height of these enclosures ranges from $40 \times 80$ mm for the largest ($E_1$) to $15 \times 30$ mm for the smallest ($E_3$), all with identical lengths of 85 mm. These enclosures consist of four walls: two vertical side walls (aluminum) and two horizontal plexiglass walls above and below the fan. The plexiglass allows access for the laser enabling vibration measurements. The back side of the enclosure is left open to allow flow from locations upstream of the fan. It should be noted that as the distance decreases between the vibrating fan and enclosure walls (from both directions), the viscous drag increases. Because a portion of the power input to the fan is then used in overcoming this added drag, the power consumption for a given vibration amplitude is substantially greater for $E_3$ than for $E_1$ (as much as 70% higher in some cases). For reference, the power needed to drive the fans is dependent on the vibration amplitude and ranges from 10-30 mW and 20-45 mW for the mylar and steel fans, respectively. These values represent power consumption with fans mounted in the configuration illustrated in Figure 3, and remain relatively unchanged when the fans are mounted in $E_1$. Any increase in power reported is in reference to these values. Further testing is needed to determine the vibration amplitude as a function of the proximity of surrounding walls for a given power input. Results in this work are presented at a specified amplitude.

Experimental uncertainty is introduced from both pressure and flow rate measurements. The largest source of error is in the flow rate resolution ($\pm 1$ l/min) which results in an uncertainty of less than 10% in flow rate estimation, while the pressure measurement uncertainties are below 5% (device with $\pm 0.05$ Pa resolution). Additional uncertainties are noted for vibration amplitude measurements, but are small (0.05%) when compared to those of pressure and flow rate.

![Enclosures](Image)

**Figure 5.** Three enclosures ($E_1$, $E_2$, $E_3$) with cross-section dimensions of (a) $40 \times 80$ mm, (b) $25 \times 50$ mm, $15 \times 30$ mm used to determine installation effects for piezoelectric fans.

**EXPERIMENTAL RESULTS**

As previously described, the experiments in this work are intended to address two different aspects related to piezofans. The first is to characterize the fan performance given the operating parameters, and the second is to determine the effect of the proximity of enclosure walls to the piezofan. The experimental results are presented in this order.

**Fan Characterization**

Using the fan assembly illustrated in Figure 3, the experimentally obtained pressure-flow rate data are shown in Figure 6 for mylar and steel fans at four different amplitudes each. For all cases the pressure is highest at the zero flow rate condition and then decreases monotonically until the maximum flow rate is reached at the zero pressure condition. As expected, the maximum pressure and flow rate both increase with amplitude. The family of curves for either fan type is directly analogous to fan curves for traditional fans at different voltage levels. The only difference is that an increase
in voltage for an axial fan causes the rotation speed to increase, whereas for a piezofan, the result is an increase in vibration amplitude. The attainable pressure is a strong function of the type of fan considered, with the steel fan resulting in much higher pressures than the mylar fan. This is true despite the fact that the vibration amplitude is much smaller for the steel fan; however, the frequency is greater for the steel fan by almost a factor of two (see Table 1). On the other hand, the maximum flow rate appears to be almost independent of fan type. Although the mylar fan shows a slight advantage (of roughly 20%), the range over the given amplitudes is approximately the same for either fan (8-28 l/min). The general relationship between pressure and flow rate is readily observed by normalizing each curve with its respective maximum values. This result is shown in Figure 7, and suggests that the P-Q curve could be estimated with an equation of the form:

\[
\frac{P}{P_0} = 1 - \left(\frac{Q}{Q_0}\right)^q
\]

(1)

where \(P_0\) and \(Q_0\) are different for each experiment and depend on the operating parameters and fan geometry. A least-squares analysis of each curve in Figure 6 yields a value for the exponent of Eq. (1) ranging from 1.2 – 2.0. This suggests the dependence of pressure on flow rate is somewhere between linear and quadratic, depending on the vibration amplitude and frequency. The mean value for this exponent is 1.6 and as shown in Figure 7 reasonably describes most of the data. The mean and maximum deviations are 5.6 and 18.9%, respectively.

![Figure 7. General P-Q curve represented by Eq. (1) obtained from normalizing data in Figure 6 by \(P_0\) and \(Q_0\).](image)

![Figure 8. Behavior of maximum (a) pressure, and (b) flow rate of the steel and mylar fans with respect to vibration amplitude.](image)
The next task is to express the quantities $P_0$ and $Q_0$ in terms of the variable parameters $A$, $\omega$, and $L$. There are two frequencies, two lengths, and eight total amplitudes to consider. The maximum pressures and flow rates with respect to vibration amplitude are shown in Figure 8(a) and (b), respectively. For either graph, the performance increases with amplitude and the observed trends of variation are similar for both types of fans. As previously mentioned, the range of flow rate seen for the two fans is roughly the same, while the range of attainable pressures is higher for the steel fan. The effect of a change in frequency for a specified vibration amplitude is seen by comparing the largest amplitude for the steel fan (6.5 mm) and the smallest amplitude for the mylar fan (6.0 mm). An approximate four-fold increase in pressure and three-fold increase in flow rate are observed for the steel fan at this amplitude, suggesting a large dependency on frequency and/or length. Next, the data is further analyzed to quantify the influence of fan geometry and operational parameters on the performance and provide design guidelines.

**Design Guidelines**

In an effort to generalize the fan performance, appropriate dimensionless pressures and flow rates are defined. Excluding any variation in fluid properties and assuming the beams under investigation are thin (where the thickness of the blade is not significant), there is one geometrical parameter ($L$) and two operational parameters ($A$, $\omega$) which can be utilized to explain the differences seen in the experimental trends for the two fans considered. While the materials from which the two fan blades are made are also different, it can be assumed that the resulting motion in the fluid remains unchanged if all other variables are held constant, since the loading of the fluid is only a function of the motion of the fan. The power needed to drive the fan would, however, depend on the blade material. It should be noted that the operational frequency is also closely tied to the material as well as the length and thickness of the beam (and the piezoelectric actuator properties).

The Reynolds number ($Re$) is defined based on the maximum tip velocity ($\omega A$) and width of the vibrating beam and is expressed as:

$$Re = \frac{\omega AD}{v} \tag{2}$$

Dimensionless pressure and flow rate can be obtained using:

$$P_0^* = \frac{P_0}{\rho \omega^2 AD} \tag{3}$$

$$Q_0^* = \frac{Q_0}{\omega ADL} \tag{4}$$

The resulting experimental values for these dimensionless parameters are plotted against $Re$ in Figure 9. Data from the two different fans collapse to a single curve each for dimensionless pressure and flow rate. In each case, a power-law correlation describes the data with the following equations:

$$P_0^* = 0.070(Re/1000)^{0.561} \tag{5}$$

$$Q_0^* = 0.095(Re/1000)^{0.888} \tag{6}$$

Both $P_0^*$ and $Q_0^*$ tend to zero as $Re$ approaches zero, i.e., when the frequency or amplitude approaches zero. Using Eqs. (5) and (6) in conjunction with Eqs. (2) – (4), it is possible to determine the relative influence of a number of geometric and operating parameters. This gives rise to design guidelines, and suggests the following scaling for maximum pressure and flow rate:

$$P_0 = \omega^{2.6} A^{1.6} \tag{7}$$

$$Q_0 = \omega^{0.9} A^{1.9} L^{0.9} \tag{8}$$

The form of these guidelines and the manner in which they would be used is similar to fan laws [7] often used for conventional fans to compare the performance of two geometrically similar fans. For piezofans, it is interesting to note that flow rate scales in an identical manner with frequency and amplitude. Generally speaking for cantilevers vibrating in their first resonance mode, a trade-off exists between these two quantities: as the frequency increases, the attainable amplitude decreases. The current work suggests that these quantities should not be considered in isolation, but rather, it is their product (the tip velocity) which should be maximized in order to obtain the highest flow rate. By this reasoning, if the tip velocity were doubled, a nearly four-fold increase in flow rate would result. For describing the pressure, the frequency has a stronger effect than amplitude. Thus, in situations where pressure is the primary factor to increase, focus should be made on the frequency, rather than the amplitude. For example, an increase of only 10% in frequency would cause a 28% increase in pressure, while the same increase in vibration amplitude (10%) would result in a pressure only 16% higher.

It is emphasized that these guidelines are based upon data from two fans, both of the same width. The influence of the fan width should also be investigated along these lines. It may be expected that when the width is doubled, the flow rate would double as well, but more research is needed considering fans of different widths.
Installation Effects

The factors which should be considered when implementing these devices into actual products are now explored. The effect on the pressure and flow rate as the enclosure around the fan becomes smaller is investigated. It is also important to understand the locations from where the fan draws the inlet flow (i.e., determining the location of the effective “inlet”). Only the mylar fan is considered in this section for the same amplitudes as in Table 1. It is assumed that the general trends are consistent for other geometrically similar fans as well.

I. Enclosure Influence

As walls are brought closer to the piezofan from any direction, the power input to the fan needed to maintain a given amplitude increases due to additional viscous drag. As previously mentioned, it is most instructive to present the data in terms of vibration amplitude rather than input voltage as the dependence of amplitude on the proximity of the walls can change with different fans. For the mylar fan mounted within an enclosure, the power requirement increases 2, 10, and 40% for the E1, E2, and E3 enclosures, respectively (see Figure 5), when compared to the power needed with no enclosure. Therefore, when the enclosure is sufficiently large, the vibration behavior is relatively unchanged (only 2% increase in power for E1). On the other hand, for the smallest enclosure (E3), the power increase is large enough that the attainable amplitude is limited so that only the smallest two vibration amplitudes (6.0 and 7.5 mm) can be achieved due to the depoling voltage of the piezoelectric element. Additional experiments are performed using this same enclosure with the plexiglass walls above and below the fan removed. This enclosure configuration is referred to as E3*, and is similar in form to the fan assembly illustration shown in Figure 3, except the distance between the side walls is now drastically reduced to 15 mm (the fan width is 12.7 mm). This configuration eliminates a large portion of the added drag previously encountered due to the enclosure walls. As a result, the voltages required for the specified amplitudes once again become reasonable.

The most illustrative results are those for the maximum pressure and flow rate, as shown in Figure 10 (a) and (b), respectively. Also shown for comparison are data initially shown in Figure 8 for the mylar fan with no enclosure. Any deviation in pressure or flow rate from this set of data in either plot signifies the effect of that particular enclosure. For the large enclosure (E1), the pressure remains relatively unaffected, especially at lower amplitudes. As the amplitude increases, the curve for E1 starts to deviate, suggesting that the size for the enclosure to remain non-intrusive must be bigger for larger amplitudes. Pressures with enclosure E2 reveal the adverse effect of confinement, which is again more pronounced as the amplitude is increased. At the largest amplitude (10.5 mm) in E2 the attainable pressure is reduced by nearly a factor of two. Enclosure E3 behaves very similarly to E2, suggesting that the reduction in pressure has leveled off. Only the smallest two amplitudes could be tested in E3. The most intriguing behavior occurs for the E3* configuration where the walls above and below the fan are removed. The pressure has now increased substantially, providing a 30% advantage over the case without an enclosure. For an axial fan, greater pressures at the same rotational speed imply a tighter pressure seal. A similar conclusion may be drawn regarding piezofans, suggesting that with sidewalls so close to the vibrating fan and top and bottom walls removed, the result is a tighter pressure seal.

Figure 10. Effect of various enclosure conditions for maximum (a) pressure, and (b) flow rate. E3(no T/B) refers to the condition with the fan mounted in enclosure E3 with the top and bottom walls removed.

The flow rate is less sensitive to surrounding walls as seen in Figure 9(b), only showing a deviation for the smallest enclosure. However, in the case of E3*, the flow rate again matches the other cases. This suggests that the attainable flow rate is indeed a function of operating and geometric parameters of the fan only. It is argued that enclosure E3 limits the flow by removing some of the inlet area. This effect is treated in more detail in the next section, but implies that the flow primarily enters from above and below the vibrating fan as opposed to lateral or upstream locations of the fan. Therefore, the proximity of the side walls has little impact on the attainable flow rate. An experiment where walls slowly
approach the vibrating fan from above and below would be analogous to experiments often conducted with axial fans where a flat plate placed upstream of the fan slowly approaches the inlet. Just as in the case of axial fans, it is expected that there is a certain distance below which the flow rate begins to be affected and experiences a sharp drop.

II. Inlet Considerations

With the intention of providing insight into significant parameters regarding the inlet side of the fan, additional experiments are performed using a slightly modified $E_3^*$ configuration. From the previous section, it is suggested that the “effective inlet” location for a piezofan can be considered directly above and below the vibrating cantilever. The two different configurations used for these inlet experiments are shown in Figure 11 where the length of the inlet opening ($S$) is measured from the fan tip and represents the portion of the top and bottom walls which are removed. Air can be pulled through this opening as well as from upstream of the fan for the configuration Figure 11(a). For the second configuration (Figure 11(b)), the upstream side of the fan is sealed off, thereby allowing airflow through only the top and bottom openings. The first and second configurations found in Figure 11(a) and (b) are referred to as inlet$_A$ and inlet$_B$, respectively. A vibration amplitude of 9 mm is maintained for all the inlet experiments.

Results from these experiments are shown in Figure 12 (a) and (b) for the maximum pressure and flow rate, respectively. For the largest inlet opening ($S = 32$ mm), both the pressure and flow rate are approximately equal to values obtained from the $E_3^*$ configuration. The impact of the upstream blockage is minimal as well, suggesting that nearly all the flow for this inlet size is drawn from above and below the vibrating fan. It is important to note that the size of the inlet in this case is nearly equal to the exposed blade length of the mylar fan under investigation (see Table 1). It appears that if the top and bottom walls do not cover blade locations experiencing large vibration amplitudes, the performance remains relatively unchanged. As the inlet opening becomes smaller, the top and bottom walls begin to cover more of the exposed blade. The adverse effect of this is observed in Figure 12 in both the attainable pressures and flow rates. The effect of upstream blockage also has a larger influence when the inlet is small. For example, for the smallest inlet opening ($S = 9$ mm), a 35% decrease in flow rate occurs in the presence of the upstream blockage relative to the no-blockage case. A general guideline that emerges is that any top and bottom walls should not be placed near portions of the fan which experience large vibrations. If this is not an option for a particular application, such walls should be positioned sufficiently far from the fan, at a distance of 4-5 vibration amplitudes for the flow rate to remain unchanged, and slightly farther (6-7 vibration amplitudes) for the pressure to remain unchanged.

CONCLUSIONS

This paper presents experimental measurements of the pressure and flow rate characteristics of piezoelectric fans. This work enables a direct comparison of these devices to any fan where the fan curve is known. Preliminary design guidelines are developed from which the relative influence of a number of important parameters can be determined. The attainable flow rate is found to have a nearly quadratic dependence on the tip velocity, while the vibration frequency
is found to be extremely influential in determining the attainable pressure. The installation effects can be summarized as follows:

1) A large enclosure (E₁) has relatively little influence on the pressure or flow rate.
2) As the size of the enclosure becomes smaller (E₂), only pressure is adversely affected.
3) There is a certain enclosure size (between E₂ and E₃) where the flow rate begins to be affected and a sharp drop in performance is observed.

In addition, the inlet flow is found to be drawn primarily from above and below the portions of the vibrating fan experiencing the largest amplitude and these portions should remain uncovered to ensure the largest flow rate possible. The fan appears to change the flow path by nearly 90°.

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REFERENCES