Aerodynamic performance of a vibrating piezoelectric fan under varied operational conditions

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Abstract. This paper experimentally examines the bulk aerodynamic performance of a vibrating fan operating in the first mode of vibration. The influence of operating condition on the local velocity field has also been investigated to understand the flow distribution at the exit region and determine the stalling condition for vibrating fans. Fan motion has been generated and controlled using a piezoelectric ceramic attached to a stainless steel cantilever. The frequency and amplitude at resonance were 109.4 Hz and 12.5 mm, respectively. A test facility has been developed to measure the pressure-flow characteristics of the vibrating fan and simultaneously conduct local velocity field measurements using particle image velocimetry. The results demonstrate the impact of system characteristics on the local velocity field. High momentum regions generated due to the oscillating motion exist with a component direction that is tangent to the blade at maximum displacement. These high velocity zones are significantly affected by increasing impedance while flow reversal is a dominant feature at maximum pressure rise. The findings outlined provide useful information for design of thermal management solutions that may incorporate this air cooling approach.

1. Introduction

Recent trends in thermal management of electronics using air cooling approaches have demonstrated the need for technologies that provide enhanced heat transfer performance at higher efficiency and reliability than the current technologies in widespread use. This has led to an improvement in conventional rotational fan designs and also the emergence of solid-state air movers such as synthetic jets and vibrating fans [1,2].

In the late 1970’s, Toda [3] and Toda and Osaka [4] presented the concept of using a vibrating fan, driven by a piezoelectric polymer, to induce an airflow that could be used for electronics cooling or ventilation. The vibrating fan was compared to a small rotational fan, and it was noted that the solid-state piezoelectric fan required 1/7 of the power input to achieve similar exit velocities. Despite the potential of piezoelectric vibrating fans as an air cooling technology identified by Toda and Osaka [4], it was twenty years later when more intensive research into these air movers began. Yoo et al. [5] presented a number of piezoelectrically driven vibrating fan designs as a solution to remove electromagnetic noise for sensitive electronic systems that require forced cooling. The cantilever parameters of length and material were varied and it was noted that phosphor bronze provided the highest performance over aluminium and brass.

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Previous research into vibrating flexible cantilevers as air movers suggests that such devices can provide an increase in local heat dissipation [6] and also augment heat transfer due to auxiliary air flows [7]. Kimber and Garimella [6] measured the local heat transfer coefficients on a flat surface cooled by a piezoelectric fan. The vibrating fan was positioned vertically and normal to the flat surface for various gap distances between the surface and fan tip. Heat transfer distributions were shown to vary from a lobed pattern at close gaps to elliptical as the gap distance was increased. Local heat transfer coefficients were of order 100 W/m²K demonstrating the potential use of vibrating fans for improving discrete cooling applications. Lin [7] considered the use of a vibrating fan in the wake region behind a cylinder in cross flow. The aim of this approach was to investigate the potential for improving the local heat transfer due to the adverse impact of the wake region of a cylinder in cross flow. A twin jet like structure was observed and for a range of operational criteria including fan vibration amplitude and cylinder Reynolds number, an increase in heat transfer was apparent in the wake region. However, it was also noted that variation of these criteria beyond certain limits can increase the wake region and ultimately reduce the local heat transfer performance over the baseline case.

Bulk cooling through the implementation of piezoelectric fans with finned heat sinks has also been analysed both experimentally [8-10] and numerically [8]. Abdullah et al. [8] and Shyu and Syu [10] both examined the potential of integrating piezoelectric fan arrays within the fin gap of parallel fin heat exchangers to reduce the cooling solution volume. In both studies, enhancement in heat transfer over natural convection was found to be above a factor of 1.5 and up to 3.3. Li et al. [9] experimentally examined the flow field and heat transfer due to positioning a piezoelectric fan adjacent to parallel fin heat sink geometries. The study focused on the position of the fan relative to the heat sink, and it was concluded that a horizontally orientated fan should be placed at the entrance to the heat sink for increased thermal performance.

The previous studies [6-10] examined heat transfer characteristics due to piezoelectric fans with the intended application of electronics cooling. The majority of research studies on piezoelectric fans have focused on the velocity and heat transfer performance in an environment free of the global constraints that exist within a practical system. However, the high density of contemporary electronic systems implies that these fans are likely to experience flow resistances that are significantly greater than this type of operating condition. There is limited research in this area, despite the low pressure-flow characteristics of vibrating fans [11] and the higher sensitivity to confinement compared to rotational fan designs. Kimber et al. [11] investigated the pressure-flow characteristics of two piezoelectric fans to determine the relationship between aerodynamic performance and the frequency and amplitude parameters. The bulk measurements were conducted using conventional fan performance characterisation standards and provided a comparison between an axial fan of similar scale. The axial fan provided superior pressure rise and lower efficiency than the piezoelectric fan under consideration. Although this study highlights the typical performance characteristics of piezoelectric fans, the recommended operating region and occurrence of stall effects have yet to be investigated in the literature.

In addition to bulk pressure-flow information, understanding the momentum field generated by air movers under varied operational conditions is important to ensure optimum performance is achieved for a combined fan – heat exchange solution. Egan et al. [12] demonstrated increased heat dissipation of up to 20% by aligning miniature centrifugal fan exit flows with parallel fin configurations. This was achieved using velocity field information for the combined cooling solution. Grimes et al. [13] highlighted the change in swirl characteristics at the exit region of an axial fan air mover with changes in operating pressure. Higher operating pressures result in an increase in the radial velocity component, and therefore alter the flow distribution for a convective heat exchange system. Considering both of these studies, an optimum fan - heat exchanger arrangement for maximal heat dissipation is dependent on knowledge of the air mover response to the system characteristics, or operating condition.
The aim of the current study is to determine the effect of operating conditions on the exit flow field of a vibrating piezoelectric fan. Using this information, the occurrence of stall for this type of air moving device will be examined. The stall region has been defined as the region along the pressure-flow characteristic where flow reversal into the blade passage occurs. Combined with the bulk aerodynamic performance characteristics, this information will be used to determine the recommended operating region for vibrating fans. Therefore, the results presented in the current study will assist in the optimisation of thermal management solutions incorporating vibrating fans subjected to practical system conditions.

2. Experimentation
In this section the experimental testing facility designed and implemented for the measurement of bulk pressure-flow characteristics and velocity fields in the fan exit region is presented. Geometric and material information is also provided on the vibrating fan used in the current investigation.

2.1 Pressure-flow characteristics
The measurement of pressure-flow characteristics was achieved by developing an experimental facility based on BS 848 [14] standards outlined for performance testing of fans. This facility is shown in figure 1 a) and was designed to span a wide range of fan performance characteristics that can range from 0 – 1000 Pa in static pressure rise, and 0 – 0.04 m³/s in volumetric flow rate. In addition to the design guidelines recommended by the standards [14], a refinement was carried out using computational fluid dynamics to ensure that a suitable auxiliary fan was selected and flow conditioning in the test chamber was acceptable for the inflow to the test air moving device.

Measurement of static pressure rise was achieved through settling chamber pressure taps, and volumetric flow rate was indirectly determined by measuring differential pressure across an orifice plate and using a pre-calibrated relationship between the orifice pressure drop and flow rate. Accuracy in the measurement of differential pressure across a wide range was achieved using three pressure transducers which spanned three different scales. A Novasina PascalSwitch transducer was used for 0 – 20 Pa range with uncertainty of +/-0.05 Pa for < 4 Pa and +/- 1.5% of measured value for > 4 Pa. For 20 – 200 Pa, a Novasina Pascal-STV200 Z instrument was used with uncertainty of +/-0.2% of full scale. Finally, three Freescale semiconductor MPXV5004DP transducers were used for measurements > 200 Pa and calibrated to +/- 10 Pa. For the vibrating fan performance characteristic presented in this study, the maximum uncertainty in static pressure rise and volumetric flow rate are +/-0.068 Pa and +/- 0.6 x 10⁻⁵ m³/s. For all experiments, the ambient air temperature was constant at 21.5°C +/- 0.1°C.

An enhanced view of the air moving device arrangement is illustrated in figure 1 b). The measurement facility is based on an inlet-side design, where air is moved from the settling chamber and expelled to the environment. The vibrating piezoelectric fan was positioned within the settling chamber and anchored to a support structure. This design maintained rigidity and ensured the electrical energy input to the piezoelectric ceramic was used solely for the mechanical vibration of the cantilever fan. The support structure had cross-sectional dimensions 11mm x 3mm and the edge of the cantilever was 15mm from the support leg. The structure dimensions were kept to a minimum and the distance of both legs from the fan was maximized to remove confinement effects that could be detrimental to the fan aerodynamics. Table 1 contains the geometric and material properties of the vibrating piezoelectric fan used for the current study. The fan was operated at the first vibration mode which resulted in maximum displacement amplitude, or tip deflection. A TTi TG1000 function generator supplied a sinusoidal waveform which was amplified to provide an electrical input to the piezoelectric ceramic of 50VRMS. The outflow region, defined by dimensions w and h in figure 1 b), was larger than the fan width and displacement amplitude by 0.5 mm to allow clearance for the fan motion. The free end of the cantilever was centred and coplanar with the outflow region when in the unexcited state (zero electrical power to piezoelectric ceramic).

It is anticipated that the outlet geometrical configuration described in figure 1 b) will also have some impact on the performance of the vibrating fan compared to a fan in free space, or due to other geometrical confinements. This was also noted by Kimber et al. [11] who measured bulk performance
Figure 1. a) Apparatus for the characterisation of air moving devices and b) a detailed view of the vibrating fan installation.

Table 1. Details of the vibrating fan.

| Vibrating fan parameters |    |
|--------------------------|----|
| Length                   | 48 mm |
| Height                   | 12.5 mm |
| Blade material           | Stainless steel |
| Blade thickness          | 75 μm |
| PZT patch length         | 24 mm |
| Input waveform           | Sine wave |
| Frequency                | 109.4 Hz |
| Displacement amplitude   | 12.5 mm |

using an outlet-side characterisation method. The dependence of performance on installation is a difficulty if the aim is to determine a generic fan performance characteristic for vibrating fans or one which is as robust in practice as for rotational fan designs. However, by examining the performance of different air movers in a similar test environment [14] relative comparisons can be useful for thermal design decisions.

2.2 Local velocity field
Local velocity field measurements were recorded using particle image velocimetry (PIV) on the exit flow field simultaneously with the bulk aerodynamic performance characteristics described previously in section 2.1. The experimental configuration is shown in figure 2. A laser sheet with 532nm wavelength and 7.5Hz pulse frequency was generated using a Quantel Big Sky Laser. The laser sheet illuminated glycol-based tracer particles < 10μm in diameter that were introduced into the test environment using a Magnum 2500 hazer. A homogeneous particle distribution was achieved after approximately 10 minutes of continuous issuing of tracer particles. This period of time was necessary to ensure sufficient seeding was achieved both internally and externally to the settling chamber. Once
sufficient tracer particle density was achieved, the hazer was turned off for 1 minute prior to recording velocity field data. This approach ensured that air currents unrelated to the vibrating fan motion in the test environment were alleviated.

The laser illuminated the tracer particles twice in succession for the plane of interest. A single plane was considered for the purpose of investigating the impact of operating conditions. This plane was located at the centre of the blade height in a region where the fluid motion exiting the fan is dominant. Timing between pulses was adjusted based on flow velocities at each operating point considered. A TSI PowerView Plus 4MP CCD camera was positioned perpendicular to the laser sheet plane to record the scattered light from the particles on two separate frames for the region of interest illustrated in figure 2. The camera and laser were synchronized using a TSI laser pulse synchronizer. Each image pair was split into interrogation regions that were 32 x 32 pixels and processed using TSI Insight 4G software. The resulting spatial resolution of the vector field was 245 μm.

Fixed errors for the PIV system were estimated to produce a relative uncertainty in velocity magnitude of < 2% [15]. A total of 2000 image pairs were recorded for each operating condition of interest. Based on the vibration frequency of the air mover and the acquisition rate of the PIV system, a random sampling assumption is valid. The method presented by Stafford et al. [15] was used to analyze convergence of full field measurement statistics. It was determined that the ensemble average velocity field is sufficiently converged towards the time-average within 3.24% using this sample size.

The vibrating fan is a composite of PZT and stainless steel materials with epoxy bond layers. Vibration is induced at the lower section of the cantilever, where the rigidity differs from the more flexible stainless steel free end of the fan. Therefore, the motion in space and time is dependent on the material and geometric properties of the cantilever. Due to this complexity and importance for the fan aerodynamics, the shape of the vibrating cantilever has been measured over the cycle period. A TSI PowerView Plus 2MP camera was used to measure the shape of the flexible cantilever at the maximum vibration amplitude to assist in discussing the resultant vector field. This test was conducted using ambient lighting and the exposure of the camera was adjusted to ensure the shape of the fan path was captured on a single frame.

3. Results and discussion
The pressure-flow characteristic for the vibrating piezoelectric fan is presented in figure 3. The data highlights the low pressure rise and flow rate that is typically achieved using vibrating piezoelectric fans, particularly when compared to small scale rotational fans [16,17]. Volumetric flow rate monotonically increases from the point of maximum static pressure rise to zero static pressure rise. The shape of the characteristic curve is similar to that observed by Kimber et al. [11] for other piezoelectrically actuated vibrating fans with different material and geometric properties. Figure 3 has no distinctive stalling dip that usually occurs in static pressure curves for rotating axial fans. This is...
expected, as the static pressure rise generated by a vibrating piezoelectric fan is not produced by airfoil lift as in rotating axial fans. Air is moved by the oscillation of a simple blade structure, and comparisons are better suited with the aerodynamic characteristics of centrifugal-type fans. Static pressure for centrifugal fans is generated through the centrifugal action, and various blade designs produce a pressure-flow characteristic that is void of an obvious stalling point [18]. However, these air movers do experience similar stall symptoms as there is a static pressure region which results in flow separation and reversal in the blade passages of the fan. In the current study, the characteristics leading to stall for vibrating fans will be discussed using the simultaneous local velocity field measurements. The occurrence of stall has been defined as the point when flow is reversed and the vibrating fan begins to entrain air from the fan exit region.

Three points along the characteristic curve of figure 3 have been highlighted and labelled as b), c) and d). These points correspond to the vibrating fan operating at maximum, medium and zero pressure rise conditions. The local velocity magnitude and vector field at the exit region of the vibrating fan is presented in figure 4 for each of these operating points.

Figure 4 a) is a measurement of the path taken by the vibrating fan over a full cycle, scaled to the velocity field axes. The piezoelectric ceramic patch and stainless steel regions of the cantilever are defined and the influence of this anisotropic design is evident in the shape at maximum amplitude. The ceramic is displaced almost linearly from the anchor point, whereas displacement of the flexible stainless steel region is characteristic to the motion of an isotropic cantilever beam. The combination of figure 4 a) and b) provide an insight into the relationship between the fan motion and the resultant velocity field at the exit. High velocity zones correspond to the maximum displacement of the vibrating fan and are apparent for all operating points in figure 4. In this measurement plane, these zones form a time-averaged v-shaped flow structure.

Figure 3. Performance characteristic for a vibrating fan operating at a resonant frequency of 109.4 Hz and amplitude of 12.5 mm. Operating region (OR) has been outlined based on the flow field analysis.
Figure 4. Measured fan modal shape a) and ensemble averaged velocity fields at the exit region for various operating conditions b), c), d) spanning the performance characteristic shown in figure 3.

The time-averaged distributions in figure 4 are due to the oscillating behaviour of the piezoelectric fan. During this motion, a pressure differential is generated across the fan surfaces, producing two counter-rotating vortices that roll over on the suction side of the fan and has been discussed previously in the literature by Lin [19]. Simultaneously, a negative pressure gradient exists between blade tip to root due to the difference in blade tip and root velocity. This static pressure gradient has been confirmed by Choi et al. [20]. Collectively, this provides the formation and development of a tip vortex which is forced to detach when the fan motion goes to zero velocity at maximum displacement [20]. It is this release of high momentum fluid at the maximum displacement which produces the time-averaged behaviour observed in figure 4 and the stable pressure-flow curve in figure 3. This is confirmed by examining the velocity vectors in the high momentum regions which are tangent to the fan blade profile at maximum displacement. A recent experimental study by the authors on the three-dimensional velocity field surrounding a vibrating piezoelectric fan in a free field also provides further detail on the local flow behaviour in the blade passage region [21].

Although the location of the high velocity zones is similar for figure 4 b) – d), there are discernible differences in the flow fields due to varied operating conditions experienced by the vibrating fan. At zero pressure rise and maximum flow rate (d), air exits radially outwards with two high velocity zones as previously discussed. Utilization of this distribution of momentum is important, as it influences the location and geometry of heat exchanger structures in the design for maximal heat...
dissipation. For example, heat sink structures could be aligned with the flow as this approach has been found to increase heat dissipation in forced convection heat sinks at miniature scales [12].

As static pressure is increased to just above 50% of $\Delta P_{\text{max}}$ (c), two weak recirculation zones emerge in the low velocity region ($< 0.7 \text{ m/s}$) of the vector map. These are formed by the interaction between the high velocity flow exiting and the entrainment of ambient fluid from outside the test chamber due to flow reversal. At this point the onset of stall begins, and the vibrating fan is able to provide approximately 60% of $Q_{\text{max}}$. The increase in static pressure also alters the shape and increases the angle between the high velocity zones when comparing figure 4 c) and d). This may be a similar effect to that observed for conventional rotational fans [13] however further investigation is necessary to confirm.

In figure 4 b), the vector field resultant from operating at maximum pressure rise and zero flow rate is presented. At this operating point, the vibrating fan is fully stalled and flow reversal dominates the flow field. The net volumetric flow rate is zero; therefore the original exit flow region is performing two functions as both an exit and inlet. Air is entrained from the external environment at $x = 0$ and $-0.005 \text{ m} < y < 0.005 \text{ m}$ and then expelled at the fan tip regions. This also produces two recirculation zones that are centered in close proximity to the fan tip, compared to that observed for the point when stall is initiated in figure 4 c). In addition to a significant reduction in volumetric flow rate, the flow reversal associated with operating a vibrating fan in the stall region is of considerable importance for thermal management applications. During stalling, there is a risk that high temperature fluid could be entrained from downstream heated bodies and circulated within the electronic system to heat nearby sensitive components. However, this flow reversal effect could also be used to benefit the thermal system if applied correctly at the initial design stages.

Both the bulk aerodynamic performance characteristics and local velocity field analysis have been used to determine the initiation of stall. This has been used to provide a recommended operating region (OR) for vibrating fans. The operating region is defined in figure 3 and extends to 50% of the static pressure rise. Operating in this region of the pressure-flow characteristic curve ensures that the vibrating fan avoids stalling effects.

4. Conclusions
The aerodynamic performance of a vibrating piezoelectric fan has been experimentally investigated to understand the impact of operating conditions on the exit flow field and determine the stalling characteristics of this type of air moving device. A custom fan characterisation apparatus has been designed for the measurement of the vibrating fan pressure-flow rate performance characteristics. This apparatus also enabled simultaneous measurement of the velocity field at the exit flow region of the vibrating fan using particle image velocimetry. The bulk aerodynamic performance assessment demonstrated the stable pressure-flow characteristic that vibrating fans generate. However, time-averaged exit flow fields have been shown to alter from the zero to maximum static pressure rise performance range. At maximum static pressure, flow reversal dominates the flow field. The initiation of flow reversal at the fan exit, classified as the onset of stall, was found to occur at approximately 50% of the maximum static pressure. The local velocity field observations have been used to outline a recommended operating range and also provide valuable information for the design of thermal management solutions incorporating vibrating fans.

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