Feasibility and design study of a frictionless air mover for thermal management of electronics

R. Schacht¹, A. Hausdorf¹, B. Wunderle²

¹ Brandenburg University of Technology Cottbus-Senftenberg
Grossenhainerstrasse 57, Senftenberg, D-01968, Germany
² Chemnitz University of Technology
Reichenhainerstrasse 70, Chemnitz, D-09126, Germany

E-mail: ralph.schacht@b-tu.de

Abstract. A frictionless air mover concept is introduced in this paper. As opposed to a piezoelectric driven fan, the air mover is based on a flexible blade whose vibration is driven by means of a magnetic field. The blade is based on a polymer material. The paper presents the results of a feasibility analysis and an on-going comprehensive design study. The performance of the prototype amounted to 65% of a comparable piezoelectric fan. To enhance the performance, two different blade materials were investigated, as well as the influence of the coil shape and value. A further goal is to reduce the size and to investigate the influence of a casing. The design study resulted in a prototype of size of 50 x 14 x 35 mm² including a casing. The performance could be doubled, to attain a volumetric flow rate of $\dot{V} \sim 14$ l/min and a static pressure of $p_{\text{stat}} = 3$ Pa.

1. Introduction
Reliable miniaturized air movers (Mean Time To Failure, MTTF > 100 000 h) with a small form factor are of interest as they can enhance heat transfer by increasing the fluid circulation in regions which are otherwise thermally not manageable. Commercially available miniaturized axial or radial fans use ball bearings that degrade because of friction and have an MTTF < 14 000 h [7]. Frictionless fans like piezoelectric fans or synthetic jet drivers should have lower degradation but, in case of the piezoelectric blade structure, its MTTF is given as > 4 600 h [7], because of the degradation of the adhesive layer between the polymer blade sheet and the piezoelectric material. Additionally, it needs a driving (high) voltage of > 40 V and its price, compared to an axial or synthetic jet, is approximately six times higher (~ USD 180) [2][7]. Synthetic jet fans have an MTTF > 100 000 h, but their form factor and weight (Ø 60 mm, 125 g), compared to a piezoelectric fan (60x 15x 30 mm², 10 g), are larger [4][9].
The approach of this paper is to combine the advantages of a blade structure from a piezoelectric fan and the magnetic field excitation mechanism from a synthetic jet, resulting in a frictionless blade air mover concept that is more wear-resistant.
A first feasibility study has shown good potential for low form factor, low weight and low cost [1]. A comparison between a piezoelectric driven fan [3] and the magnetic field driven air mover introduced in [1] has shown that for the equal blade dimensions (length $L = 70$ mm, height $h = 12$ mm, thickness $t = 0.1$ mm) the magnetic field driven air mover had a 65% lower volume flow rate ($\dot{V} = 10.5$ l/min).
than the piezoelectric driven fan ($\dot{V} = 27.5$ l/min), and the energy consumption was \(\sim 97\%\) higher ($P_V = 1$ W in comparison with 0.03 W).

The overall aim of this design study is to reduce the form factor towards the size of a matchbox (52 x 15 x 35 mm$^3$). Further objectives are to increase the volumetric flow rate and the static pressure, and to reduce the power loss.

To achieve those goals, this paper will mainly target the following variables:

- Volumetric flow rate, $\dot{V}$
- Static pressure, $\rho_{\text{stat}}$
- Power loss, $P_V$
- Form factor / cabinet size.

First, this paper introduces the frictionless air mover design and its working principle. Secondly, two experimental set-ups are described, which were used to gauge the air mover variables and to demonstrate its cooling performance. Finally, the experimental details, the results, an optimized prototype and a discussion are given.

2. Frictionless blade air mover design

The blade air mover introduced in [1] consists of a polymer blade, which is fixed at one end. A magnetic steel clip is plugged on the blade, as shown in Figure 1a. Two magnetic rod core shaped coils are assembled on a base plate in position of the steel clip (see Figure 2).

![Figure 1a. Blade with steel clip](image1a)

![Figure 1b. Blade excitation by means of a magnetic field](image1b)

![Figure 2. Blade air mover assembly and its air flow direction](image2)

The blade is stimulated by means of a magnetic field in 180° phase shift generated by the coils (see Figure 1b). Thus, air flow streams are generated in the direction to the movable end of the blade.

3. Experimental details

To enhance the air mover efficiency and, in addition, to reduce the air mover size, at least four target variables were identified (Table 1).

| Table 1. Target variables under investigation. |
|-----------------------------------------------|
| Variable                | Values                              |
| Mag. excitation material | steel clip / spring steel sheet (1.4310) |
| Coil core shape         | rod core / U-core                   |
| Coil inductance         | 330µF, 1000µF, 10000µF              |
| Notch                   | with, without, notch                |
| Blade bulk material     | PVC-U / MYLAR                       |
3.1. Magnetic excitation material

The steel clip (0.5 mm) used in [1] was bulky and slowed down the blade movement because of its mass. To reduce its mass, the clip was substituted with two spring steel sheets, which were glued on the blade surface in position of the coils. Two different thicknesses (10µm, 20 µm) were investigated. The effect on the blade movement and volumetric flow rate, using the spring steel sheets instead of the steel clip, were examined.

3.2. Coil shape

The magnetic field flux of a rod core shaped coil surrounded the coil in a circular manner. To concentrate the magnetic field flux in the direction to the ferromagnetic material, the influence of a U-core shaped coil was investigated.

3.3. Coil inductance value

The magnetic force $F_{\text{mag}}$, which is the driving force for the blade movement, is based on Eq. 1:

$$F_{\text{mag}} = \frac{I^2 \cdot L_{\text{ind}}}{2 \cdot s}$$

where $s$ is the distance between the coil and the ferromagnetic spring steel sheet, $I$ is the current through coil and $L_{\text{ind}}$ the coil inductance. The magnetic force, $F_{\text{mag}}$, increases in direct proportion with the coil inductance $L_{\text{ind}}$. To confirm this, three different coil inductances were also studied.

3.4. Notch at the vibration node

To check if a notch, located at the vibration node, leads to an additional acceleration at the blade tip, and thus could enhance the efficiency, the blade structure and its dimensions shown in Figure 3 was examined. In the first step, only a notch height of 2 mm was investigated.

![Figure 3. Schematic of the blade and the investigated dimensions](image)

3.5. Blade bulk material

In [1], a simple overhead projector PVC-U (Poly-vinyl chloride) sheet as blade bulk material was used. To test the importance of the blade bulk material on the air mover performance, a spring steel sheet and a Mylar (Poly-ethylene terephthalate) sheet, known from [3], were investigated as bulk materials. A spring steel sheet should have the best long-term in comparison with polymers but, due to the higher stiffness of the steel, a larger current through the coils is necessary. In addition, its movement becomes very noisy. Consequently, it was excluded from this examination. Mylar (from DUPONT) is wear- and chemical-resistant as well as isotropic, has a wide operating temperature range of -70 °C to +150 °C, and is available in various thicknesses: stepwise from 0.012 mm to 0.13 mm. To make a direct comparison between the PVC-U and the Mylar material, equal sheet thicknesses of $t = 0.1$ mm were investigated.
4. Experimental set-ups

Figure 4 shows the experimental set-up to determine the fan performance curve, according to the AMSE/ANSI Standard 210-07 [6]. It includes a flow chamber, a hot-wire anemometer and a manual inclined tube manometer. The flow chamber has a movable shutter at the air flow outlet to record a fan characteristic. The air flow inlet is located in the centre at the front plane of the flow chamber. Its opening height was set to 14 mm, which takes into account the investigated blade height of 12 mm. Its opening width was set to 26 mm, which considers the maximum blade amplitude at the tip.

Volumetric flow rate \( \dot{V} \) was determined from Eq. 2:

\[
\dot{V} = 60000 \cdot w \cdot h \cdot v \quad [l/min]
\]  

(2)

where \( w \) [m] and \( h \) [m] are the flow chamber outlet opening channel dimensions, and \( v \) [m/s] the air speed. The air speed was measured with a hot wire-anemometer from KIMO CTV210 (resolution limit \( \Delta v = 0.01 \) m/s). The static pressure, \( \rho_{\text{stat}} \), was measured with a manual inclined tube manometer from DOSCH, Type D580 (resolution limit \( \Delta \rho_{\text{stat}} = 0.25 \) Pa). The fan power loss, \( P_v \), was determined by the product of the current \( I \) through the coils and the voltage \( V \) across the coil terminals.

The static fan efficiency \( \eta \) was determined by Eq. 3:

\[
\eta = \frac{\rho_{\text{stat}} \cdot \dot{V}}{P_v}
\]  

(3)

In Figures 6 and 7, a test set-up is shown, that was used to demonstrate the cooling performance for microelectronic assemblies. As a representative microelectronic component, a thermal test chip (TTC) was chosen. The TTC consisted of a 3 by 3 thermal test die. The TTC was mounted using flip-chip technology on a FR-4 board (chip area: 11.8 x 11.8 mm\(^2\), substrate-area: 25 x 20 mm\(^2\)) (Figure 5a).

With this TTC, it was possible to have a small-outline heat source. The heat source could be easily controlled using internal resistances. The junction temperature could be measured using an internal temperature-sensitive diode (Figure 5b).

![Figure 4. Experimental set-up to determine the air flow rate \( \dot{V} \) and the static pressure \( \rho_{\text{stat}} \).](image)

![Figure 5. a) TTC assembly b) thermal test die structure.](image)
The air mover was placed in front of the TTC, so that the forced air stream flowed directly over the active TTC surface. The transient junction temperature behaviour of the TTC surface was additional recorded by an IR-camera. The ambient temperature influence was observed by a NiCr thermocouple. The overall set-up was enclosed by a box to avoid external ambient influences.

![Figure 6. Schematic of test set-up to demonstrate cooling potential](image)

![Figure 7. Photograph of the test set-up.](image)

5. Results and discussion

Firstly, the experimental results for the target variables identified in Table 1 are presented. Secondly, a designed prototype, based on the experimental results, is presented, followed by an evaluation regarding its cooling performance for an electronic component. Finally, the classification of the introduced prototype in relation to other small scale fans is discussed.

The following results are based on the blade dimensions and the ferromagnetic material position given in Figure 8. To have comparable results with [1], the previously tested PVC-U was used as the blade material. If not stated otherwise, two rod core shaped coils with a coil inductance of $L_{\text{ind}} = 1000\mu\text{H}$ each and a total power loss of $P_V = 1\text{W}$ were used.

![Figure 8. Blade dimensions and ferromagnetic material position.](image)

The volumetric flow rate was determined with the shutter in fully opened position (Figure 4) and, for the static pressure, the shutter was fully closed.

5.1. Magnetic excitation material

In Figures 9 and 10, the blade length dependent volumetric flow rate and the static pressure are presented for the each excitation material, the steel clip and the spring steel sheet. Two different thicknesses (10 µm and 20 µm) of the spring steel sheet were investigated.
In Figure 9, it can be observed that the volumetric flow rate shows a linearly decreasing behaviour for the blade with the glued spring steel sheet (thickness 20 µm) compared to the blade with the assembled steel clip. The 10 µm thick spring steel sheet shows only a low response, probably due to a decreasing magnetic response of the magnetic flux.

Figure 10 presents the static pressure behaviour as a function of the blade length. For the 20 µm thick spring steel sheet, the static pressure demonstrates a decaying behaviour. For the steel clip and the 10 µm thick spring steel sheet, a maximum static pressure of only ~ 0.5 Pa could be observed. A maximum for both variables could be found at a blade length of ~ 40 mm using the 20 µm thick spring steel sheet.

5.2. Notch at the vibration node and coil inductance value

In Figures 11 and 12 it can be seen that a notch at the vibration node has a positive influence on the volumetric flow rate and the static pressure.

It can also be seen that, in this investigated case, a notch could help to increase the volumetric flow rate and to double the static pressure. In addition, it can be noted that increasing the coil inductance value by up to one order of magnitude, both variable values could only nearly be doubled. In this investigation commercially-available rod core shaped coils were used.

5.3. Coil shape

Commercial small scale U-Core shaped coils were not available, so a custom wire winding had to be done. Because of the small scale U-core form factor (9.9 x 8.2 x 2.82 mm³), a maximum practical wire
turn ratio could only be realized for an inductance value of $L_{\text{ind}} = 470 \, \mu\text{H}$, which will be investigated further.

![Figure 13](image1.png)  ![Figure 14](image2.png)

**Figure 13.** Volumetric flow rate $\dot{V}$ as function of the blade length comparing U-core shaped coils instead of rod core ones.

**Figure 14.** Static pressure $\rho_{\text{stat}}$ as function of the blade length comparing U-core shaped coils instead of rod core ones.

In Figures 13 and 14, it can be observed that the volumetric flow rate and the static pressure can be increased by decreasing the blade length and using U-core shaped coils.

5.4. Blade material

The result presented in Figures 15 and 16 are based on the use of a U-core shaped coil with an inductance value of $L_{\text{ind}} = 470 \, \mu\text{H}$.

![Figure 15](image3.png)  ![Figure 16](image4.png)

**Figure 15.** Volumetric flow rate $\dot{V}$ as function of blade length comparing PVC-U and Mylar as blade material.

**Figure 16.** Static pressure $\rho_{\text{stat}}$ as function of blade length comparing PVC-U and Mylar as blade material.

A maximum of both variables can be observed with Mylar as blade bulk material at a blade length of $L = 40 \, \text{mm}$. Due to the fact that the Young’s modulus from Mylar ($\sim 3.6 \, \text{GPa}$) is 1 GPa lower than that for PVC-U ($\sim 4.6 \, \text{GPa}$), the blade amplitude $\gamma$ increases and the static pressure is increased, from 1 Pa to 4.8 Pa. The volumetric flow rate is also increased from 16 to 17 l/min.

5.5. Prototype design

Based on the experimental results, it was possible to reduce the length of the blade from 70 mm down to 40 mm, in order to achieve a ‘matchbox’ size. To this end, the blade bulk material was changed to MYLAR and, instead of the steel-clip, two magnetic steel sheets of 20 µm thickness were used. In addition, the two rod-core shaped coils were replaced by two U-core ones. As mentioned above, only a maximum coil inductance of $L_{\text{ind}} = 470 \, \mu\text{H}$ for the U-core shaped coils was available. With these settings, the power loss could be reduced from $P_V = 1 \, \text{W}$ to 0.24 W ($U_{\text{ind}} = 1.2 \, \text{V}$, $I_{\text{ind}} = 200 \, \text{mA}$) and a maximum volumetric flow rate of $\dot{V}_{\text{max}} = 14 \, \text{l/min}$ ($\rho_{\text{stat}} = 0 \, \text{Pa}$) as well as a maximum static pressure...
of $\rho_{\text{stat, max}} = 3 \text{ Pa}$ ($\dot{V} = 0 \text{ l/min}$) could be observed. In Figure 17, an initial fan performance curve is presented, based on different shutter positions.

![Figure 17. Fan performance curve of the prototype](image)

Due to the shortened blade length, the operating frequency shifted, from 18 Hz [1] to 125 Hz, into the audible range.

5.6. Cooling performance for electronic components
In this investigation, a Mylar foil as blade bulk material with a blade length of $L = 40 \text{ mm}$, two 20 µm thick spring steel foils, a notch at the vibration node, two U-core coils with an inductance value of $L_{\text{ind}} = 470\mu\text{H}$ were used. The power loss was reduced to $P_V = 0.7 \text{ W}$. In Figure 16, the transient junction temperature, $T_J$, behaviour, after switching on the heated TTC ($t = 0 \text{ s}$), for three constant driven power losses $P_{V,\text{TTC}}$ are presented.
Figure 16. Junction temperature $T_J$ behaviour of TTC after switching on the blade fan at different constant driven TTC power losses.

As an illustrative result, the steady state junction temperature, $T_J$, of the TTC could be reduced: e.g. for $\Delta T \sim 30$ K, from $T_J = 85 \, ^\circ\text{C}$ ($P_{V,TTC}/A = 597 \, \text{mW/cm}^2$), in case of natural convection (blade air mover turned off), down to $56 \, ^\circ\text{C}$ when the blade air mover was turned on.

5.7. Classification

Several frictional and frictionless fan solutions are currently on the market. In order to classify the magnetic field driven prototype presented in this paper, Table 2 compares a selection of small scale radial and axial fans [5][7] as well as the piezoelectric driven fan given in [3].

|                          | SUNON mighty radial mini fan | Small scale axial fan | Piezoelectric driven fan | Magnetic field driven prototype |
|--------------------------|-----------------------------|-----------------------|--------------------------|--------------------------------|
| L x W x H [mm$^3$]       | 9 x 9 x 3                   | 25 x 25 x 10          | 69 x 30 x 15             | 50 x 35 x 14                   |
| $P_V$ [mW]               | 185                         | n. a.                 | 30                       | 240                            |
| $U_{\text{supply}}$ [V]  | 3                           | n. a.                 | > 40                      | 1.2                            |
| $\rho_{\text{stat, max}}$ [Pa] (at $V = 0$ l/min) | 4.44                        | 43                    | 3.5                       | 3                              |
| $V_{\text{max}}$ [l/min] (at $\rho_{\text{stat}} = 0$ Pa) | 1.17                        | 13                    | 27.5                      | 14                             |
| $\eta$ [%]               | 0.047                       | n. a.                 | 5.35                     | 0.249                          |

In case of the static pressure, the frictional fans could reach a one order of magnitude higher maximum static pressure than the frictionless fans. In case of the volumetric flow rate, it can be noticed that the piezoelectric driven fan, due to the design concept, shows a two times better maximum flow rate. Comparing the power loss and the efficiency, the piezoelectric driven fan shows the lowest power consumption and the best fan efficiency, but a much higher supply voltage (of > 40 V) must be provided. In the case of the geometrical dimensions, the frictional fans have a smaller form factor than the frictionless ones.

The magnetic field driven prototype can be classified in between the frictional and frictionless fans. It has a comparable volumetric flow rate to the radial fan and is twice as large as the radial fan. Compared to the piezoelectric driven fan, it has similar maximum static pressure and comparable geometrical dimensions. Only the power loss and the voltage supply behave inversely. Because the
magnetic field driven concept was planned as a more wear-resistant solution, it shows a good potential to enhance the heat transfer by increasing the fluid circulation in regions which are otherwise thermally not manageable.

6. Conclusions
This paper has presented the results of an on-going design study to reduce power loss and size of a frictionless air mover, as well as to enhance the volumetric flow rate and static pressure. This overall goal was reached. An optimized prototype with a size of 50 x 14 x 35 mm³ (including casing) was realized. The maximum volumetric flow rate was enhanced from \( V_{\text{max}} = 10 \text{ l/min} \) to 14 l/min (at \( \rho_{\text{stat}} = 0 \text{ Pa} \)), and the maximum static pressure was enhanced from \( \rho_{\text{stat,max}} = 0.5 \text{ Pa} \) to 3 Pa (at \( \dot{V} = 0 \text{ l/min} \)). In addition, the power loss \( P_V \) was lowered from 1W to 0.24 W. In addition, the fan efficiency was more than doubled (0.1 % to 0.25 %). The air mover is low weight (< 50 g) and could be manufactured at low cost (material costs < USD 5). Depending on the fan system working point, the volumetric flow rate can be adjusted from \( \dot{V} \sim 5 \text{ – 12 l/min} \) at a static pressure range of \( \rho_{\text{stat}} \sim 1 \text{ – 2.7 Pa} \). An examination of the thermal performance has shown that a representative active electronic component could be cooled down adequately and is therefore thermally manageable. The next steps are to conduct long term stability examinations and to aim to double the current volumetric flow rate.

7. References
[1] R. Schacht, A. Hausdorf, B. Wunderle, 2012, “Frictionless Air Flow Blade Fan for Thermal Management of Electronics,” 13th IEEE IThERM Conference, San Diego, USA, 978-1-4244-9532-0/12/$31.00 ©2012 IEEE
[2] Mark Kimber, Kazuhiro Suzuki, Nobutaka Kitsunai, Kenichi Seki, and Suresh V. Garimella, 2008, “Quantification of Piezoelectric Fan Flow Rate Performance and Experimental Identification of Installation Effects” 978-1-4244-1701-8/08/$25.00 ©2008 IEEE.
[3] Mark Kimber, Kazuhiro Suzuki, Nobutaka Kitsunai, Kenichi Seki, and Suresh V. Garimella, 2009, “Pressure and Flow Rate Performance of Piezoelectric Fans” IEEE Transactions on Components and Packaging Technologies, Vol. 32, No. 4, December 2009.
[4] Kercher, Lee, Brand, Allen, and Glezter, 2003, “Microjet Cooling Devices for thermal Management of Electronics”, IEEE Transactions on Components and Packaging Technologies, vol. Vol. 26, pp. 359-366, Juni 2003
[5] Jason Stafford, Ed Walsh, Vanessa Egan, “Local heat transfer performance and exit flow characteristics of a miniature axial fan”, International Journal of Heat and Fluid Flow 31 (2010) pp. 952 – 960.
[6] Laboratory Methods of Testing Fans for Certified Aerodynamic Performance Rating, ANSI/AMCA Standard 210-07.
[7] Radial fan, www. Titan-cd.com
[8] Piezo fan, www.piezo.com (Part Number: RFN1-LV-02)
[9] Memran fan, SynJet® Low Profile Cooler. Co. Nuventix (www.nuventix.com)