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Heat transfer by a piezoelectric fan on a flat surface subject to the influence of horizontal/vertical arrangement

Sheng-Fu Liu [a], Ren-Tsung Huang [a], Wen-Jenn Sheu [a], Chi-Chuan Wang [b,*]

[a] Department of Power Mechanical Engineering, National Tsing Hua University, Hsinchu 300, Taiwan

[b] Energy & Environment Research Laboratories, Industrial Technology Research Institute, D100 EEL/TRI, Bldg. 64, 195∼6 Section 4, Chung Hsing Rd., Chutung, Hsinchu 310, Taiwan

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ABSTRACT

This study presents an experimental work concerning the thermal performance of piezoelectric fans. A total of six piezoelectric fans with various blade geometries are made and tested. The influence of geometric parameters, including the horizontal/vertical arrangement, and location of the piezofan, on the performance of piezofans is examined. It is found that the heat transfer augmentation of the piezofan comes from the entrained airflow during each oscillation cycle and the jet-like air stream at the fan tip, yet these two modes are of the same order of magnitude. The heat transfer performance for vertical arrangement shows a symmetrical distribution and peaks at the center region whereas the horizontal arrangement possesses an asymmetrical distribution and shows an early peak at $x/L = 0.25$. It is also found that the heat transfer performance for horizontal arrangement is not necessarily lower than that of vertical one. Based on the dimensionless analysis to the test results for the all six fans, a correlation applicable for $x/L = 0$ is proposed. The mean deviation is 4.8% that can well describe the influence of geometrical parameters.

1. Introduction

Piezoelectric fans are an innovative design which are gaining acceptance as feasible solution for portable electronic products. These fans generally consist of a patch of piezoelectric material attached to a blade of various flexible materials as illustrated in Fig. 1. When an alternating voltage is applied to the piezoelectric patch, it expands and contracts in the lengthwise direction, causing bending moments at both ends of the patch. These moments drive the attached blade to oscillate at the same frequency. The amplitude of oscillation reaches a maximum when the input voltage is applied at the resonance frequency of the fan. This oscillatory motion drives air flow which can be exploited for cooling. Toda and Osaka [1] pioneered the research on piezoelectric fans and proposed that the vibrating structure can operate as small fans with sufficient air-moving capability to cool electronic equipments. The vibrating blade drove the air to flow radially outwards like a traditional manual fan. Subsequently, Toda [2] investigated the factors which affect the performance of piezoelectric fans. His theoretical analyses and experimental data conclude that the air volume flow rate was proportional to the blade width and thickness, but was independent of the blade length. Yoriniga et al. [3] investigated the construction of the bimorph that is suitable for the fan. They also examined the air volume flow rate and the noise level of a fan placed between two fixed plates. Yoo et al. [4] developed several types of piezoelectric fans and tested them at a frequency of 60 Hz with an applied voltage of 110 V and 220 V, respectively. It was found that the most effective fan was the one made from a phosphor bronze shim and with PZT in a bimorph configuration. This fan showed a tip displacement of 35.5 mm and a produced wind velocity of 3.1 m/s driven by a 220 V, 60 Hz power source. Ihara and Watanabe [5] performed a visualization study by means of smoke wire method and a numerical simulation of the flow field driven by single and double oscillating plates. They found that the normal flow in the downstream direction was greatly affected by the distance between two plates for counter-phase oscillation, but such flow was scarcely affected for in-phase oscillation. Açıkalın et al. [6] analyzed a single piezoelectric fan vibrating near a heat source to find out the optimal heat transfer coefficient. They studied the effect of fan length, amplitude, frequency and distance from the heat source. Meanwhile, Açıkalın et al. [7] tested the feasibility for implementing piezoelectric fans in electronic devices. Kim et al. [8] presented a visualization study on the flow structure generated by a piezoelectric fan. They found that each vibration cycle generated a pair of counter-rotating vortices between which a high velocity region was formed. Wait et al. [9] studied the characteristics of piezoelectric fans running at higher resonance modes with numerical simulation and flow visualization. The results revealed certain advantages, such as greater fluid mixing and electro-

* Corresponding author. Tel.: +886 3 5916294; fax: +886 3 5820250.
E-mail address: ccwang@itri.org.tw (C.-C. Wang).

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Nomenclature

A	surface area, m^2
Cp	specific heat, kJ kg^{-1} K^{-1}
D	tip displacement, mm
d	distance defined in Fig. 2(c), mm
f_r	resonant frequency, Hz
g	gravity, m s^{-2}
H_h	fan height defined in Fig. 2(c), mm
H_v	fan height defined in Fig. 2(c), mm
Hor	horizontal arrangement
h	heat transfer coefficient, W m^{-2} K^{-1}
L	heat source length, mm
l	length, mm
Nu	Nusselt number, dimensionless
Q	heat transfer rate, W
Ra	Rayleigh number, dimensionless
T	temperature, °C
t	thickness, mm
V	voltage, V
$Vert$	vertical arrangement
w	width, mm
x	distance defined in Fig. 2(c), mm

Greek symbols

α	thermal diffusivity of air, m^2 s^{-1}
β	coefficient of thermal expansion of air, K^{-1}
λ	thermal conductivity of air, W m^{-1} K^{-1}
μ	dynamic viscosity of air, kg m^{-1} s^{-1}
ρ	density, kg m^{-3}
ξ	enhancement ratio, *dimensionless*

Subscripts

a	air
b	blade
c	convection
e	equivalent
exp	experiment
f	fluid film
n	natural convection
p	piezo-ceramic
pf	piezoelectric fan
pred	prediction
T	total

Superscript

-	average quantity

mechanical energy conversion, of the piezoelectric fan being operated at higher resonance modes were offset by the increased power consumption and decreased flow delivery.

Kimber et al. [10] presented the local heat transfer coefficients induced by piezoelectric fans by using the infrared camera technique. They found that the local heat transfer coefficient transition from a lobed shape at small gaps to an almost circular shape at intermediate gaps. At larger gaps, the heat transfer coefficient distribution becomes elliptical in shape. By exploiting a design of experiments (DOE) analysis, Açıkalın et al. [11] revealed the fan frequency offset from resonance and the fan amplitude as the critical parameters. For the best case, an enhancement in convective heat transfer coefficient exceeding 375% relative to natural convection was observed, resulting in a temperature drop at the heat source of more than 36.4 °C.

The foregoing researches showed the capacity of piezoelectric fans as a means for electronic cooling. Regardless of its promising features, there are still many aspects must be resolved. In practice, since the piezofan is operated near the heat source, related geometric parameters such as distance to the heat source, relative

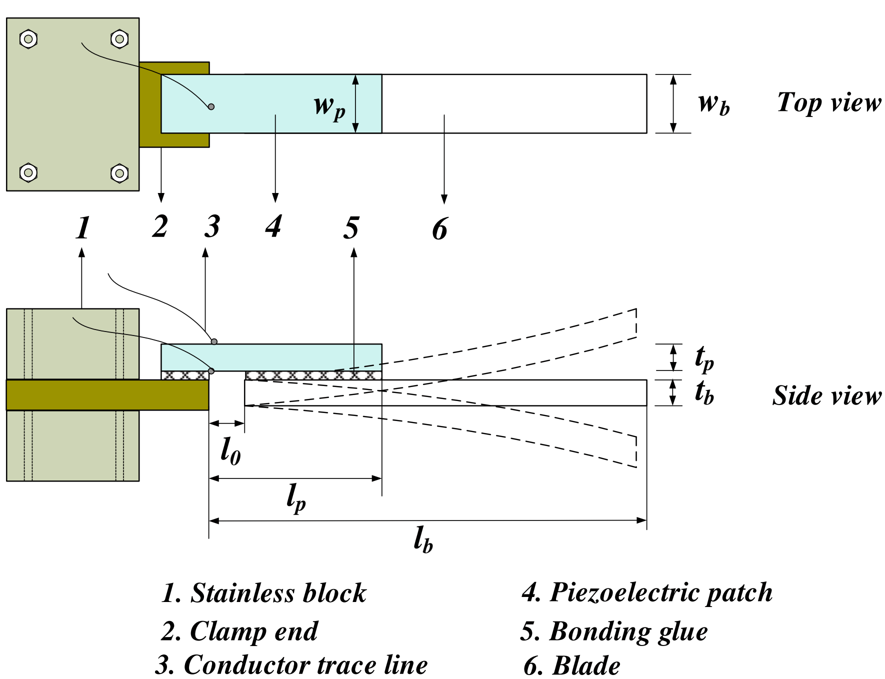

Fig. 1. Construction of the piezoelectric fan.

location with the heat source, and fan arrangement (vertical or horizontal) are quite important for the air flow generation in a limited physical space where complex interactions are likely. This eventually gives rise to the need of accurate, reliable and fast estimation of the performance of piezofans. In view of this need, it is the objective of this study to provide relevant test results and correlation to quantitatively describe the associate influence.

2. Experimental setup

Experiments are performed in an environmental chamber whose volume is 0.86 m × 0.86 m × 1.16 m (L × W × H). The environmental chamber can provide a temperature condition in the range of $0\,^{\circ}\mathrm{C} \leqslant T_a \leqslant 50\,^{\circ}\mathrm{C}$ with a controlled resolution of 0.2 °C. To simulate the natural flow condition, the air ventilator is turned off inside the test chamber when the ambient temperature reaches 25 °C. In particular, the air conditioner outside the test chamber continues to operate to maintain the room temperature at 25 °C. The test facility is inside the test chamber which consists of an aluminum plate, a heater, and an insulation box. The test plate is made of aluminum alloy 5083 with a thermal conductivity of $121\ \mathrm{W\,m^{-1}\,K^{-1}}$. The plate is cleaned and tested at an ambient temperature of 25 °C with the power inputs ranging from 3 W to 15 W. A Kapton heater with an identical size as the test plate is used to eliminate the spreading resistance. In this study, we employ two plates to conduct the natural/piezofan tests. These plates are square in shape with a dimension of 45 mm and 102 mm, respectively. An insulation box made of bakelite with a low thermal conductivity of $0.233\ \mathrm{W\,m^{-1}\,K^{-1}}$ is placed beneath the heater to reduce the heat loss. In addition, a high thermal conductivity grease ($k = 2.1\ \mathrm{W\,m^{-1}\,K^{-1}}$) is used to connect the heat sink and the heater. For further minimization of the contact resistance, four M4 screws with fixed applied pressure located at the corners of the base plate are employed. The heater is powered by a DC power supply. The experimental procedure for natural convection tests generally follows that by Huang et al. [12]. For typical natural convection tests, the aluminum plate is heated for 2 h to reach the steady state where the equipped thermocouples fluctuate within 0.2 °C. With the data for natural convective performance, the signal generator is then turned on to produce forced convection for heat transfer enhancement. Empirically, an additional 40 min are required to reach thermal equilibrium for the piezofan tests. The data are again taken for evaluating the piezofan performance.

Fig. 1 shows the construction of the piezoelectric fan. The test piezofan is simply constructed by bonding a piezoelectric patch to a thin blade sheet. The piezoelectric ceramics used in the present study is PZT which is commercially available as sheets (Piezo Systems, Inc.). The physical and electrical properties of the PZT are shown in Table 1. The bonding glue is black epoxy which is supplied in twin packs of resin and hardener. The blade is made of Mylar with two different thicknesses of 0.188 mm and 0.25 mm. A total of six piezoelectric fans (#1–#6) are made and tested with relevant geometric details of the fans listing in Table 2. In an effort to control the test condition for a fair comparison, the clamp end is fastened between two identical stainless blocks of 0.295 kg with a fixed pressure applied to the corners of the blocks. The tip displacement of the piezoelectric fans is measured by a laser displacement sensor (Keyence LK-30) with a resolution of 0.05 μm. All the measurement data are taken at the resonant frequency of the running piezoelectric fans. The influence of geometric parameters, such as distance to the heat source, relative position of the piezofan within the heater, and fan arrangement is investigated in this study. Fig. 2(a) is a schematic of the piezofan test system in which the signal generator and the amplifier used in the present study are Tektronix AFG3022 and Peizomechanik Gmbh SVR 500-3, respectively. An oscilloscope (Qmax UM2202) is used to measure the applied voltage to the piezoelectric fans. A schematic of these parametric influences is shown in Fig. 2(c).

Table 1
Physical and electrical properties of PZT.

Density (kg/m^3)	7800
Young's modulus (GPa)	50
Shear modulus (GPa)	62
Curie point (°C)	230
Mechanical quality factor	30
Piezoelectric constant, d_{31} ($\times 10^{-12}$ m/V)	−320

Table 2
Geometric details of tested piezoelectric fans.

	w (mm)	t_b (mm)	l_b (mm)	f_r (Hz)
Fan #1	12	0.188	61	47
Fan #2	12	0.25	63	53
Fan #3	15	0.188	70	30
Fan #4	15	0.188	65	38
Fan #5	37	0.188	75	28
Fan #6	37	0.25	66	40

3. Data reduction

In the present study, the ambient air temperature is always controlled at 25 °C and the thermophysical properties in the Nu and Ra numbers are evaluated at the film temperature, i.e.

$$T_f = \frac{1}{2}(T_a + T_b) \tag{1}$$

The average heat transfer coefficient can be calculated from the following:

$$\bar{h} = \frac{Q_t}{A_t(T_b - T_a)} \tag{2}$$

$$\overline{Nu} = \bar{h}L/\lambda \tag{3}$$

The index of piezoelectric fan performance is characterized by the enhancement ratio (ξ) which represents the average heat transfer coefficient under a running piezoelectric fan divided by that under natural convection:

$$\xi = \bar{h}_{pf}/\bar{h}_n \tag{4}$$

The experimental uncertainty is estimated using the uncertainty propagation equation proposed by Kline and McClintock [13]. The highest uncertainty for the heat transfer coefficient is about 8.4%, occurring at the lowest input power of 3 W. For the purpose of comparison, test results in terms of Nusselt number for the present two test plates in natural convection are compared with the correlation proposed by Al-Arabi and El-Riedy [14] as shown in Fig. 3. The satisfactory agreement between measurements and the correlation substantiates the usability of the present test system.

4. Results and discussion

It is well known that the flow structure caused by the piezo fan is actually rather complex, yet the influence becomes even more complicated when the fluid motion is limited by the distance of the heat source. Fig. 4 shows the effect of vertical/horizontal arrangement for the piezofan (fan #3) on the overall heat transfer performance pertaining to $L = 45$ mm and $Q_t = 3$ W. The abscissa of Fig. 4 is x/L, representing the tip of the piezofan into the heat source, whereas the ordinate is the heat transfer enhancement ratio ξ ($= h_p/h_n$). It is interesting to know that the enhancement ratio

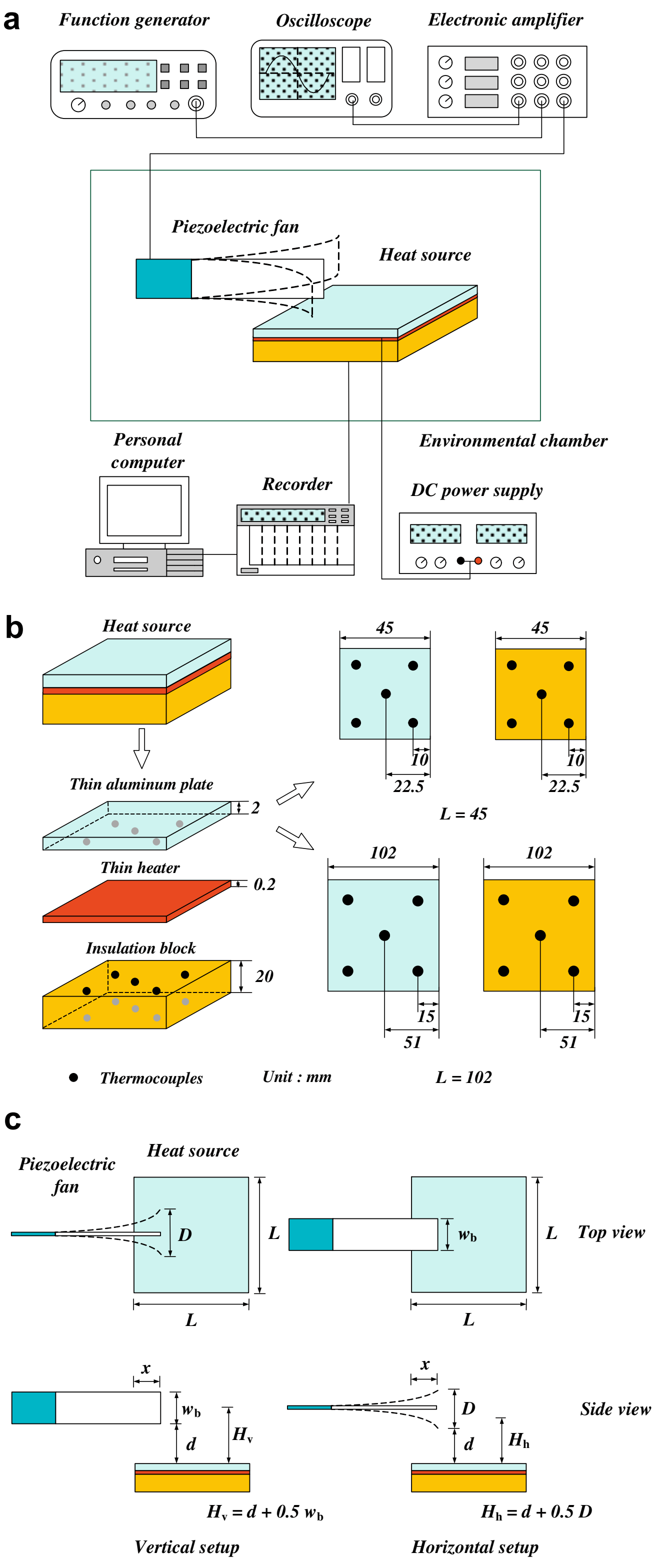

Fig. 2. Schematic of (a) test system of the piezofan, (b) temperature measurement points and (c) relevant geometric arrangement of the piezofan.

for horizontal arrangement is symmetrical at $x/L = 0.5$ whereas the vertical one possess an asymmetrical nature and peaks at $x/L = 0.25$. This phenomenon is applicable for both amplitudes ($D = 6$ mm and $D = 11$ mm). In general, the augmentation of the piezofan comes from two places: the first is the jet flow in front of the fan tip and the other is the induced air flow during each

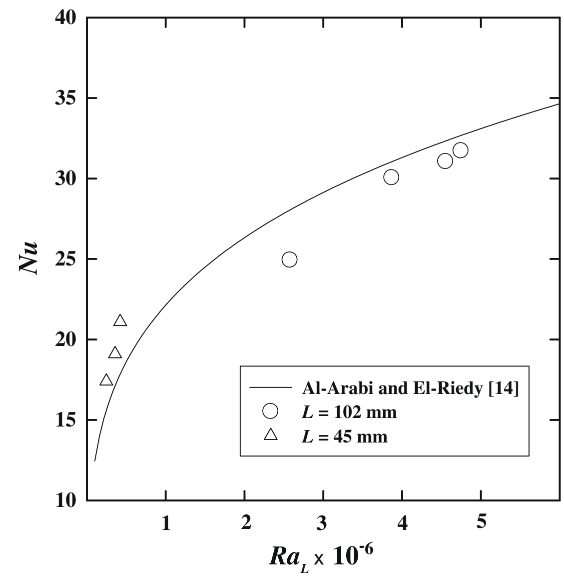

Fig. 3. Comparison for natural convection amid the present measurement and the correlation by Al-Arabi and El-Riedy [14].

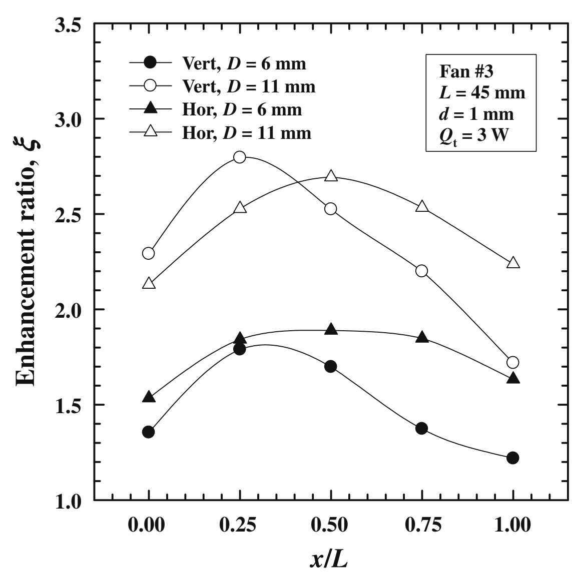

Fig. 4. Influence of piezofan location and arrangement on the heat transfer augmentation.

vibrating cycle, yet the induced air is generally turned into two rotating vortices shedding from the fan tip of piezoelectric fans. However, as pointed out by Açıkalın [11], an additional pair of small secondary flow may appear nearby by the fan tip and heat source. This normally complicates the actual performance of the piezofan with the presence of heat source.

In the meantime, there are some differences between the induced air flow amid vertical and horizontal arrangements. At the entrance of the heat source, i.e. $x/L = 0$, the major heat transfer augmentation mainly comes from the air flow at the fan tip, suggesting small difference for these two arrangements. As the piezofan is placed into the heat source, the induced airflow pattern may be quite different. For a horizontal arrangement, as schematically shown in Fig. 5, the induced air flow becomes more and more pronounced when the piezofan is brought to the center of the heat source. Since both entrained air flow and the jet-like flow at the fan tip contribute to the overall heat transfer enhancement, there-

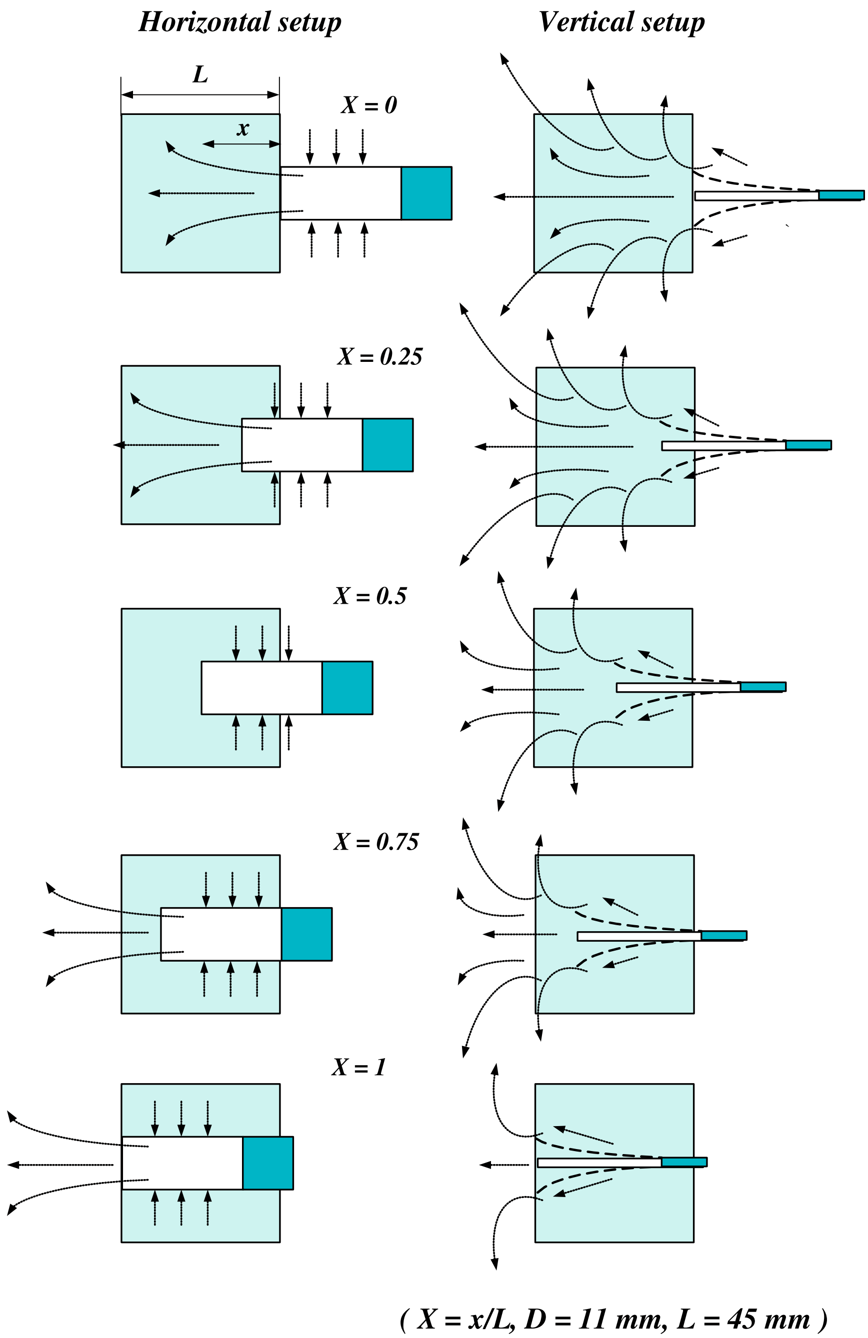

Fig. 5. Schematic showing the airflow structure amid vertical and horizontal configurations.

by leading to a continuous rise in heat transfer performance. The heat transfer performance peaks at $x/L = 0.5$, and starts to decline thereafter. This is somehow expected because the influenced area of the jet-like region at the fin tip decreases accordingly. When the piezofan is placed to the full coverage, i.e. $x/L = 1$, the effect of jet-like region is smallest whereas the effect of entrained air flow is the largest. As shown in Fig. 4, the heat transfer augmentation at $x = L$ is roughly the same as that of $x = 0$. The results imply that the two modes of heat transfer augmentation (entrained air flow and jet-like air flow at the fin tip) are actually of the same magnitude.

Unlike that of horizontal arrangement that possesses a symmetrical augmentation factor at $x/L \approx 0.5$, the vertical arrangement shows an early peak at $x/L = 0.25$, and the performance dwindles thereafter. Moreover, the performance drops considerably when $x/L > 0.5$. This phenomenon is in fact related to the nature of the vertical arrangement. The entrained air flowrate comes from top and bottom of the fan blade. With the vertical configuration, most entrained air comes from the upper side of the piezofan whereas the entrained air from the bottom is restricted by the heat source surface. This blockage becomes more and more pronounced as the vertical piezofan is placed further into the heat source. One can notice that the entrained air will not be obstructed for the horizontal configuration. As a consequence, the decay of heat transfer performance for vertical arrangement is rather evident compared with the horizontal configuration.

However, it is interesting to know that the performance of vertical arrangement does not always surpass that of horizontal configuration. The results are not in line with those reported by Açıkalın et al. [7] for their study shows that the performance for

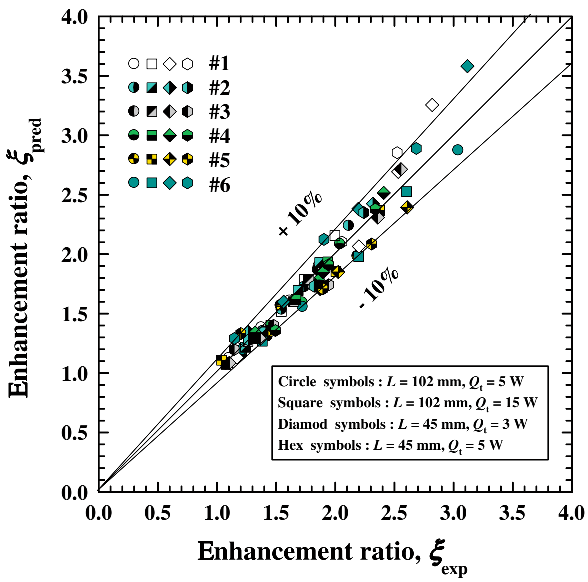

Fig. 6. Comparison between the proposed correlation and the experimental data.

horizontal configuration is always below that of vertical one. Basically, the difference arises from the test environment. This study is performed at an open environment whereas theirs are conducted in a confined space. The entrained air flow from the horizontal configuration is further suppressed for this confinement. Hence the heat transfer performance for the vertical arrangement always exceeds that for the horizontal arrangement.

The performance of the piezofan is strongly related to the geometric placement and the interactions with the heat source. In this study, efforts are made to correlate the associate performance for various piezofans under vertical arrangement. Based on the Buckingham Pi theory (Buckingham [15]), a dimensionless analysis is made to qualitatively investigate the performance of the piezofan. It is found that the total number of variables which describe the piezofan system is thirteen in number, including. h, C_p, λ, ρ, μ, g, w, D, β, L, ΔT, and f_r. By employing the dimensionless analysis yields four major dimensionless parameters: $(hL\lambda^{-1})$, (wL^{-1}), (DL^{-1}), $(f_rL2\nu^{-1})$, $(g\beta T\ L^3\alpha^{-1}\nu^{-1})$. Or correspondingly, the Nusselt number, the dimensionless fan width, the dimensionless displacement, the frequency parameter, and the Rayleigh number. For most of the applications, the piezo fan is placed outside the heating source. Hence, we had correlated the test results with $x/L = 0$ and vertical arrangement. The Nusselt number of the piezofan is further termed into an augmentation factor representing the enhancement of piezofan relative to that of natural convection. The final correlation takes the following form:

$$\xi = 1 + 0.0274\left(\frac{D}{L}\right)^{2.223}\left(\frac{f_rL^2}{\nu}\right)^{1.482}\left(\frac{W_b}{L}\right)^{0.741}Ra_L^{-0.346} \quad (5)$$

The applicable range of this correlation is $2\times10^5 \leqslant Ra_L \leqslant 6\times10^6$ and the mean deviation of this correlation is 4.8%. Fig. 6 shows the predictive ability of this correlation against the six piezofan being tested.

5. Conclusions

This study examines the influence of geometric parameters, including the horizontal/vertical arrangement and location of the piezofan, on the performance of piezofans. A total of six piezofans are made and tested to examine the heat transfer performance along two flat plates. Based on the foregoing discussions, the following results are then concluded:

(1) The heat transfer augmentation of the piezofan comes from the entrained airflow during each oscillation cycle and the jet-like air stream at the fan tip. These two modes of heat transfer enhancement are actually of the same order of magnitude.
(2) The heat transfer performance for horizontal arrangement shows symmetrical distribution and peaks at the center region whereas the vertical arrangement possesses an asymmetrical distribution and peaks at $x/L \approx 0.25$. Compared with the horizontal arrangement, the heat transfer performance decays quite considerably for vertical arrangement. This is due to the blockage of entrained air flow.
(3) It is found that heat transfer performance for the horizontal arrangement is not necessarily lower than that for the vertical one.
(4) Based on the dimensionless analysis to the test results for the all six fans, a correlation applicable to vertical arrangement is proposed. The mean deviation is 4.8% is proposed that can well describe the influences of geometrical parameters.

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