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# Low-voltage electromagnetic actuators for flapping-wing micro aerial vehicles

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ABSTRACT

This paper presents a new type of electromagnetic actuator driven by a low voltage of 5.5 V for potential applications in insect scale Flapping-wing Micro Aerial Vehicles (FMAVs). The actuator can generate vibratory motions via cantilever-beam or cantilever-plate structures to drive artificial wings through flapping motion. The induced magnetic Lorentz force is analyzed by a simplified lumped mass model and the dynamic characteristics including resonant frequency and amplitude are investigated through experimental tests. In the prototype designs and tests, a peak amplitude of 29.8 mm from a 25 mm-long cantilever beam structure has been achieved. Furthermore, a prototype device with the cantilever beam actuator using small magnets of 197.2 mg has demonstrated 5.7 times higher output power density at 0.56 W/Kg as compared with that of another prototype device using larger magnets of 2409.2 mg. This implies the potential for further miniaturized systems toward FMAVs. Under a driving voltage of 5.5 V, a cantilever plate actuator has been shown to drive a pair of artificial wings to achieve 20° flapping amplitude at 101.4 Hz, which results in a high power density of 9.52 W/Kg. By applying AC current to a planar coil, the power density of the actuator can be further enhanced to 48.56 W/Kg. As such, the low-voltage electromagnetic actuators could broaden the driving mechanisms for insect scale FMAVs.


## 1. Introduction

Flapping-wing Micro Aerial Vehicles (FMAVs) have potential applications in the operations of rescue, exploration, and reconnaissance due to their high maneuverability and agility. Over the past two decades, the development of FMAV has attracted strong research interests in a variety of small-scale actuator developments [1]. In general, these actuators are designed to transform their fixed mechanical output motions to specific flapping motions to achieve high lift force. However, both complex structural and sophisticated electronic components are not preferred for these actuators due to possible excess weight and tedious fabrication processes. Given that, small-scale actuator with simple structure and power electronics is still challenging in the field of FMAV actuation.

Based on the output motions, the state-of-art small-scale actuators in FMAVs can be classified into two key categories: rotary actuators and vibratory actuators. Conventional electromagnetic motors [2–5] can produce rotary motions with advantages in low driving voltage and high power density. Furthermore, electronic components to drive and control these motors have been well-developed. However, these motors suffer greatly in the weight-to-lift efficiency when the dimension of FMAVs decreases to centimeter or millimeter scale [6]. Furthermore, some complex transmission mechanisms, such as crank-rocker mechanism, are required to transfer rotary motions to flapping motions. As a result, conventional motor is often used in large scale FMAVs [3]. For insect scale FMAVs (wing span less than 5 cm), micro actuators based on piezoelectric effects [7–12] and electrostatic effects [13–15] have been adopted as alternative drive options. Instead of outputting rotary motions, these actuators produce vibratory motions and transform it into biomimetic flapping motions for the artificial flapping wings with simple structures (such as cantilevers or simply-supported beams). However, these vibratory actuators all require high operating voltages (hundreds of volts for piezoelectric

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actuators [7–12] and thousands of volts for electrostatic actuator [13–15]), which hinders the development of FMAVs [16]. To overcome the drawbacks of high operation voltage, vibratory electromagnetic actuator has been proven feasible to drive the insect scale FMAV with low voltage and simple structure [17,18].

In this investigation, we report a new type of electromagnetic actuator for insect scale FMAVs. The actuator is driven under a low voltage of a few volts to generate vibratory motions via cantilever-beam or cantilever-plate structures. The actuator's dynamic performances including resonant frequency and amplitude are studied experimentally with a theoretical model. Since power density is a key parameter of the actuator to reflect the overall performance of the system, it is evaluated in details for potential future miniaturizations for FMAVs. The cantilever-plate type design is found to have high output power density in our preliminary experiments. Compared with piezoelectric or electrostatic actuators, the presented actuator provides a new driving scheme with much lower operation voltage for the insect scale FMAVs.

## 2. Configuration

Fig. 1(a) illustrates the schematic diagram of the FMAV system driven by a pair of the proposed electromagnetic actuators. The airframe provides support for the whole system while the electromagnetic actuators generate vibrational outputs to drive the artificial wings for the aerodynamic lift force. Fig. 1(b) illustrates configuration of the actuator which consists of two permanent magnets, a cantilever beam, an isolated base and a conductive wire. Two NdFeB magnets are placed in parallel with a gap distance of 0.5 mm to establish a stable magnetic field. A metal beam made of Ni-Ti alloy with a diameter of 100 μm is firstly wrapped with a Si-Al alloy conductive wire with a diameter of 25.4 μm and fixed to the isolated base. Thereafter, the beam and the conductive wire are placed in the gap between the two magnets. To prevent the two magnets from being attracted to each other, a 0.5 mm-thick partition plate is inserted between the magnets and fixed by superglue.

The working principle of the actuator can be explained as follows: when an AC voltage is applied to the conductive wire, the beam (along with the conductive wire) will subject to the periodic Lorentz force. When the driving frequency is adjusted to match the natural frequency of the cantilever system, the cantilever beam will be excited into resonance and outputs vibration motion. Since the AC current amplitude is in the range of hundreds of milliamperes, the required operating voltage is several volts.

In above configuration, the gap distance is set as 0.5 mm in the prototype system for easy assembly and avoidance of possible mechanical touching of the cantilever beam during operations. The cantilever beam can undergo large deformation under resonance, the material of Ni-Ti is chosen for its property of superelasticity. The Si-Al bonding wire is selected as the conductive wire for its low stiffness and low density to reduce its influence on the operations of the cantilever beam.

## 3. Dynamic modeling

A simplified model is developed to analyze the vibration characteristics of the actuator system as illustrated in Fig. 2(a). The cantilever beam vibrates under the combination of the electromagnetic force, $F_E$, air damping force, $F_A$, and elastic restoring force of the beam structure. In this investigation, the vibration system is simplified as a system with one single degree of freedom (DOF) as illustrated in Fig. 2(b), which has equivalent mass, $m_e$, equivalent stiffness, $k_e$, and equivalent air damping coefficient, $c_e$. In the model, the electromagnetic force is assumed to be concentrated and applied at the right side of the magnets, where $x = a$. The peak vibration amplitude at $x = a$ is defined as the effective length of the magnets (indicated as "$b_e$" in Fig. 2(a)). The dynamic equation of the single DOF system is given by:

$$m_e\ddot{y} + c_e\dot{y} + k_e y = F_E \sin(\omega t) \tag{1}$$

where ω is angular frequency of the applied AC current.

The vibration shape of the beam is assumed to be the first vibration mode of a cantilever subjected to a periodic force and the following equations can be obtained as [19,20]:

$$\begin{gathered} y(x) = Y(x)sin(\omega t + \varphi) \\ Y(x) = C_1[\cos\beta_1 x - \cosh\beta_1 x + r_1(\sin\beta_1 x - \sinh\beta_1 x)] \end{gathered} \tag{2a}$$

$$\cos\beta_1 L = 1.8751, \quad r_1 = \frac{\sin\beta_1 L - \sinh\beta_1 L}{\cos\beta_1 L + \cosh\beta_1 L} \tag{2b}$$

where $y(x)$ is the vibration amplitude at a location, x; $Y(x)$ is the vibration mode function; $L$ is the length of the cantilever; $C_1$ and $\beta_1$ are constants related to boundary conditions which can be eliminated in subsequent derivations; $\omega$ is the angular frequency; and $\varphi$ is the initial phase of the AC current.

The total energy of the real cantilever system (Fig. 2(a)) is set to be equal to that of the simplified DOF system (Fig. 2(b)) in the derivations. As such, the kinetic energy of the vibration system remains the same before (right side of Eq. (3)) and after the simplified analyses (left side of Eq. (3)) and the equivalent mass can be derived from:

$$\frac{1}{2}m_e\dot{y}(a)^2 = \frac{1}{2}m_{wire}\dot{y}(a)^2 + \int_0^L \frac{1}{2}\rho A\dot{y}^2 dx \tag{3}$$

It is noted that $m_{wire}$ is the mass of the conductive wire wrapped around the beam as shown in Fig. 1(b); $\rho$ and $A$ are density and the cross sectional area of the cantilever beam.

To obtain the equivalent air damping coefficient $c_e$, the air damping force is firstly calculated. Since the estimated Reynolds number is over 1 for the cantilever system, a simplified dish string model [21–23] is utilized and the air damping force per unit length is given by:

$$df = (3\pi\mu d + 0.75\pi d^2\sqrt{2\rho_a\mu\omega})\dot{y}dx \tag{4}$$

where $\rho_a$ is the air density; $\mu$ is the air viscosity; and $d$ is the diameter of the beam.

Considering the energy dissipated by air damping in one cycle remains the same before and after the simplified analyses, the equivalent air damping coefficient can be derived from:

$$\int_0^T c_e\dot{y}_a \cdot \dot{y}_a dt = \int_0^T\int_0^L df \cdot \dot{y}dt \tag{5}$$

where $T$ is the vibration period.

To calculate the equivalent stiffness, the cantilever is assumed to be subjected to a concentrated loading by electromagnetic force; $y_a$ is the displacement at $x = a$; and the equivalent stiffness can be obtained from:

$$k_e = \frac{F_E}{y_a}, \quad F_E = BIl \tag{6}$$

where $B$ is the magnetic flux density which can be measured by a gauss-meter; $I$ is the AC current; and $l$ is width of the magnets. With equivalent mass and stiffness being calculated, the resonant frequency can be given by:

$$f_r = \sqrt{\frac{k_e}{m_e}} \tag{7}$$

After obtaining the equivalent mass, stiffness and air damping coefficient based on the above theoretical derivations, Eq. (1) can