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Axial Excitation Response of Polygonal Mirror Scanner Motor Supported by Thrust Magnetic Bearing and Radial Air Bearing

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Abstract. *This article describes an axial transient vibration and axial excitation response of a polygonal mirror scanner rotor driven by a flat-type brushless dc motor and supported by a passive thrust magnetic bearing and a radial air bearing. From results of experimental investigations, the following characteristics of the vibration have been deduced: (1) Repulsive magnetic force was induced between the rotor magnet and stator coils of the dc motor under current passage. The force caused axial static displacement and axial transient vibration. (2) Because the stiffness of the thrust magnetic bearing was small, large axial vibration took place due to the axial excitation. (3) Two types of damper were introduced to suppress the axial transient vibration and the axial excitation response; one was an eddy-current damper and the other was an air damper. It was demonstrated that the air damper was effective for reducing the axial vibration. © 2006 Society for Imaging Science and Technology.* [DOI: 10.2352/J.ImagingSci.Technol.(2006)50:1(111)]

INTRODUCTION

A polygonal mirror scanner motor is used in the exposure subsystem of the digital electrophotographic engine to scan the laser beam and to write latent images on the photoreceptor.[1] The rotation of the mirror is required to be stable, constant velocity, and low vibration to realize high image quality without banding.[2–4] It is also necessary to be high speed for high print speed and high resolution machines, long life, small power loss, low acoustic noise, and low cost. The motor shown in Figs. 1 and 2 has been developed to meet these requirements.[5,6] The rotor is driven by a flat type brushless and coreless dc motor. The polygonal mirror is attached to the rotor. The rotor is supported in the radial direction by an outer rotor type self-acting grooved journal air bearing[7] and in the axial direction by an axially stable passive magnetic bearing. The magnetic bearing utilizes an axial restoring magnetic force between radially magnetized rotor and stator magnets.[8]

Two types of axial vibration take place in this motor, because the stiffness and the damping of the thrust magnetic bearing were small. One is axial transient vibration of the rotor during startup operation due to the axial magnetic repulsive force induced in the flat motor.[5] It prolongs the startup time and in the worst case the rotor collides with a casing in the axial direction. The other is a large axial vibration due to the external excitation. Because electrophotographic copy machines or laser printers are not always mounted on a rigid floor, they are sometimes excited by the floor vibration. High image quality must be maintained even when an external vibration force is applied to the machine.

To suppress these vibrations, we developed two types of damper, one is an eddy-current damper[8] and the other is an air damper. The former is a copper ring mounted near the stator magnet as shown in the left side of Fig. 1. Eddy-current caused by magnetic flux change in the ring absorbs vibration energy and thus the copper ring performs as the

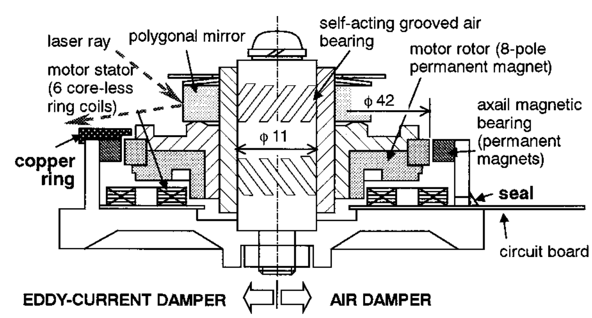

Figure 1. Configuration of polygonal mirror scanner rotor driven by flat-type brushless dc motor and supported by axial magnetic and radial air bearing. Rated speed = 340 s^{-1}.

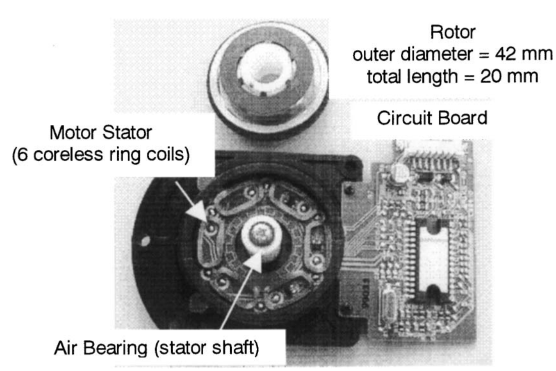

Figure 2. Polygonal mirror scanner motor. The rotor (upper) is disassembled from the stator (lower).

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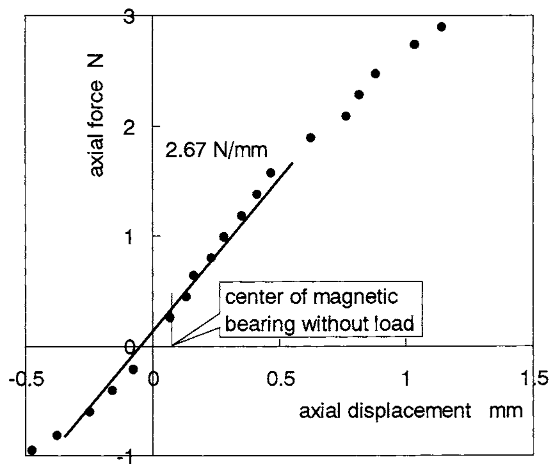

Figure 3. Axial static stiffness of magnetic bearing.

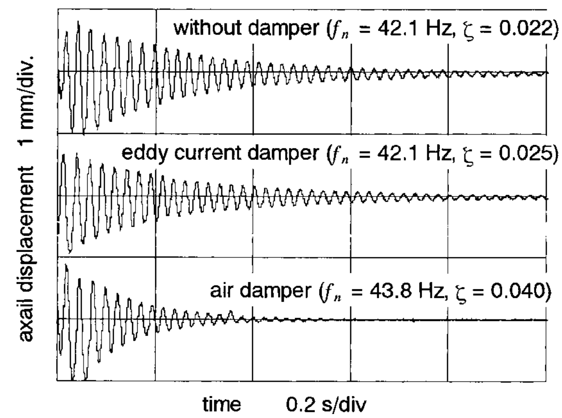

Figure 4. Free vibration responses without damper, with eddy-current damper, and with air damper.

vibration damper. On the other hand, the air damper utilizes viscosity of air to restrict airflow at the small gap (3–6 μm) of the magnetic bearing by sealing large holes in a support of the stator magnet. The objectives of the present investigation are to confirm performance of these dampers in order to realize highly reliable digital copy machines and laser printers.

AXIAL STIFFNESS AND DAMPING

Weights were added stepwise on the center of the rotor without rotation and axial static displacement of the rotor was measured by a laser displacement meter (Keyence, Tokyo, LK-2000) to deduce the axial static stiffness of the magnetic bearing. Figure 3 shows the result. The origin of the axial displacement designates a static equilibrium position of the rotor. The force was almost linear to the displacement near the center. The deduced stiffness was 2670 N/m that corresponded to the axial natural frequency of 42.4 Hz.

Next, the axial stiffness and the damping constant were measured by the impact test method. Free vibration responses without damper, with the eddy-current damper, and with the air damper are shown in Fig. 4. The stiffness without damping and with the eddy-current damper coincided to the static measurement, but was slightly increased by the air damper. Although the eddy-current damper was not effective, the damping effect was significantly increased by the air damper.

AXIAL VIBRATION DURING START-UP OPERATION

The axial vibration was measured by the laser displacement meter by the same method described in Ref. 5. Figure 5 shows the measured transient vibration during startup operation. The characteristics of the vibration were as follows:

At the Beginning of Startup Operation

When the motor current, about 0.5 A, was applied, the rotor started to rotate. The axial lumped repulsive force was applied by the flat motor to the rotor and simultaneously the rotor was displaced statically in the upwards direction.[5]

During Startup Operation

The axial lumped force induced not only the axial static displacement but also the axial transient vibration.[5] The

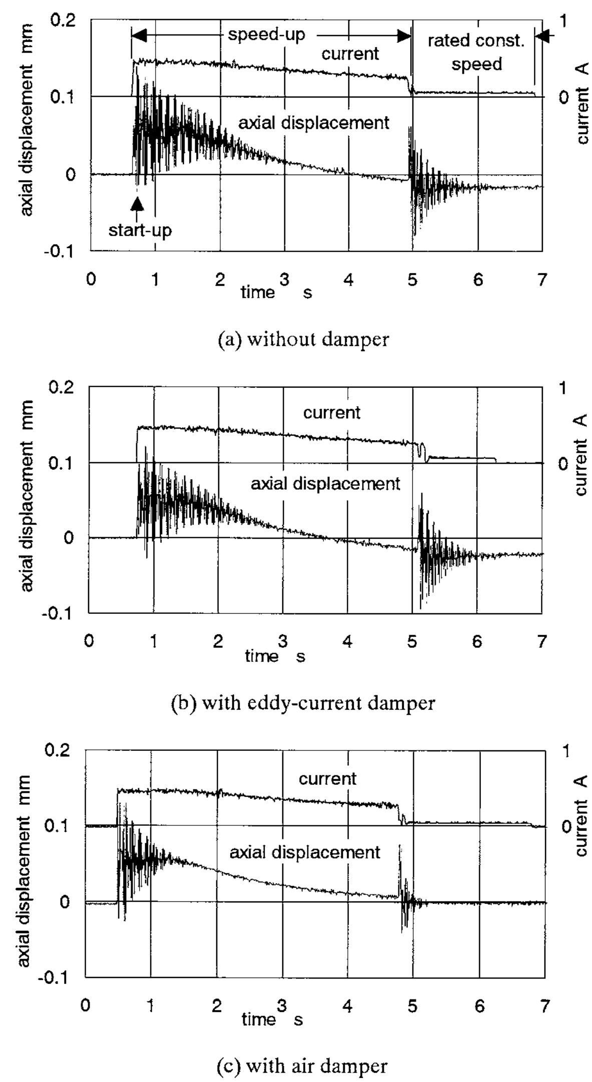

Figure 5. Transient motor current and vibration response during speed-up and rated operation.

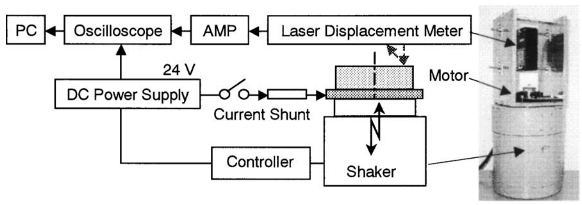

Figure 6. Experimental setup of axial excitation test.

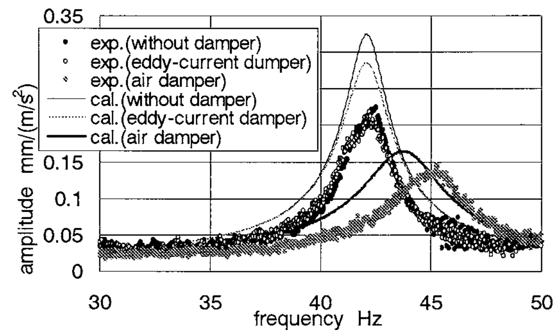

Figure 7. Axial excitation responses without damper, with eddy current, and with air damper at the rated speed in case of 0.33 Hz/s sweep-up rate and 1 m/s^2 acceleration.

measured frequency of the vibration was 43.0 Hz without damper or with the eddy-current damper, and it was 44.5 Hz with the air damper. These corresponded to the 2745 and 2939 N/m axial stiffness, respectively. These dynamically derived values of stiffness during the startup operation were about 100 N/m higher than those measured by the impact tests. The increment corresponded to the stiffness due to the current passage, because the restoring force was induced between the motor stator and rotor.[5] It was clearly demonstrated that the air damper was effective for suppressing the transient vibration, although the eddy-current damper was not effective, as expected from the result of the impact test.

Just before and after Rated Speed

A large hunting of the current was observed just before and after the rotor speed fell in the rated speed (340 s^{-1}) and this induced a large transient vibration.[5] The air damper was also effective against this vibration. Because the motor current was very low, 0.05 A, the axial stiffness was not increased by the motor and therefore the frequency of the transient vibration was almost the same as without rotation.

AXIAL EXCITATION RESPONSE

An axial excitation test was performed with a setup shown in Fig. 6. The motor and the laser displacement meter were mounted on a magnetic shaker (Shinken, Tokyo, G-2005D) and excited with the sine wave of constant acceleration, 1–3 m/s^2, at the rated speed of the motor. The frequency of the sine wave was swept up from 30 to 50 Hz and then swept down to 30 Hz at the rate of ±0.33 Hz/s. The acceleration was measured by an accelerometer (PCB, New York, 353B52).

Figure 7 shows axial excitation responses without damper, with the eddy current, and with the air damper at the rated speed in case of the sweep-up scheme of acceleration frequency and 1 m/s^2 constant acceleration. Calculation was based on a simple linear system model with one-degree-of-freedom and the effective stiffness and damping coefficient derived by the impact test method. It was confirmed that the stiffness, and therefore the resonant frequency, was increased by the air damper, and that the air damper was effective for suppressing the axial excitation response. However, the agreement between calculation and measurement is not satisfactory. Two reasons were assumed for this discrepancy. One is due to the substantially high sweep-up rate of the excitation frequency, and the other is nonlinearity of the stiffness. Figures 8 and 9 show these effects. Figure 8 shows axial excitation responses with the air damper at the rated speed in case of the sweep-up scheme of acceleration frequency and 1, 2, and 3 m/s^2 acceleration, and Fig. 9 shows the responses in case of sweep-down. Comparing Fig. 8 with Fig. 9 we can see that although the nonlinearity was small, a substantial hysteresis was observed which caused the discrepancy.

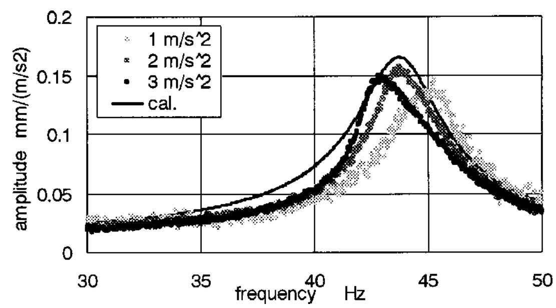

Figure 8. Axial excitation responses with air damper at the rated speed in case of 0.33 Hz/s sweep-up rate.

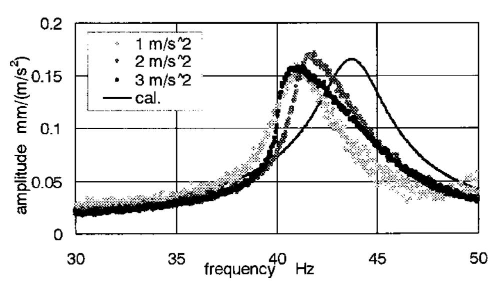

Figure 9. Axial excitation responses with air damper at the rated speed in case of −0.33 Hz/s sweep-down rate.

Finally, the temperature rise on adapting the air damper was measured, as the airflow in the stator housing was restricted by the seal of the air damper, and therefore there was a risk of unacceptable temperature rise. However, the actual temperature rise in the motor housing was very small, only 5°, as shown in Fig. 10. There is adequate margin for the reliability and durability of the motor such as with insulation of the motor coil.

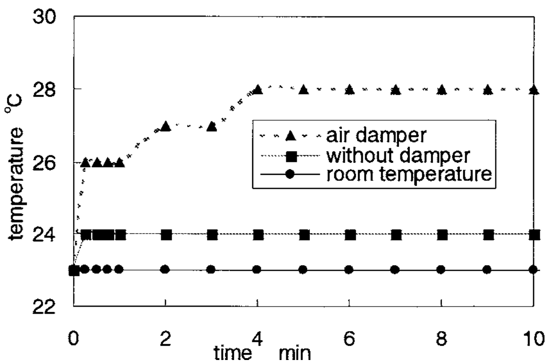

Figure 10. Temperature rise without and with air damper in stator housing.

CONCLUDING REMARKS

Axial transient vibration and axial excitation response were investigated on a polygonal mirror scanner rotor driven by a flat-type brushless dc motor and supported by a passive thrust magnetic bearing and a radial air bearing. Because the stiffness of the thrust magnetic bearing was small, large axial vibration occurred due to axial force. To suppress the axial vibration, two types of damper were introduced; one was an eddy-current damper, and the other was an air damper. It was demonstrated that the air damper effectively reduced the axial vibration, although the eddy-current damper showed only a small effect. Although the air damper had a risk of unacceptable temperature rise due to the sealing of the airflow in the housing, it proved to be satisfactorily small. In conclusion, the air damper, which is simple and cost effective, is proposed as an axial damper of the motor.

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