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CHARACTERIZATION OF SUPERCONDUCTING MAGNETIC BEARINGS
(Dynamic stiffness and damping coefficient in axial direction)

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ABSTRACT

High-Tc superconductor as a stator and permanent magnets for a rotor were assembled into a superconducting magnetic bearing. The dynamic stiffness and the damping coefficient of the superconducting magnetic bearing in axial direction were measured. The dynamic stiffness depended on an axial gap between superconductor and permanent magnet. The superconducting magnetic bearings are advantageous for a passive bearing, because they have a vibration damping effect that a permanent magnet bearing does not have. The tendency of its vibration damping coefficient indicated an increase as the resonant frequency increased.

INTRODUCTION

Since a high-Tc superconductor was discovered, many efforts have been made on developing a superconducting magnetic bearing (SMB) (ref.1). The SMB has been appreciated by several factors, levitation pressure, dynamic stiffness (ref.2), damping coefficient, and the rotational speed (ref.3).

Recently a melt process on bulk $YBa_2Cu_3O_X$ has been improved (ref.4) and a levitation pressure has also become large.* The SMB has been able to support a rotor weighing about 1 kg, rotating at a speed of 5,000rpm. †

One of the purpose of this work is to examine the dynamic stiffness and the damping coefficient of SMB in axial direction. They are very significant for bearing design.

EXPERIMENT

Construction of SMB

A superconducting magnetic bearing can be assembled by using high-Tc superconductor as a stator and permanent magnets for a rotor. The superconductor used in this test was a bulk $YBa_2Cu_3O_X$ prepared by quench and melt growth (QMG) process. The diameter of the bulk was 43mm and 12mm in thickness, and was made by Nippon Steel Corporation. The rotor had 4 permanent magnets (Nd-Fe-B, Sumitomo Special Metal Co., Ltd.). The surface flux density of the magnets was 0.45T. Each of the permanent magnets has 20mm in diameter and 4mm in thickness.

* K.Miyamoto and K.Sawano et al. :"Quench and Melt Growth (QMG) Processed Superconducting Materials for Levitation and Magnetic Flux Trapping", Submitted to Proc. ISS'90 (1990) in Sendai, Japan

† R.Takahata, H.Ueyama and T.Yotsuya : "Load Carrying Capacity of Superconducting Magnetic Bearings", Submitted to Proc. The International Symposium on Application of Electromagnetic Forces (1991) in Sendai, Japan

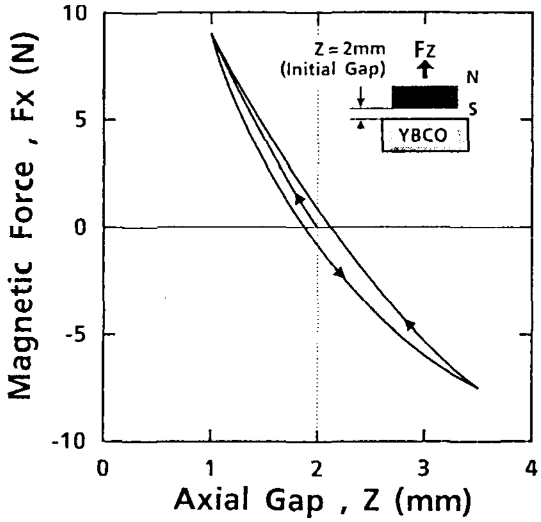

Fig.1 Static stiffness of SMB

The maximum carrying load capacity of the tested bearing was 5 kg in axial direction and the levitation pressure was 3.4N/cm^2. Its value was measured by means of road cell, when the magnet approached the surface of bulk $YBa_2Cu_3O_X$ cooled down to liquid nitrogen temperature.

The static stiffness of tested bearing was 9N/mm as shown in Fig.1. It was also measured by means of road cell when the magnets were moved slowly in axial direction after the bulk $YBa_2Cu_3O_X$ was cooled down by liquid nitrogen at 2mm axial gap between magnet and superconductor.

Test equipment

The apparatus shown in Fig.2 was constructed to measure the vibration damping and the dynamic stiffness of SMB in axial direction. A shaft having permanent magnets on its lower side was located vertically above the bulk $YBa_2Cu_3O_X$. The shaft was able to be supported with no mechanical contact by active magnetic bearings in radial direction and by SMB in axial direction. To decrease the damping effect brought about by eddy current, conducting materials were located as far from permanent magnets as possible. The total mass of the shaft was 1.118 kg.

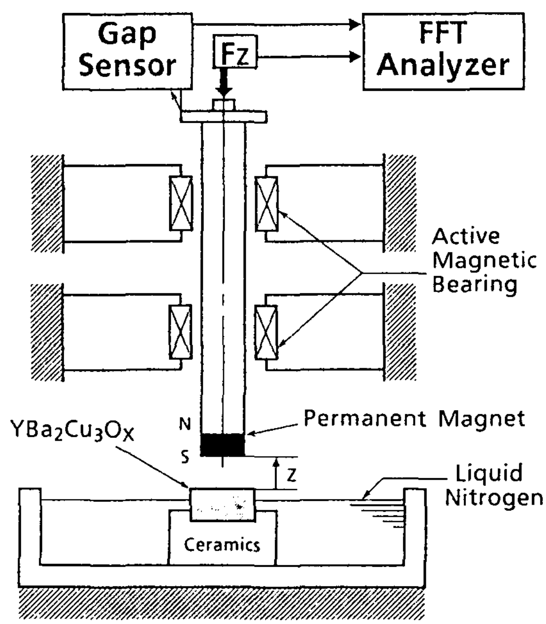

Fig.2 Dynamic stiffness measuring apparatus

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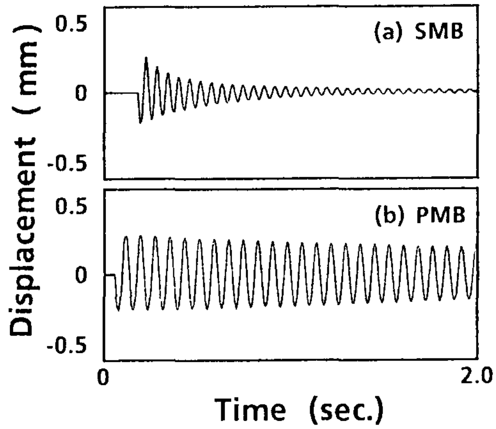

Fig.3 Vibration of shaft in axial direction;
(a) in the case of SMB, (b) in the case of PMB

Test Procedure

The shaft was supported by active magnetic bearings in radial direction and fixed at initial gap Z_0 in axial direction; then the bulk $YBa_2Cu_3O_X$ was cooled down to about 77K by liquid nitrogen. After above field cooling, the shaft was released and supported with no mechanical contact. At that time an axial gap of SMB decreased to $Z<Z_0$. That is the equilibrium position.

RESULT AND DISCUSSION

SMB model in one degree of freedom

A vibration of the shaft in axial direction was measured by gap sensor when an external impulse force Fz acted on the shaft that was supported by SMB. It is shown in Fig.3(a). Fig.3(b) shows a vibration of the shaft supported by a permanent magnet bearing (PMB) by the same procedure. The vibration damping of SMB was larger than that of PMB using repulsive forces of permanent magnets. Therefore an SMB model in one degree of freedom can be shown in Fig.4(b). The motion of this model can also be described by equation (1).

$$m\ddot{z} + c\dot{z} + kz = Fz \qquad (1)$$

In this equation m is the mass of shaft, k is the spring constant and c is the damping coefficient respectively.

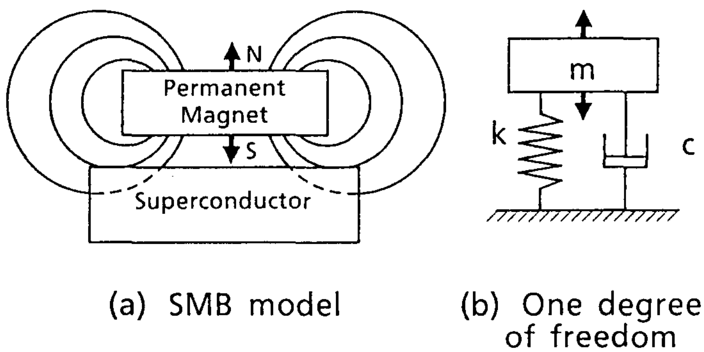

Fig.4 SMB model in 1 degree of freedom

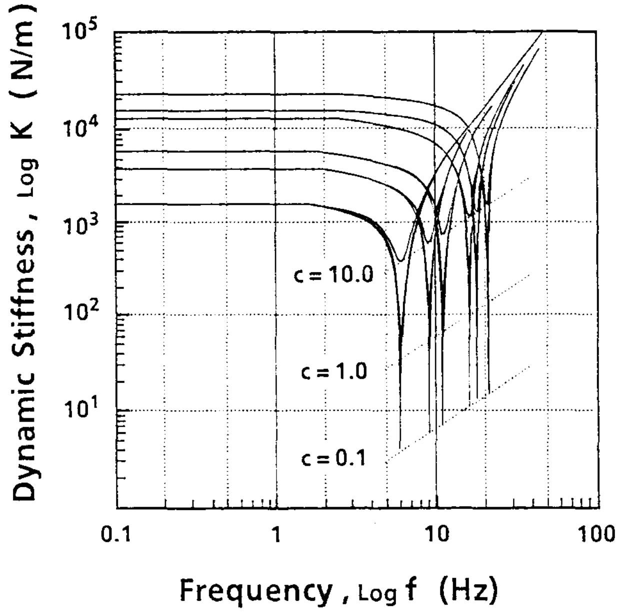

Fig.5 Dynamic stiffness vs vibration frequency (simulation)

The dynamic stiffness K of axial displacement for external force Fz can be also described by equation (2).

$$|K| = \sqrt{(k - m\omega^2)^2 + \omega^2 c^2} \tag{2}$$

where ω is the angular velocity. Arbitrary values (k=1.6, 3.7, 5.4, 11.5, 14.2, 20.0×10^3N/m, c=0.1, 1.0, 10.0Ns/m) in equation (2) are simulated and the results are shown in Fig.5. The minimum value of dynamic stiffness appears at resonant frequency ω_0 that depends upon only spring constant k ($\omega_0=\sqrt{k/m}$). And that minimum value is decided by the damping coefficient c. So it is possible to estimate the damping coefficient by means of measuring dynamic stiffness.

Measured spring constant, k

When the shaft was struck by an impulse hammer in axial direction, the axial vibration of it was measured at several gaps of Z. The values of Z of SMB were 0.8, 1.3, 1.9 mm which corresponded to the initial gap Z_0 = 2.0, 3.0, 4.0mm respectively. Z did not change after the hammering

Spring constant of axial direction is able to be given by equation (3)

$$k = 4\pi^2 \cdot m \cdot f_0{}^2 \tag{3}$$

where f_0 is the resonant frequency.

The value of spring constant k calculated from equation (3) using the measured f_0 is shown in Fig.6. The spring constant increased as the axial gap decreased. The value of spring constant was 15,000N/m at Z=1mm.

Measured damping coefficient, c_s

It is also possible to estimate damping coefficient c_s from the decreasing characteristics of vibration amplitude. Damping coefficient c_s can be described by equation (4).

$$c_s = 2m \cdot \beta \tag{4}$$

where β is a natural logarithm of damping ratio per second in vibration amplitude.

Measured damping coefficient, c_s is shown in Fig.7. Damping coefficient c_s also increased as an axial gap decreased.

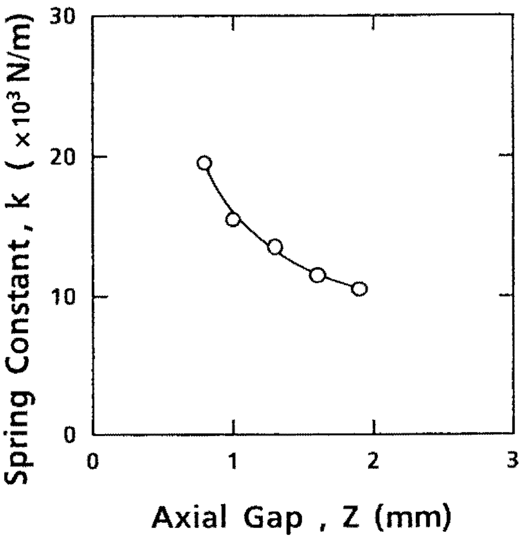

Fig.6 Change in spring constant with axial gap

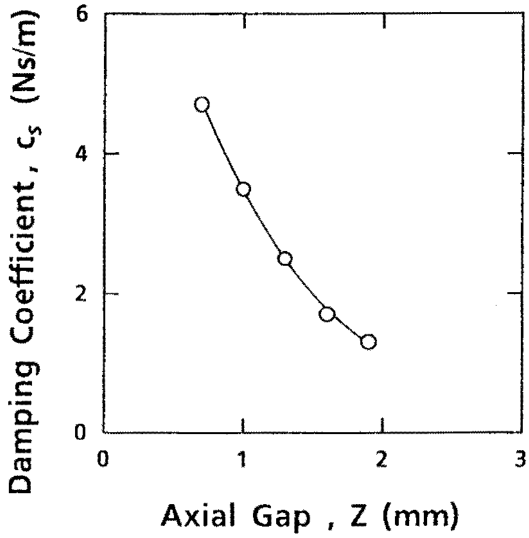

Fig.7 Change in damping coefficient with axial gap

But in this test equipment, c_s included the intrinsic damping coefficient c_i in addition to the damping coefficient $c_{(SMB)}$ due to SMB was caused by eddy current loss and air resistance in a vibration of the shaft. It was clear that the vibration of the shaft supported by the permanent magnet bearing, not having a damping factor, had actually a little damping as shown in Fig.3(b). The damping coefficient $c_{(SMB)}$ is given by equation (5).

$$c_{(SMB)} = c_s - c_i \tag{5}$$

Measured dynamic stiffness, K(ω)

We also measured dynamic stiffness K(ω) of SMB directly by means of FFT analyzer and frequency changeable external force Fz(ω) acting on the shaft with no mechanical contact. That force was attractive force generated by an electromagnet and swept sine-wave excitation current I(ω).

The dynamic stiffness can be measured as the transfer function from the external force Fz(ω) to the displacement Z(ω) of the shaft as shown in equation (6).

$$K(\omega) = Fz(\omega) / Z(\omega) \tag{6}$$

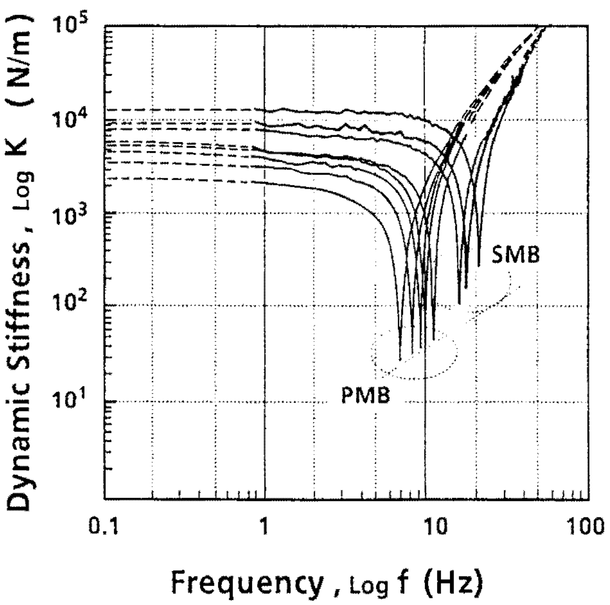

Fig.8 Dynamic stiffness vs vibration frequency in the SMB and PMB

The measured dynamic stiffness of SMB and PMB are shown in Fig.8. The damping coefficient of PMB was constant independently of frequency by judging from the minimum value of dynamic stiffness in comparison with Fig.5. Its value of damping coefficient was 0.6Ns/m and seemed to indicate the value of c_i, while the values of c_s were 1.0, 1.3, 1.9 Ns/m at resonant frequency f_0 = 16.0, 17.75, 21.0Hz respectively, so $c_{(SMB)}$ were calculated from equation (5), $c_{(SMB)}$=0.4,0.7,1.3(Ns/m). The value of $c_{(SMB)}$ seems to increase as the resonant frequency f_0 increased.

These values of c_s in this test are a little smaller than the damping coefficients c_s, measured by hammering, shown in Fig.7, because here $K(\omega)$ was measured by taking $I(\omega)$ instead of $Fz(\omega)$ in equation (6). As $I(\omega)$ is swept sine-wave excitation current, the actual $Fz(\omega)$ acting on the shaft in this test is proportional to $I(\omega)^2$ and is larger than $I(\omega)$ for vibration at resonant point so that the measured dynamic stiffness appearing in Fig.8 seems to be a little smaller than actual dynamic stiffness. Therefore we think it is necessary to linearize $I(\omega)$ for $Fz(\omega)$ in the next step.

Bearing design

Measured damping coefficients of SMB were smaller at least by one order of magnitude than that of active magnetic bearings. It is very important for bearing design to improve the damping coefficient. This damping factor of SMB seemed to be dependent on hysteresis loop shown in fig.1. Therefore an improvement of $YBa_2Cu_3O_X$ materials with a larger hysteresis loop and stiffness than that is expected for one of the ways to develop a practical SMB. If the material improvement is impossible, it will be necessary to add another damping mechanism for SMB to be used in a cryogenic environment.

CONCLUSIONS

Dynamic characteristics of the superconducting magnetic bearing, using high-Tc superconductor as a stator and a permanent magnet for a rotor, in axial direction were measured.

(1) Dynamic stiffness and damping coefficient of SMB were dependent on an axial gap.
(2) SMB had a damping factor that had never been observed on PMB.
(3) Damping coefficients of SMB increased as the resonant frequency increased. These values were smaller at least by one order of magnitude than that of the active magnetic bearing.

It is very significant for bearing design to clarify the dynamic stiffness and damping coefficient of SMB. Accordingly the test equipment will be more improved to measure those factors exactly.

ACKNOWLEDGMENTS

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