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AN EVALUATION OF SOME STRATEGIES FOR VIBRATION CONTROL OF FLEXIBLE ROTORS

C R Burrows

Professor of Systems Engineering
Head of School of Mechanical Engineering
and
Director of the Fluid Power Centre

University of Bath
United Kingdom

INTRODUCTION

The arguments against using magnetic bearings have remained fairly constant in the last 15 to 20 years, meanwhile a fledgling industry has emerged to confound the critics. The design, development and manufacture of magnetic bearings have reached a stage where their suitability as machine elements has been demonstrated (ref. 1 and ref. 2). In spite of this progress, their use is frequently challenged on the basis of cost or reliability.

The potentially catastrophic effect of bearing failure in a high speed machine due to power failure or electronic component failure is cited as a reason for retaining rolling-element or oil-film bearings. If that argument is successfully countered, excessive cost is the next hurdle which is raised. There is a circular element in this argument. The initial price will not fall until magnetic bearings are in more widespread use; when they are, the spectre of reliability may disappear or be reduced, thereby encouraging more applications and helping to reduce cost. There is already evidence that the reliability of magnetic bearings has achieved an acceptable level in applications when high cost can be tolerated (ref. 3).

This movement towards acceptability would be enhanced if the inherent capability of magnetic bearings as **active** control elements were fully utilised. The capability to 'shape' machine dynamics is not fully exploited when a magnetic bearing or actuator is controlled using only local feedback as is frequently the case.

The central argument in this paper is that the full technological and commercial promise of magnetic bearings will be fulfilled only if attention is focussed on the control problems associated with their use. This fact has stimulated the activity of several groups of academic researchers. Schweitzer has played a leading part in this activity (ref. 4, ref. 5, and ref. 6). The problems of using closed loop control led Burrows and his co-workers to develop an open-loop adaptive strategy (ref. 7, ref. 8 and ref. 9). Other workers have now begun to explore open-loop control strategies (ref. 10 and ref. 11).

Although this paper focuses upon magnetic bearing control as the key issue, it does not deny the importance of striving to improve their force density, safety, and reliability at lower cost. All of these factors are crucial, but secondary to the control issue.

NOMENCLATURE

$\boldsymbol{C}$, $\boldsymbol{K}$, $\boldsymbol{M}$	damping, stiffness and mass matrices respectively
$\boldsymbol{F}(j\omega)$, $\boldsymbol{F}_d(j\omega)$	force vectors (frequency domain)
i	mass station
j	$\sqrt{-1}$
k	integer
p	number of control forces
$\mathcal{Q}$, $\mathcal{Q}_d$, $\mathcal{Q}_m$, $\mathcal{Q}_x$, $\mathcal{Q}_y$	displacement vectors (frequency domain)
$\boldsymbol{R}$, Δ	receptance matrices (frequency domain)
$\boldsymbol{R}_x$, $\boldsymbol{R}_y$	receptance vectors
$\overline{\boldsymbol{R}}$	conjugate of $\boldsymbol{R}$
$\boldsymbol{U}$	control vector
$\boldsymbol{U}_x$, $\boldsymbol{U}_y$	trial force (frequency domain)
u_x, u_y	forces applied by magnetic actuator
$\delta(\ )$	change in value
ω	frequency
$(\ )^{-1}$, $(\ )^T$	matrix inverse and transpose respectively
$(\hat{\ })$	estimated value

BEARING OR ACTUATOR

In rotating machines various types of bearings can be used to support a rotor. Oil-film bearings provide some measure of vibration control as well as supporting the rotor. Magnetic bearings have the potential for providing more flexible vibration control, but their load carrying capacity is not as effective as conventional oil-film journal bearings. (For comparison, a typical bearing load for a magnetic bearing is $100kN/m^2$ compared with $4000kN/m^2$ for an oil-film bearing). There is clearly a benefit in separating the load carrying capacity from the control function. That was the approach studied by Burrows and Sahinkaya (ref. 7) and by Chan Hew Wai and Morel (ref. 2). Another benefit of this configuration is that the passive bearings provide a degree of vibration isolation or control if the magnetic actuator develops a fault.

This paper will focus on the control of transverse vibration of rotors. However, the open-loop adaptive strategy described later can be applied to reduce the effect of longitudinal vibrations (ref. 12) and the same strategy can be adapted to control active equipment mounts.

BACKGROUND

The degree of vibration control that can be exercised by a bearing is dependent upon its stiffness and damping characteristics, as well as the location of the bearings. A bearing located at a node can exercise no control. In a magnetic bearing the stiffness and damping coefficients can be varied by the use of position and velocity signals fed to the control amplifier. The bearing or actuator can be used in an open-loop or closed-loop configuration.

The characteristics of a magnetic bearing for the active control of rotor vibrations were examined by Schweitzer and Lange (ref.13) who derived a multivariable representation for these elements relating the output control force vector to the input vector which can be regarded as current or voltage. The objective is to control the force to counteract the disturbing forces acting on the rotor, for example, fluid forces, internal shaft friction or rotor out of balance. Some force terms cause instability, thus the control strategy must be capable of stabilising the rotor.

The simplest case dynamically is that of a rigid rotor. Bleuler and Schweitzer (ref. 6) described the use of two magnetic bearings to support a rigid rotor. Their scheme required the measurement, or estimation, of eight state variables and hardware to provide 32 feedback coefficients. That approach, if applied to a flexible rotor in which each mass station would be represented by eight state variables (four if only synchronous response is considered) is clearly prohibitive, thus Salm and Schweitzer (ref.5) examined the use of reduced-order models in the design of closed-loop algorithms for vibration control. They showed that unmodelled modes may lead to rotor instability, which can be avoided if the sensors and bearings are located at the same rotor station. This has a simple physical interpretation. The use of local negative feedback of position and velocity to control a magnetic bearing is **ideally** equivalent to replacing the device with **controllable passive** springs and dampers. This approach has been used in a number of commercial applications due to its simplicity and reliability, but it is conceptually unsatisfactory because the full capability of the magnetic device as an active control element is thereby sacrificed.

CONTROL-INDUCED INSTABILITY

The use of local feedback, though conceptually simple and in the **ideal** case inherently stable, may in practice lead to instability owing to the phase lag associated with sensors, signal processing and magnetic circuits. Bradfield et al (ref.14) described experiments on a supercritical shaft carried in rolling element bearings with a magnetic actuator sited in mid-span. They showed that the system could become unstable even when the electro-magnectic actuator was used to apply **nominally** pure damping forces.

The early work of Nyquist in studying closed-loop control systems highlighted the destabilising effect of phase-lag in a feedback loop. Thus the use of notch filters which have featured in some proposals to reduce synchronous vibrations (ref. 15) may cause instability due to phase lag. In a study of active mounts to isolate rotor induced vibration from a helicopter fuselage. Passalidis (ref.16) showed the inherent limitations of using a notch filter.

FEEDBACK CONTROL

Lim (ref.17) studied a rotor-bearing configuration similar to that used by Bradfield et al. Unlike them he did not incorporate a microprocessor in the feedback loop thereby reducing the phase lag in the feedback path. Lim used analogue feedback measurements of the rotor's transverse position and velocity. He employed a microprocessor outside these loops to vary the position and velocity gains to minimise a performance index at each rotor speed.

Lim did not report any cases of rotor instability. However, care must be taken even when adopting this approach because of the noise amplification caused by differentiation. In another project one of the author's students successfully (but inadvertently) measured shaft roughness. The signal, when differentiated, led to unexpected vibration of the rotor and swift operation of the panic button.

Details of Lim's rig are shown in Figure 1. The effectiveness of his approach is demonstrated in Figure 2. Although the rig was a small bench-top unit, Lim's experiments were used to explore the relative merits of closed-loop vibration control with adaptive open-loop control using the algorithm developed by Burrows and Sahinkaya (ref. 7 and ref. 9).

A typical result from station 10 is shown in Figure 2. The excellent fit between the simulated and experimental closed-loop responses was made possible by implementing a frequency-domain algorithm (ref. 18) to estimate system parameter values. The reduction in vibration amplitude using the open-loop algorithm is noticeable. Even more encouraging results were obtained in a parallel investigation involving a 100 mm diameter mild steel shaft of nominal length 2358 mm supporting two 90 mm thick, 406 mm diameter overhung steel discs carried in oil-film journal bearings. The rig has been described elsewhere (ref. 8). The rotor is shown in Figure 3 which also defines the mass stations used. In the experiments, a mid-span electromagnetic actuator was used to control vibration using full active control as outlined later. Chan Hew Wai and Morel (ref. 2) studied a similar arrangement but used only local velocity feedback to damp out shaft vibrations.

OPEN-LOOP ADAPTIVE CONTROL

In their original paper Burrows and Sahinkaya (ref. 7) developed a computationally fast and efficient least-squares algorithm to minimise the vibration of any general rotor-bearing system by the **application of external control forces**. This open-loop adaptive strategy avoids the possibility of instability associated with closed-loop structures for vibration control. However, the use of oil-film bearings introduces the possibility of bearing-induced instability associated with oil-whirl. Sahinkaya and Burrows (ref. 19) have shown that the dual task of controlling the stability threshold and the synchronous response can be achieved by a single magnetic actuator.

In the original formulation, estimation of the control force was dependent upon knowledge of the system parameters and measurement of the unbalance response. These restrictions were removed in a later paper (ref. 9) which allows the control force vector to be determined on-line without any prior knowledge of system parameters. The **algorithm is self-adaptive** to account for changes in system parameters, for example, the oil-film bearing coefficients or rotor unbalance. The latter feature has led some commentators to describe the algorithm as providing automatic balancing, but it is more versatile than that, hence the preferred title 'open-loop adaptive control of rotor vibration'.

A detailed derivation of the algorithm is given elsewhere (ref. 7 and ref. 9). It can be summarised as follows.

Open-loop adaptive control algorithm

A rotor-bearing system can be modelled as a multimass linear system. The k mass stations used in the cases considered in this paper are shown in Figures 1 and 3. If $\boldsymbol{Q}(j\omega)$ represents the complex displacement vector at these stations, then the synchronous response is given by

$$\boldsymbol{Q}(j\omega) = \Delta(j\omega)\,\boldsymbol{F}(j\omega) \tag{1}$$

where $\Delta(j\omega)$ is the receptance matrix of order $2k \times 2k$, given by

$$\Delta(j\omega) = \left(-\boldsymbol{M}\omega^2 + \boldsymbol{K} + \mathbf{C}(j\omega)\right)^{-1}. \tag{2}$$

where $\mathbf{C}, \boldsymbol{K}, \boldsymbol{M}$ are damping, stiffness and mass matrices, and the force vector is of order $2k$.

The force vector consists of components due to the distributed out-of-balance force, fluid flow-force etc denoted by $\boldsymbol{F}_d$, and the external force $\boldsymbol{U}(j\omega)$ applied by the electromagnetic actuator, that is

$$\boldsymbol{F}(j\omega) = \boldsymbol{F}_d(j\omega) - \boldsymbol{U}(j\omega) \tag{3}$$

The control force vector is generated by p control force variables where $1 \le p \le 2k$. Thus we can define a receptance matrix $\boldsymbol{R}(j\omega)$ of order $2k \times p$ which contains the columns of $\Delta(jw)$ corresponding to the control force locations.

The steady-state measured response vector $\boldsymbol{Q}_m(j\omega)$ can be written as

$$\boldsymbol{Q}_m(j\omega) = \boldsymbol{Q}_d(j\omega) - \boldsymbol{R}(j\omega)\,\boldsymbol{U}(j\omega) \tag{4}$$

where $\boldsymbol{Q}_d(j\omega)$ is the response due to forces other than the control force.

If the control force vector is changed by $\delta\boldsymbol{U}(j\omega)$, then the new steady-state displacement vector is

$$\boldsymbol{Q}(j\omega) = \boldsymbol{Q}_m(j\omega) - \boldsymbol{R}(j\omega)\,\delta\boldsymbol{U}(j\omega) \tag{5}$$

The objective is to determine $\delta\boldsymbol{U}(j\omega)$ to minimize $\boldsymbol{Q}(j\omega)$. Thus equation (5) is rewritten as

$$\boldsymbol{Q}_m(j\omega) = \boldsymbol{R}(j\omega)\,\delta\boldsymbol{U}(j\omega) + \boldsymbol{Q}(j\omega) \tag{6}$$

If the structural parameters are known and $\boldsymbol{Q}_m(j\omega)$ is measured, equation (6) can be regarded as a linear stochastic equation with the unknown parameters $\delta\boldsymbol{U}(j\omega)$ and the error term $\boldsymbol{Q}(j\omega)$.

The estimated change in $\boldsymbol{U}(j\omega)$ to minimize $\mathcal{Q}(j\omega)$ as given by a least-squares estimator (ref.18) is

$$\delta\hat{\boldsymbol{U}}(j\omega) = \left(\overline{\boldsymbol{R}}^T(j\omega)\boldsymbol{R}(j\omega)\right)^{-1}\overline{\boldsymbol{R}}^T(j\omega)\mathcal{Q}_m(j\omega) \tag{7}$$

where $\overline{\boldsymbol{R}}(j\omega)$ represents the complex conjugate and T denotes the transpose.

The difficulty of assigning reliable values to the elements of $\overline{\boldsymbol{R}}(j\omega)$, particularly when the stiffness and damping matrices include terms due to oil-film coefficients, can be overcome by estimating the receptance matrix on-line.

If the trial force is applied to the rotor in the x direction at the ith station, the response can be written from equation (5) as

$$\mathcal{Q}_x(j\omega) = \mathcal{Q}_m(j\omega) - \boldsymbol{R}_x(j\omega)\,\delta\boldsymbol{U}_x(j\omega) \tag{8}$$

$\mathcal{Q}_x(j\omega)$ denotes the measured displacement vector after application of a complex force along the x axis. Vector $\mathcal{Q}_m(j\omega)$ is known and $\delta\boldsymbol{U}(j\omega)$ is known, hence $\boldsymbol{R}_x(j\omega)$ can be calculated from equation (8).

A trial force is then applied in the y direction at the ith station to give

$$\mathcal{Q}_y(j\omega) = \mathcal{Q}_m(j\omega) - \boldsymbol{R}_y(j\omega)\,\delta\boldsymbol{U}_y(j\omega) \tag{9}$$

This allows $\boldsymbol{R}_y(j\omega)$ to be calculated.

When the process is initiated $\boldsymbol{U}(j\omega) = 0$ and $\mathcal{Q}_m(j\omega) = \mathcal{Q}_d(j\omega)$

The control algorithm can be summarised as follows.

(i) Sample displacements and perform a FFT to construct $\mathcal{Q}_m(j\omega)$

(ii) Set $\delta\boldsymbol{U}_x = \begin{vmatrix} u_x \\ o \end{vmatrix}$ sample displacements and perform a FFT to construct $\mathcal{Q}_x(j\omega)$

(iii) Estimate $\boldsymbol{R}_x(j\omega)$ from equation (8)

(iv) Set $\delta\boldsymbol{U}_y = \begin{vmatrix} o \\ u_y \end{vmatrix}$ sample displacements and perform a FFT to construct $\mathcal{Q}_y(j\omega)$

(v) Estimate $\boldsymbol{R}_y(j\omega)$ from equation (9)

(vi) Estimate the optimum force adjustment from equation (7)

(vii) Calculate statistical data on $\delta\hat{\boldsymbol{U}}$ (goodness of fit and standard deviation) to assess its validity. If not suitable go to (i), if suitable $\delta\boldsymbol{U} = \delta\hat{\boldsymbol{U}}$

(viii) Monitor speed and displacements. Whenever updating is required go to (i).

It is not feasible to measure the displacement at all of the mass stations, thus $\boldsymbol{Q}(j\omega)$ can be interpreted in terms of the response at the measured station. The selection of measurement sites is critical in terms of the quality of control. This has been discussed by Burrows et al (ref. 8 and ref. 9).

It is important to note that a background program ensures that the synchronous control forces are continuously fed to the magnetic actuator independent of the speed of computation of the control algorithm i.e. the previous value of $\boldsymbol{U}$ is retained until a change is required. The statistical data obtained from the Least Square estimator is discussed in detail in by Burrows and Sahinkaya (ref. 7).

EXPERIMENTAL RESULTS

A typical result is shown in Figure 4, this demonstrates the effectiveness of the algorithm. For example, the amplitude response at the critical speed is reduced in Figure 4b by a factor of 30. The reduction in vibration achieved by the algorithm depends upon which mass station is considered as shown by comparing Figures 4 and 5 which are for mass station 7 and 12 respectively. It also depends upon where the displacement transducers are sited.

In practice only a limited number of displacement transducers will be available and the choice of site along the rotor is critical. The results in Figures 4 and 5 show the effect of using different measurement sites; where:

$j = 1$ denotes control measurement sites 3 and 12

$j = 2$ denotes control measurement sites 3, 7 and 12

Other locations are considered by Burrows, Sahinkaya and Clements (ref. 9).

CONCLUSIONS

The open-loop adaptive control algorithm development by the author and his co-workers provides an efficient method of controlling the vibration of rotors without the need of a prior knowledge of parameter values. It overcomes the disadvantages normally associated with open-loop control whilst avoiding the problem of instability associated with closed-loop control algorithms.

The algorithm is conceptually satisfying because it utilises the capability of magnetic bearings as fully active vibration control elements rather than limiting them to act as adjustable stiffness and damping elements, as is the case when they are used with local position and velocity feedback.

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REFERENCES

1. Foster, E., Kulle, V. and Peterson R. The Application of Active Magnetic Bearings to a Natural Gas Pipeline Compressor. ASME paper 86-GT-1, Int. Gas Turbine Conf. and Exhibit., Dusseldorf, West Germany, June 8-12, 1986

2. Chan Hew Wai, C. & Morel, J. Application of an Active Magnetic Bearing for Vibration Control Near Critical Speeds in Large Rotating Machines. ASME Eleventh Biennial Conf. on Mechanical Vibration and Noise, Boston, Massachusetts, pp 151-157, 1987.

3. Heidner, M. and Kabelitz, H.-P. Reliability of Turbomolecular Vacuum Pumps: A Comparison of Rolling Element and Magnetic Bearing Systems. Jnl. Vac. Sci Technol. A8 (3), May/June 1990.

4. Schweitzer, G. Stabilization of Self Excited Rotor Vibrations by an Active Damper. Dynamics of Rotors (ed. F. I. Niordson), pp 472-493, Berlin : Springer-Verlag, 1975.

5. Salm, J. & Schweitzer, G. Modelling and Control of a Flexible Rotor with Magnetic Bearings. IMechE. Third Int. Conf. Vibs. in Rotating Machinery, paper no. C277/84, pp 553-561, York, 1984

6. Bleuler, H. & Schweitzer, G. Dynamics of a Magnetically Suspended Rotor with Decentralised Control. IASTED Applied Dynamics and Control Symposium, Copenhagen, 1983.

7. Burrows, C. R. and Sahinkaya, M.N. Vibration Control of Multi-Mode Rotor-Bearing Systems. Proc. R. Soc. Lond. A**386**, pp 77-94, 1983.

8. Burrows, C.R., Sahinkaya, M.N. and Clements, S. Electromagnetic Control of Oil-Film Supported Rotors Using Sparse Measurements. Am. Soc. Mech. Engrs. Jnl. Vib. Acoustics, Stress and Reliability in Design, **110**, pp 293-299, 1988.

9. Burrows, C.R., Sahinkaya, M.N. and Clements, S. Active Vibration Control of Flexible Rotors: An Experimental and Theoretical Study. Proc. R. Soc. Lond. A**422**, pp123-146, 1989.

10. Nonami, K. and Kawamata, S. Feedforward Control to Unbalance Force Cancellation for Flexible Rotor Systems. Proc. Int. Symp. on Advanced Computers for Dynamics and Design, Tsuchiura, Japan, Sept 6-8, 1989.

11. Higuchi, T., Mizuno, T. and Tsukamoto, M. Digital Control System for Magnetic Bearings with Automatic Balancing. Proc. 2nd Int. Symp. on Magnetic Bearings, Tokyo, Japan, July 12-14, 1990.

12. Darling , J. and Burrows, C.R. The Control of Propeller-Induced Vibrations in Ship Transmission Shafts. Proc. 2nd Int. Symp. on Magnetic Bearings, Tokyo, Japan, July 12-14, 1990.

13. Schweitzer, G. and Lange, R. Characteristics of a Magnetic Rotor Bearing for Active Vibration Control. IMechE Conf. on Vibs. in Rotating Machinery, paper no.C239/76, Cambridge, 1976.

14. Bradfield, C.D., Roberts, J.B & Karawirdiran, R. Performance of an Electromagnetic Bearing for the Vibration Control of a Supercritical Shaft. Proc. IMechE **201**, pp 201-212, 1987.

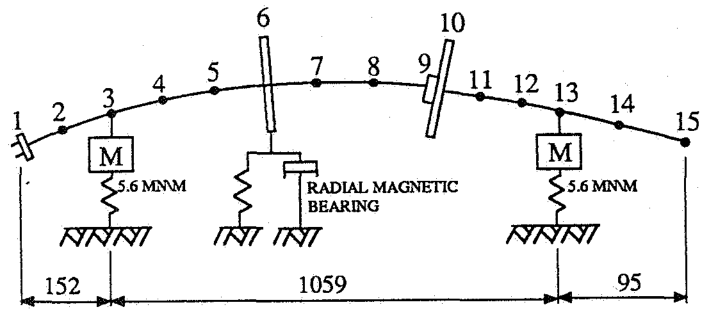
15. Haberman, H. and Brunet, M. The Active Magnetic Bearing Enables Optimum Damping of Flexible Rotors. ASME Paper 84-GT-117, 1984.

16. Passalidis, G. Active Control of Vibration of Helicopter Structures. PhD Thesis, University of Strathclyde, 1988.

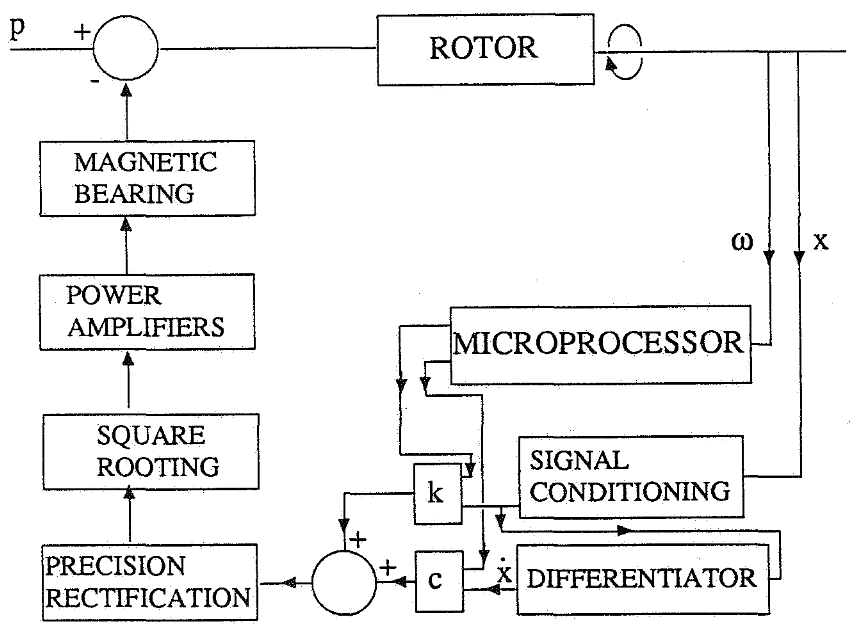
17. Lim, T.M. The Development of a Simple Magnetic Bearing for Vibration Control. PhD Thesis, University of Strathclyde, 1987.

18. Burrows, C.R. and Sahinkaya, M.N. a: Frequency Domain Estimation of Linearized Oil Film Coefficients. Jnl. Lubr. Technol. **104**, pp 210-215, 1982 b: Parameter Estimation of Multi-Mode Rotor-Bearing Systems. Proc. R. Soc. Lond. A**379**, pp 367-387, 1982.

19. Sahinkaya, M.N. and Burrows, C.R. Control of Stability and the Synchronous Vibration of a Flexible Rotor Supported on Oil-Film Bearings. Am. Soc. Mech. Engrs J. Dyn. Syst. Meas. Control **107**, pp 139-144, 1985.



SHAFT DIAMETER = 15 MM
DISC DIAMETER = 100 MM
ROTOR MASS = 3.92 kg
UNBALANCE MASS = 0.6915 x 10^{4} kgm. (station 10)



x : displacement
k : stiffness coefficient
c : damping coefficient

FIG. 1 Lim's experimental rig

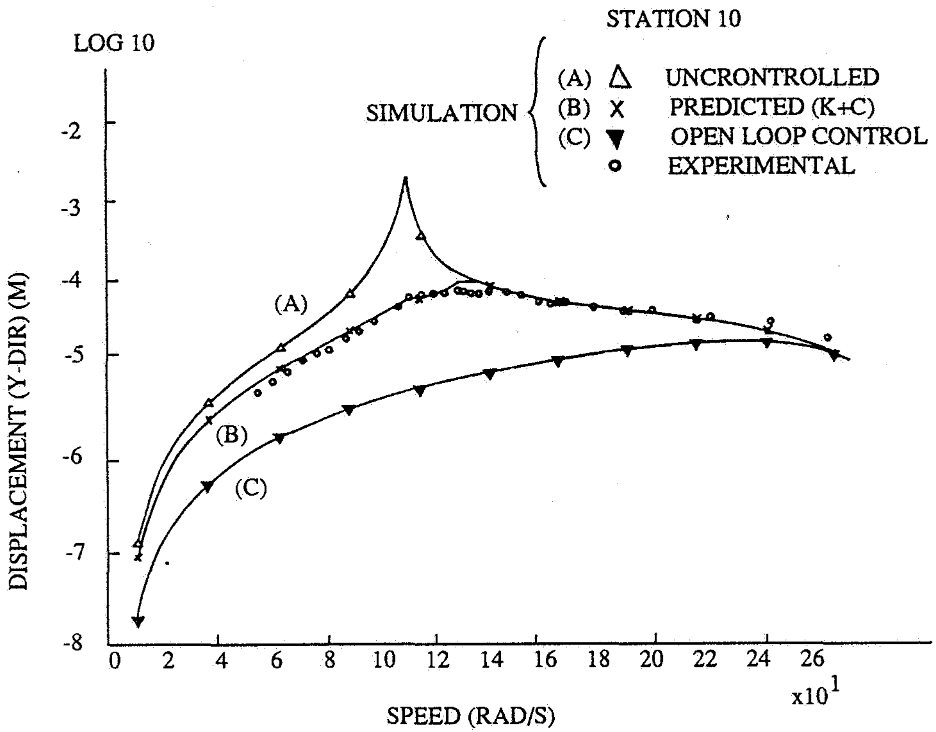


FIG. 2 Comparison of open-loop and closed-loop control (Lim's rig)

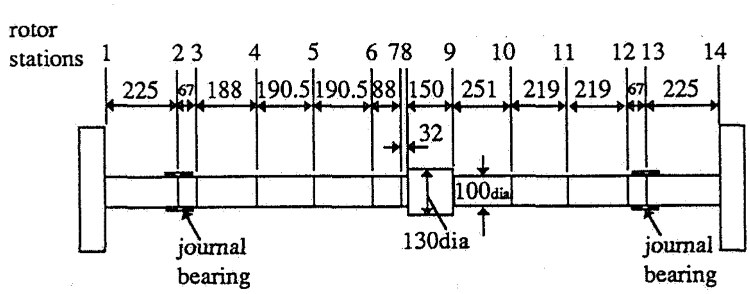


FIG. 3 Experimental rotor mass stations (Burrows, Sahinkaya, Clements)

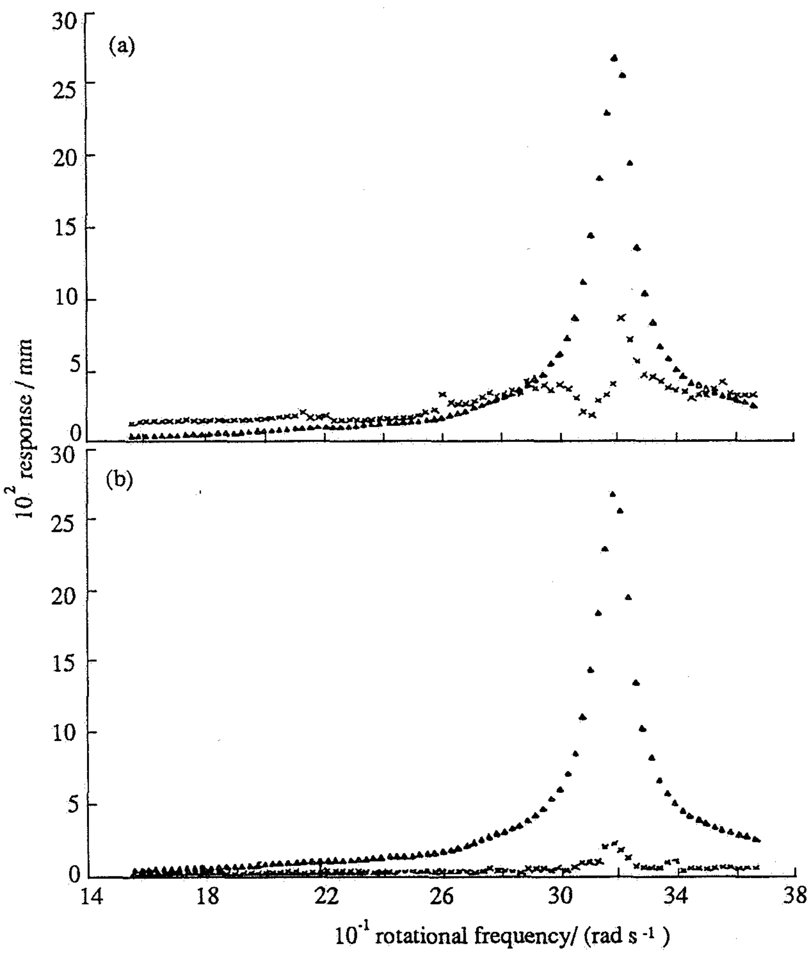


FIG. 4 Controlled and uncontrolled responses at station 7.
(a) $j = 1$ (b) $j = 2$
(Δ, uncontrolled; x, controlled)

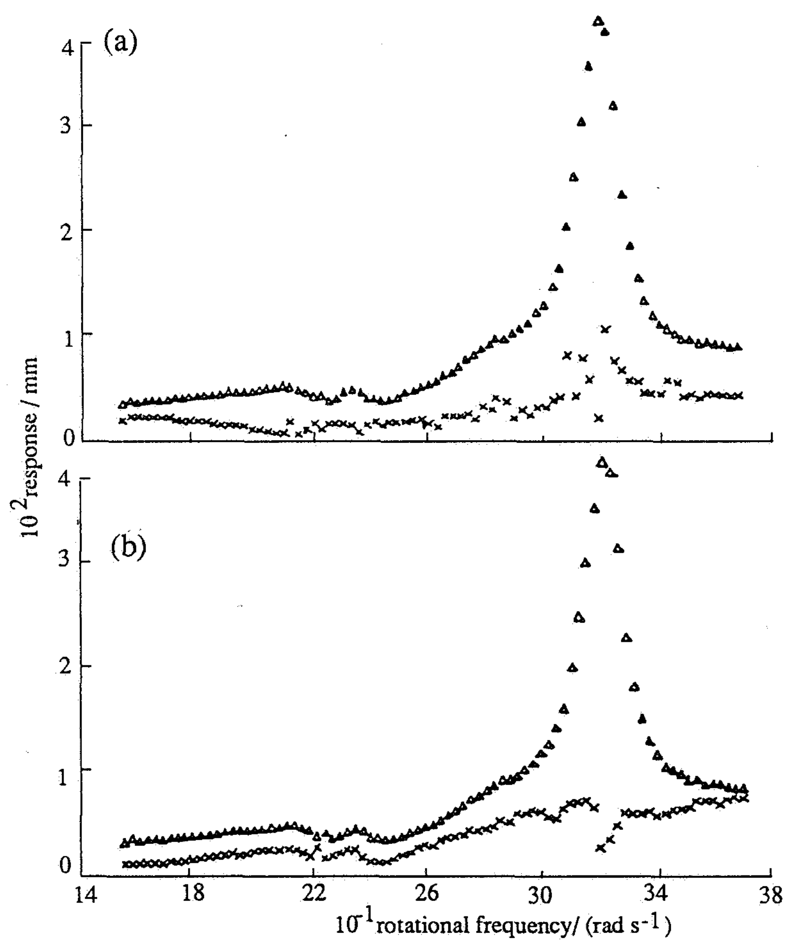


FIG. 5 Controlled and uncontrolled responses at station 12.
(a) j = 1 (b) j = 2
(Δ, uncontrolled; x, controlled)