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AUTOMATIC CONTROL OF MULTIPLE TUNED VIBRATION NEUTRALISERS

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1. INTRODUCTION

Since its inception by Ormondroyd and Den Hartog [1], the vibration absorber has been used in many applications [2]. It is usually used in one of two distinct ways; the first is where it is tuned to a problematic resonance of the host structure and the other is where it is tuned to a troublesome excitation frequency. In the latter case its purpose is to add a large mechanical impedance to the host structure at a single frequency of excitation with the aim of minimising the motion of the structure at this frequency, and to distinguish it from the tuned damper it is called a vibration neutraliser [3]. The main drawback of the neutraliser is that it is only effective over a limited frequency band, and the bandwidth and effectiveness of the device are functions of its internal damping. Recently, however, tunable vibration neutralisers have been developed, which are narrow-band devices that behave as conventional vibration neutralisers in the steady-state condition, but they can be tuned over a range of frequencies. von Flotow *et al* [4] have presented an overview on such devices. A device where the stiffness is changed by altering the pressure in a set of rubber bellows has been described by Longbottom *et al* [5] and is used in the experimental part of the work described here. A way of tuning the neutraliser by minimising the vibration of the host structure has been proposed by Ryan *et al* [6]. Long *et al* [7] chose to monitor the phase between the neutraliser mass and the host structure instead, because this signal contains additional directional information, i.e., whether the neutraliser is tuned above or below the excitation frequency. This control system was tested experimentally on the pneumatic neutraliser of reference [5] and the results reported in [7]. The work discussed in this paper is an extension of this study and examines the use of multiple neutralisers to control the vibrations of a system.

2. STEADY-STATE OPERATION OF A SINGLE NEUTRALISER

We first consider a single channel system, i.e., a single neutraliser fitted to a machine, as shown in figure 1. The machine and the neutraliser can be described by their mechanical impedances z_m and z_n, where the impedance of the neutraliser is given by:

$$z_n = \frac{j\omega m_n(1 + j\eta)}{1 - (\omega/\omega_n)^2 + j\eta} \quad (1)$$

where ω_n is the undamped natural frequency of the neutraliser ω is the excitation frequency m_r is the mass of the neutraliser and η is its hysteretic loss factor. The stiffness of the neutraliser is complex, such that $k^* = k(1+j\eta)$. If the neutraliser is tuned such that the undamped natural frequency is equal to the excitation frequency such that $\omega_n = \omega$, then assuming that $\eta \ll 1$, equation (1) reduces to:

$$z_{n(\text{tuned})} \approx \frac{\omega_n m_n}{\eta} \quad (2)$$

Provided that the impedance of the *tuned* neutraliser is much greater than that of the machine then the attenuation in the vibration of the machine at the frequency of excitation is given by [8]:

$$\text{Attenuation} = \frac{Z_{n(\text{tuned})}}{Z_{\text{machine}}} \quad (3)$$

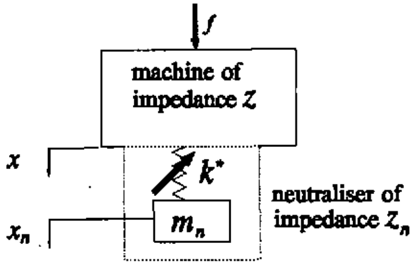

Figure 1 A machine with an adaptive tunable neutraliser fitted

Examining equations (2) and (3), it can be seen that for good attenuation the neutraliser is required to have a small loss factor and a large mass.

3. TUNING REQUIREMENTS

A practical way to tune the neutraliser is to force the motion of the machine and the neutraliser to be in quadrature. However, this is not optimal tuning as the maximum value of the modulus of the impedance in equation (1) occurs at a frequency of:

$$\omega_{max} = (1 + \eta^2)^{\frac{1}{4}} \omega_n \quad (4)$$

and the frequency when the machine and the neutraliser mass are in quadrature occurs at:

$$\omega_{quad} = (1 + \eta^2)^{\frac{1}{2}} \omega_n \tag{5}$$

If we divide equation (5) by equation (4), then we can quantify the amount by which the neutraliser is mistuned when it is adjusted such that the phase angle between the machine and the neutraliser mass is 90°. This is given by:

$$\frac{\omega_{quad}}{\omega_{max}} \approx 1 + \frac{\eta^2}{4} \tag{6}$$

It can be seen that if η is zero, then the neutraliser is perfectly tuned, and if η is large, i.e., 0.2, then the neutraliser is mistuned by only 1%. For this value of loss factor, the reduction in impedance due to imperfect tuning is less than 0.05 dB. Thus it can be seen that the penalty for setting the phase angle to 90° is small, and it will be seen that this is a convenient parameter to use in the control system. Because the phase angle between the machine and the neutraliser mass only changes rapidly in the vicinity of the natural frequency of the neutraliser, any control system using the phase angle alone will be inherently slow. To overcome this problem a two-tier control system is used as illustrated in **figure 2**. This consists of an open-loop system to roughly tune the neutraliser such that it is working within its 3 dB bandwidth and a closed-loop control system that takes over for accurate tuning. The open-loop control parameter can be determined before the neutraliser is used, as the required stiffness of the neutraliser, k is proportional to the square of the excitation frequency. The bandwidth of the neutraliser is approximately $\omega_n \pm \eta/2$. Thus it can be seen that although the loss factor should be small for the neutraliser to be effective (equation (2)), a small loss factor places greater demands on the control system. The closed-loop system uses a steepest descent method with the new stiffness being related to the old stiffness by:

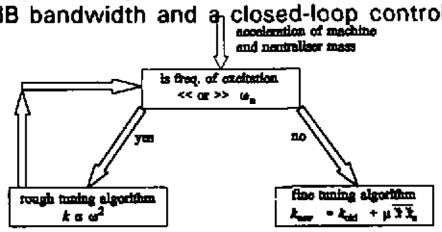

Figure 2 Control algorithm for the adaptive tunable neutraliser

$$k_{new} = k_{old} + \mu \overline{\ddot{x}\ddot{x}_n} \tag{7}$$

where μ is a convergence factor that determines the speed of control and $\overline{\ddot{x}\ddot{x}_n}$ is the time average product of the accelerations of the machine and the neutraliser mass. If the acceleration of the machine is given by

$\ddot{x} = \ddot{X}\cos(\omega t)$ and the acceleration of the neutraliser mass given by $\ddot{x}_n = \ddot{X}_n\cos(\omega t - \phi)$, then the time averaged product of these two signals is given by:

$$\overline{\ddot{x}\ddot{x}_n} = \frac{1}{T}\int_0^T \ddot{x}\ddot{x}_n \, dt = \frac{\ddot{X}\ddot{X}_n}{2}\cos\phi \qquad (8)$$

It can be seen that this is zero when the phase angle between the machine and the neutraliser mass is 90°. In this case the new stiffness is equal to the old stiffness and the device is tuned. If the tuned frequency is too low equation (8) will be positive, and the stiffness will be increased according to equation (7).

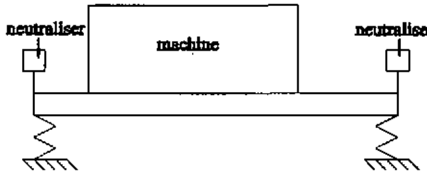

Figure 3 Configuration of a machine with more than one neutraliser fitted

Suppose now a more realistic system is chosen such as that shown in **figure 3** where the machine has more than one degree of freedom. Multiple neutralisers can be used to control the motion of the machine in this case. However, an important question is how much interaction can exist between the individual neutraliser control systems for the combined control system to continue to perform reliably. It has been found that the controllers do not interact significantly and thus can be adjusted independently, provided the *normalised* time average product is used as the fine tuning parameter in equation (7), which is equal to:

$$\text{norm}\{\overline{\ddot{x}\ddot{x}_n}\} = \frac{2\overline{\ddot{x}\ddot{x}_n}}{\ddot{X}\ddot{X}_n} = \cos\phi \qquad (6)$$

The fine tuning parameter for each neutraliser is thus simply equal to the phase angle between the neutraliser mass and the position on the machine where the neutraliser is fitted. If the neutraliser is linear, then this tuning parameter is a function of the excitation frequency and the properties of the neutraliser only. In this case the control system can be designed irrespective of the structure to be controlled and can be entirely decentralised.

4. EXPERIMENTAL WORK

The experimental work consisted of using two adaptive tunable neutralisers fitted to a stiff aluminium honeycomb plate as shown in **figure 4**. The control algorithm was implemented on a PC using Labview software. The plate was excited with a linear frequency sweep from 70-80 Hz in 5 seconds, and the control algorithm was implemented to automatically tune the neutralisers to the frequency of excitation. Acceleration signals were measured at positions adjacent to the neutralisers and on the neutraliser masses. These were conditioned before being fed into the PC via a National Instruments I/O board, AT-MIO-16F-5. Before being processed the signals were passed through identical bandpass tracking filters centred around the forcing frequency to remove unwanted noise.

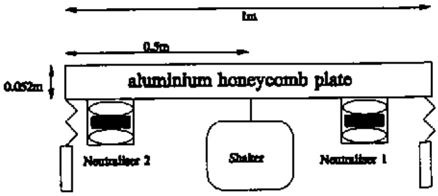

Figure 4 Experimental set up to test the control system for the neutralisers

The experimental rig was designed so that the plate would behave as a rigid body over the frequency range of interest. In this case the neutraliser attachment points would be well coupled and the independent control systems for each neutraliser could be tested under extreme conditions. **Figures 5a** and **b** show the experimental results.

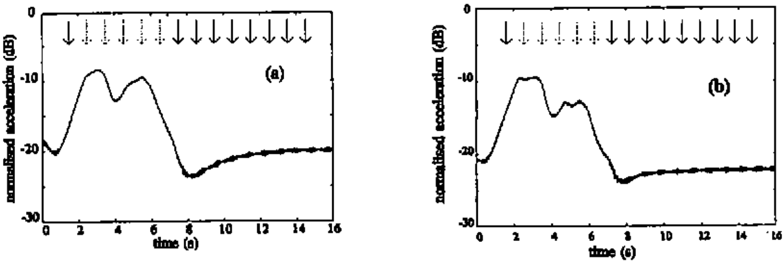

Figure 5 Normalised acceleration of plate at (a) position of neutraliser 1, and (b) position of neutraliser 2 during a frequency sweep of 70-80 Hz (dotted arrows - rough tuning, solid arrows - fine tuning)

The change in acceleration levels with both neutralisers being tuned simultaneously are depicted. The arrows on the graphs show when the coarse or fine tuning algorithms were operating. It had been established previously that the pneumatic neutralisers used in this experiment had non linear characteristics, since they tend to act as softening springs. Thus if the amplitude of vibration increases then the resonance frequencies of the devices shift to lower frequencies. The phase parameter used in the fine tuning control algorithm was not, therefore, completely independent of the action of the other neutraliser, i.e., the neutraliser control systems were not completely uncoupled. This coupling remains small, however, and the control system worked well. Similar experimental results were achieved when the excitation frequency was swept downwards, though in this case the neutralisers were slower to respond. This was a function of the neutraliser design and not the control system.

5. CONCLUDING REMARKS

Adaptive tunable neutralisers offer an alternative to both passive and active vibration control systems. They are relatively simple, inherently stable and can be controlled using much simpler systems than fully active systems. Their effectiveness is dependent upon the mass and damping in the neutralisers, and current research is focused on ways of using new materials and configurations to make more effective devices.

6. REFERENCES

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