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ACTIVE ISOLATION OF MULTIPLE STRUCTURAL WAVES ON A HELICOPTER GEARBOX SUPPORT STRUT

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A helicopter gearbox support strut has been set up in the laboratory under realistic loading conditions to investigate the active control of longitudinal and lateral vibration transmission to a connected receiving structure. Three magnetostrictive actuators were clamped to the strut to introduce secondary vibration in the frequency range 250–1250 Hz, the control objective being to minimize the kinetic energy of vibration of the receiving structure. Using an extensive set of frequency response measurements, it was possible to predict on a linear basis the attenuation in the kinetic energy of the receiving structure at any discrete frequency in the measurement range for a wide range of conditions. Calculations based on frequency response measurements showed that with the installed steel bearings on the strut, attenuations in the kinetic energy of the receiving structure of 30–40 dB were possible over a range of frequencies between 250 and 1250 Hz. At some frequencies in this range, notably around 500 Hz and 800 Hz, the control was less effective. This was due to torsional motion of the strut which was amplified by the secondary actuators. Good control was also predicted when the primary excitation to the strut was applied laterally rather than longitudinally.

Real-time active control has been implemented at discrete frequencies on the test strut and has generally confirmed the linear predictions. Attenuations in excess of 40 dB were measured in a number of cases. The tests confirmed that the active control of vibration transmission through a helicopter strut is practical at frequencies up to at least 1250 Hz.

1. INTRODUCTION

Successful noise control often depends on effective isolation of a vibrating machine from its support structure. If it is not suppressed at source, vibrational energy tends to propagate through a structure, causing unwanted sound radiation at some remote location. In the case of a helicopter such as the Westland/Agusta EH101, the main rotor drive and gearbox assembly are linked to four points on the passenger-carrying airframe structure by a total of eight cylindrical steel struts, of which four take the entire vertical load (Figure 1). The main function of these gearbox support struts is to maintain a rigid link between the drive

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unit and the fuselage. As a result, they offer little isolation and are the major transmission path for structural vibration.

The gearbox support struts provide an unwanted path both for low frequency vibration, mainly at blade passing frequency (17·5 Hz), and also for much higher frequency tonal vibration due to the meshing of gear teeth in the main drive unit (250–1250 Hz). The gear meshing tones contribute to high and disturbing noise levels for passengers. Because of the key structural role played by the struts, there is little scope to apply conventional passive isolation techniques in this application.

In the late 1980s, Westland Helicopters Ltd developed an active vibration isolation system operating at the 17·5 Hz blade passing frequency, termed ACSR (active control of structural response) [1]. In the ACSR system each strut was modified to incorporate a single hydraulic shaker in parallel with a compliant ring. Driving the four shakers to minimize airframe vibration as measured by ten accelerometers, the active controller reduced average vibration levels by around 75%.

In this paper are presented the results of laboratory tests carried out to investigate the possibility of isolating the higher frequency gear meshing tones using active techniques. The tests were carried out on an actual helicopter strut set up in the laboratory under realistic static loading conditions. Active isolation of high frequency structural vibration is a challenging problem, because many more vibrational modes of the strut are excited at high frequencies. It has been clear from previous work [2] that more than a single actuator would be required to control the strut motion and to isolate the airframe from this subjectively annoying tonal noise source. In this paper the types of wave which can propagate on the strut (a hollow cylinder) are discussed and we go on to discuss the appropriate location of sensors and actuators in the light of published work on active control of multiple structural waves. A multiple-channel feedforward control philosophy has been adopted, and frequency response measurements are used as a basis for prediction of its performance over a wide range of discrete frequencies. Finally, an adaptive multiple-actuator control system is implemented at selected discrete frequencies.

Research work on a multiple-actuator active isolation system has formed part of a BRITE–EURAM European research initiative (RHINO) involving Westland Helicopters Limited, the Italian manufacturers Agusta, the U.K. Defence Research Agency and others. The objective has been to demonstrate the principles involved rather than to engineer a complete working system.

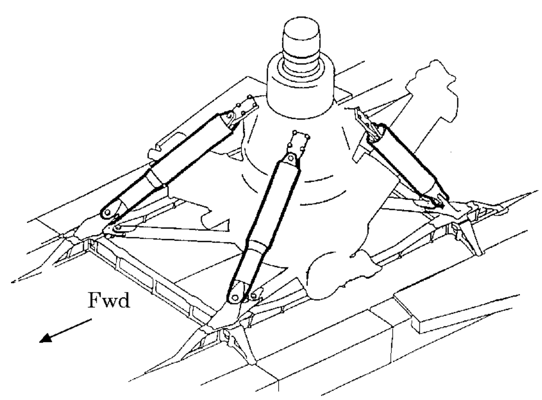

Figure 1. The helicopter main rotor gearbox installation, showing the location of the support struts.

2. ACTIVE ISOLATION OF STRUCTURAL WAVES

In recent years a number of writers have considered the active control of single and multiple wave types on an infinite beam, and also on other more-or-less idealized structures [3–11]. Although the principles have become well-established, few applications have been reported on practical engineering structures.

The active control of flexural waves alone propagating on a thin infinite beam has been studied by Mace [3], Elliott and Billet [4] and others. The fourth order flexural wave equation gives rise both to dispersive propagating waves and to near-field evanescent waves which may have a significant field of influence at low frequencies. If the error sensor (e.g., an accelerometer) is placed less than three-quarters of a wavelength from the secondary source, both the evanescent and the propagating components will contribute significantly to the measured signal. In this case a single secondary actuator set to drive the error sensor output to zero will attenuate the propagating wave by at most 20 dB. If the error sensor were colocated with the secondary actuator, only a 3 dB reduction in the propagating wave amplitude would be achievable [4]. Two control forces or moments are needed to suppress both wave types downstream of the secondary source array.

A general approach to the active control of multiple wave types on a finite structure has been presented by von Flotow [6, 7] in the context of large flexible spacecraft structures. Subsequently, a theoretical study of the control of multiple wave types on an infinite aluminium beam was carried out by Pan and Hansen [8]. The dimensions of the beam were chosen to be such that it could support four wave types; flexural waves in two planes, and longitudinal and torsional wave motions. Control forces were applied to the beam at various angles and were adjusted to minimize the power flow at a point on the beam. The study showed that if a single primary force was set up to excite all the wave types, then generally none of the waves would be completely cancelled by a single control force minimizing power flow. However, if the control force was oriented to excite just one wave type (e.g., a transverse control force to excite flexural motion in one plane) then, in the absence of scattering, that wave type would be suppressed completely, leaving the others unchanged. A conclusion of the study, which assumed no coupling between the various wave types, was that effective control of all the waves would require one actuator per wave type. An experimental investigation confirming the predictions was reported in reference [9].

Gardonio and Elliott [10] have examined the control of *coupled* flexural and longitudinal waves on a thin beam of finite length, using the calculation technique for structural networks presented in reference [7]. One end of the beam was terminated by an asymmetric mass which scattered longitudinal waves into flexural waves. The control of the coupled wave motions was analyzed assuming that a pair of piezoceramic actuators and two error sensors measuring axial and transverse motion. When the primary force was set up to excite both longitudinal and flexural waves on the beam, the study showed how control of only one wave type results in a standing wave along the beam of *both* wave types, due to scattering of the uncontrolled wave motion at the termination. Control of both the axial and the transverse displacement at a point on the beam was necessary to suppress the propagating waves completely. Experimental measurements supporting this study are presented in reference [11].

In practice, it will be seen that a helicopter strut is far from being an idealized structure, not least because of the spherical steel bearings at each end which link it to the rest of the craft. Nevertheless, the published work on finite and infinite beams forms a basis for interpreting the measurements made on a practical structure.

3. WAVE TYPES ON A HELICOPTER STRUT

The EH101 strut used for the tests is shown in Figure 2. In manufacture it is formed from the solid and comprises a central hollow cylinder of outer diameter 74·5 mm and thickness 2·7 mm, with winged sections at the ends to support the bearings. Dispersion curves for waves propagating on a uniform cylindrical shell are well established [12]. The number and characteristics of the wave types depend strongly on how the applied frequency compares with the *ring frequency* f_r of the cylinder, given by $f_r = c_l'/2\pi a$ (c_l' is the phase speed of longitudinal waves in a thin plate of the material; $2\pi a$ is the circumference of the cylinder). At this frequency the cylinder vibrates in a breathing mode. In the case of the helicopter strut, the ring frequency is 22·5 kHz, while the frequencies of interest for active control (250–1250 Hz) are much lower.

At low frequencies (less than 2000 Hz) there are five independently propagating wave types on the strut which fall into three groups [13]: axisymmetric waves (termed $n = 0$); bending waves, in which the cylinder section is displaced but not distorted ($n = 1$) and circumferential modes, in which the cross-section is distorted ($n = 2, 3, \ldots$). The types which apply in the case of the helicopter strut are as follows:

$n = 0$. Longitudinal waves similar to those in a slender rod. Torsional waves.

$n = 1$. Bending waves in two planes. An evanescent near-field component is associated with each wave.

$n = 2$. The first circumferential bending mode of the cylinder cross-section (the "ovalling mode") is estimated to cut in at 1307 Hz for the strut under test. The ovalling mode will not propagate below this frequency, but an evanescent contribution can be expected.

The various wave types do not all result in forces of similar amplitude being applied to the receiving structure. Brennan [14] analyzed the propagation of longitudinal and bending waves on a finite-length strut, which was assumed (for simplicity) to be pinned at one end to a rigid receiving structure. In this analysis the main gearbox at the top of the strut was represented as a constant velocity source in the longitudinal or lateral direction, as appropriate. The strut was modelled as a rod for longitudinal disturbances and as a thin beam for lateral disturbances. On this basis it was possible to calculate the transfer impedance of the strut in each direction; that is, the force applied to the rigid receiving structure per unit velocity applied at the gearbox. This analysis indicated that at most frequencies below 2 kHz the longitudinal forces would dominate but that at flexural modes (predicted to be at 300 Hz and 1200 Hz in this case) the lateral forces could exceed the longitudinal forces for a given velocity input.

The outcome of these initial calculations was a decision to use three actuators for experimental studies of active control of the strut. It was considered that torsional motions were unlikely to be excited to a great extent in practice, and ovalling and other circumferential modes cut in at too high a frequency to affect the control significantly. The actuators were to be capable of controlling at the same time longitudinal disturbances and lateral disturbances in two planes.

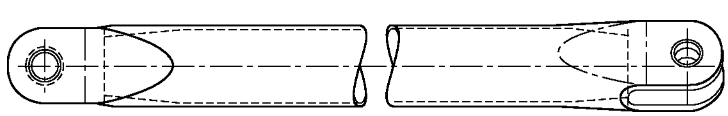

Figure 2. The main gearbox mounting strut used for the active control tests.

4. EXPERIMENTAL TEST ASSEMBLY

A test assembly to measure the dynamic response of a helicopter strut under realistic static loading conditions was designed and developed by the U.K. Defence Research Agency at Farnborough and is described in [15]. In Figures 3 and 4 is shown the test rig, in which a single EH101 strut complete with bearing assemblies was attached to two endplates and excited by a shaker. The endplates were suspended by a system of stretched elastic supports designed to impose static loads up to 4 tonnes force. The elastic supports also served as dynamic isolators: the natural frequency of the entire strut assembly moving vertically on its elastic supports was estimated to be not more than 13 Hz.

A Derritron VP4 shaker rated for 130 N operation provided the primary force input to the upper endplate and was applied at various angles for different tests. The applied force reacted against the mass of the shaker itself. The shaker was connected by a short length of piano wire to an impedance head device, which was fitted to a steel block attached to the upper face of the top endplate. The impedance head measured input force and acceleration. The steel block was formed from a 50 mm cube, with one edge bevelled at 30 degrees from the horizontal to allow the shaker to be fitted as close as possible to the vertical (z) direction without fouling the elastic cables. The x and (nominal) z inputs were coplanar with the strut axis; the y input was deliberately chosen to be off-axis to excite torsional motion of the strut assembly.

A second 50 mm steel cube was fitted to the underside of the lower endplate assembly. Six accelerometers were attached as indicated in Figure 3 to measure vibration of the lower endplate in all possible rigid body degrees of freedom. The mass of each endplate assembly

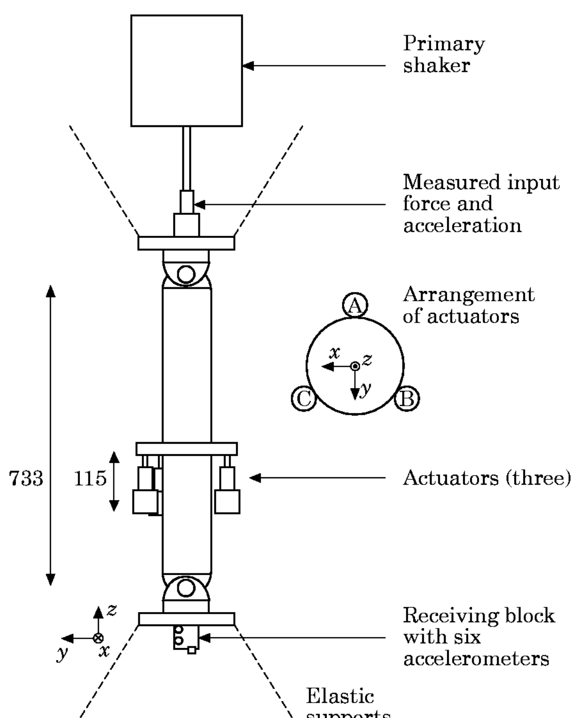

Figure 3. A sketch of the experimental rig used for active control measurements. Dimensions in mm.

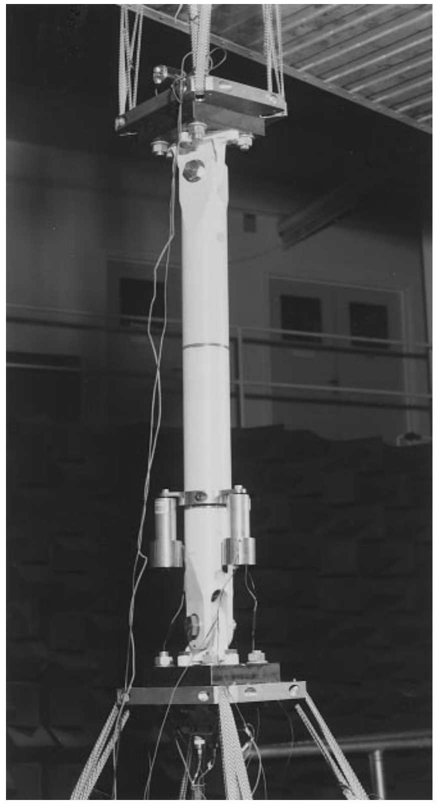

Figure 4. The experimental rig used for strut active control measurements. The three magnetostrictive actuators were attached to the strut externally by means of a collar.

was about 17·6 kg altogether, while the strut alone (excluding bearings and actuators) had a mass of about 6 kg. Spherical steel bearings at each end of the strut were designed to allow static rotation about all axes, but their dynamic response at the frequencies of interest in this application was found to be more complicated in practice [15].

The three secondary actuators were attached to a collar, which was clamped to the outside of the strut as shown in Figure 5. In a practical application this arrangement would ensure that the structural integrity of the strut is not compromised. The actuators were disposed symmetrically about the central axis of the strut and each one applied a longitudinal force to the collar. They were magnetostrictive devices, chosen because of their capacity to generate high forces over a wide frequency range from a compact and relatively light package. Magnetostrictive actuators are marketed commercially by Edge Technologies, Inc. [16]. An inertial actuator design was used allowing point forces to be applied to the structure. Acting together, the actuators were able to apply to the strut (1) longitudinal force along the strut axis (defined as the z-direction), (2) moment about the x-axis, and (3) moment about the y-axis; or any combination of these by linear superposition.

The collar could be loosened and moved to different positions along the strut. For the tests described below the actuators were positioned at one or other of the bands seen in Figure 4, corresponding to 25% of the strut length between bearing centres (lower position) or 60% (upper position).

5. ACTUATORS AND SENSORS FOR CONTROL

5.1. SECONDARY ACTUATORS

Each of the three secondary actuators was a magnetostrictive device, reacting against the mass of its own housing together with an additional attached steel endcap, to give a total inertial mass of 0·75 kg. The devices chosen for the experimental studies presented here were of type 50/6-MP, a packaged unit of length 100 mm containing a prestress spring and a permanent magnet to establish the appropriate bias conditions. The active element is a 50 mm rod of Terfenol-D rated for 50 μm peak-to-peak displacement, 490 N rated load and requiring a coil current of 1 A r.m.s.

Three factors influence the choice of location of the actuators along the length of the strut. First, earlier work on active wave control on a finite simply supported beam [5] had shown clearly that the secondary force required for active control could be expected to vary markedly with position. Flexural waves on the strut travelling downstream from the primary source are reflected back upstream by the secondary actuators, while reflection also occurs at the upstream termination. The result is a standing wave field, the amplitude of which depends strongly on actuator position. At any given frequency, the secondary force required to cancel downstream waves can therefore also be expected to vary strongly with actuator position. This feature of active wave control is discussed by Nelson and Elliott [17] in the context of one-dimensional sound propagation in a duct.

A second factor is the evanescent component of bending waves on the strut. As discussed by Elliott and Billet [4], the evanescent near field caused by the secondary actuators will

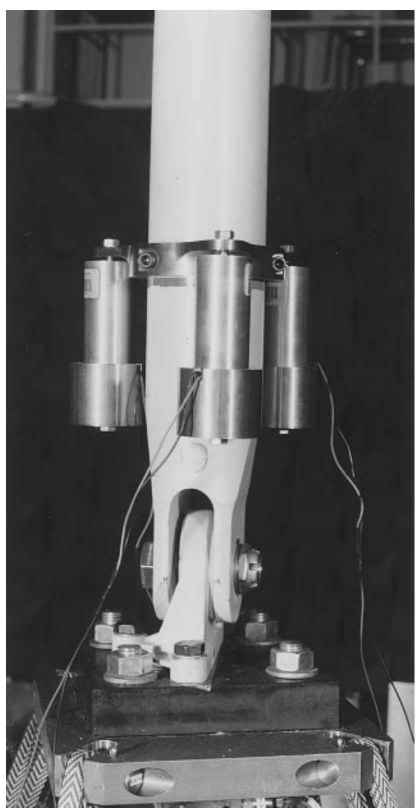

Figure 5. A detail of the experimental rig, showing the three secondary actuators.

affect the output of the error sensors if the actuators are placed too close to them. This would clearly be important if the error sensors were attached to the strut itself. The near field of the actuators may also affect the motion of the receiving structure, but this effect depends on the nature of the linkage attaching the strut to the receiving structure.

A third and final consideration arises if the impedance of the primary source is high, as may occur in practice where the primary vibration is due to a massive and stiff substructure, such as the helicopter main drive and gearbox. The conservative assumption is that the primary source behaves as a velocity source; that is, a source of infinite internal impedance, the velocity of which is unaffected by conditions on the strut. This upstream boundary condition then determines the magnitude of the required secondary force. For longitudinal waves in particular, where the wavelengths are longer than the strut, the secondary actuators must act against the compliance of the strut itself to accommodate the gearbox motion and bring the receiving structure to rest. Clearly, the axial stiffness of the section of the strut between the primary source and the secondary actuators will be much increased if the actuators are placed towards the top of the strut, closer to the primary. For a lower secondary force requirement, it would be preferable to place the actuators toward the bottom of the strut.

Summarizing, the preferred location of the three secondary actuators is influenced by the following:

(1) The flexural standing wave field between the primary source and the actuators: the best location (for minimum force requirement) depends strongly on frequency.

(2) Near field effects, which may affect the error sensors if the actuators are placed too close to them.

(3) The high impedance of the primary source, which would require that the secondary actuators be placed far from the primary source to avoid the need for very high secondary forces when controlling the longitudinal wave.

5.2. ERROR SENSORS

In the work on finite and infinite beams reviewed earlier, the error sensors for control were positioned on the beam itself, some distance downstream of the actuators, thus avoiding the flexural near field effects referred to above. Provided that a sufficient number of actuators were used to achieve control of all the wave types, vibrational energy would not then propagate down the beam to any receiving structure which might be attached. In the case of the helicopter strut, if all the wave types could be detected by accelerometers on the strut and actively suppressed by the actuators, then the receiving structure (e.g., the helicopter fuselage in a practical application) would be isolated from the disturbance without the need for detailed measurement of its vibration transmission characteristics. This strategy has the appeal of being local to the strut and relatively independent of the attached structure.

On the other hand, an alternative approach would be to measure and minimize the vibration of the receiving structure to which the strut is attached. It is the vibration of this structure and the resulting noise that is actually of concern. In practice, it was found that the best results were obtained by actually measuring the quantity to be attenuated; specifically, the vibration of the receiving structure. Six accelerometers were attached to the lower endplate assembly (the receiving structure for the laboratory experiments) and were oriented to detect all six of its rigid body degrees of freedom. A convenient cost function to be minimized by the control system was the kinetic energy of the lower endplate assembly, which is a quadratic function of the error signals.

6. ACTIVE CONTROL PREDICTIONS

6.1. FREQUENCY RESPONSE MEASUREMENTS

A set of frequency response measurements were carried out on the strut test assembly at DRA Farnborough in order to provide a basis for prediction of the response of the active control system. The strut test assembly described in section 2 was set up with loadings of 1 tonne and 4 tonnes and with actuator positions of 25% (lower position) and 60% (upper position). Measurements were carried out with the standard spherical steel bearings and also with an experimental pair of softer elastomeric bearings, which were fitted as part of a separate investigation into passive means of isolating helicopter vibrations. For each test configuration, the frequency response of the strut was measured with the primary shaker attached to the top endplate in the x, y and z directions in turn. In addition, the frequency response with respect to each of the three secondary actuators was also measured.

A detailed set of mobility measurements for the strut have been presented elsewhere [15] and will not be discussed in any detail here. Accelerometers attached to the lower endplate assembly typically showed around six or seven major resonant peaks in their response to applied primary force in the range 250–1250 Hz. Investigation of the strut motion showed that the spherical steel bearing at each end of the strut had a significant compliance, so that modes of the complete experimental set-up included mass–spring resonances involving the two endplates and the compliance of the bearings in addition to standing-wave phenomena on the strut itself.

As discussed below, the measured frequency responses of the rig have been manipulated to predict the response of the strut with a constant velocity source rather than the shaker which was actually attached. The required drive voltages to the secondary actuators have been included in this prediction.

6.2. CALCULATION METHOD

The helicopter gearbox meshing tones that are the subject of this study occur at only a few discrete frequencies. However, it is much more illuminating to examine the characteristics of an active control system with good resolution over a wide frequency range. If this is done, it becomes possible to draw general conclusions about its behaviour and limitations. The control characteristics of an experimental rig such as the strut considered here *can be predicted directly from the measured frequency responses* if linearity of the actuators, sensors and structure can be assumed. In this application it is necessary to accept that the characteristics of the magnetostrictive actuators in particular will change with operating level, but it remains possible to draw useful conclusions about the control system purely from linear operations on the measured frequency response data.

If a harmonic voltage $x(\omega)$ is applied at some frequency ω to the primary shaker, and at the same time three harmonic voltages $u_1(\omega)$, $u_2(\omega)$ and $u_3(\omega)$ are applied to the three secondary actuators A, B and C respectively, then assuming linearity the resulting acceleration of the lower endplate is the superposition of the accelerations due to each voltage applied separately. Thus, if $\mathbf{a}$ is a 6×1 vector of complex signals from the six accelerometers attached to the lower endplate, at frequency ω:

$$\mathbf{a}(\omega) = \mathbf{P}(\omega)x(\omega) + \mathbf{S}(\omega)\mathbf{u}(\omega), \tag{1}$$

where $\mathbf{u}(\omega) = [u_1 \;\; u_2 \;\; u_3]^{\mathrm{T}}$ is the vector of complex voltages applied to the actuators, $\mathbf{P}(\omega)$ is a 6×1 vector of measured frequency responses of each accelerometer to the primary drive voltage and $\mathbf{S}(\omega)$ is the 6×3 matrix of frequency responses of each accelerometer with respect to each of the three secondary actuators. The explicit dependence of the variables on ω will be dropped below for notational convenience.

In a practical situation, the vibration originates in the helicopter main drive/gearbox to which the strut is attached. As noted above, the internal impedance of this mechanism is unknown, but is likely to be high. The most conservative assumption is that the internal impedance of the gearbox is infinite; that is, its velocity is independent of any reaction forces presented to it by the strut. In this case linear predictions of the active control characteristics can be made provided that the input acceleration applied by the shaker during the frequency response measurements has been recorded. Division by $\mathrm{j}\omega$ then yields the input velocity.

The velocity at the point of application of the primary shaker to the upper endplate clearly depends on the voltage x applied to the primary shaker; however, it also depends on the voltages $\mathbf{u}$ applied to the secondary actuators, since each actuator excites the whole structure to some extent. If $v(\omega)$ is the input velocity at the top of the upper endplate (measured in the direction of the applied force), then

$$v = T_1 x + \mathbf{T}_2\mathbf{u}, \tag{2}$$

where T_1 is the measured frequency response of the input velocity v to the primary drive voltage x and the 3×1 vector $\mathbf{T}_2$ is the set of measured frequency responses of v with respect to each actuator drive voltage. In order to *simulate a constant velocity source in place of the shaker*, we will consider that v is maintained at a constant amplitude by adjustment of the primary drive voltage x. We can therefore rearrange equation (2) to express v as the independent variable, in which case x becomes a function of both v and of the secondary drive levels $\mathbf{u}$:

$$x = (v - \mathbf{T}_2\mathbf{u})/T_1. \tag{3}$$

The vector of accelerations of the lower endplate then becomes (from equation (1))

$$\mathbf{a} = (\mathbf{P}/T_1)(v - \mathbf{T}_2\mathbf{u}) + \mathbf{S}\mathbf{u}. \tag{4}$$

Rearranging, and writing for convenience

$$\mathbf{P}_1 = \mathbf{P}/T_1, \qquad \mathbf{S}_1 = \mathbf{S} - \mathbf{P}_1\mathbf{T}_2, \tag{5}$$

we have

$$\mathbf{a} = \mathbf{P}_1 v + \mathbf{S}_1\mathbf{u}. \tag{6}$$

The function of the active control system is to define the voltages $\mathbf{u}$ in such a way that the vibrational kinetic energy of the lower endplate is minimized at each frequency. The control signals are assumed to be derived from the input velocity so that

$$\mathbf{u} = \mathbf{w}v. \tag{7}$$

where $\mathbf{w}$ is a vector of controller responses selected to minimize a suitable cost function. The cost function in this case is chosen to be kinetic energy of the endplate structure for a given applied velocity:

$$J_v = (\mathbf{a}/v)^{\mathrm{H}}\mathbf{A}(\mathbf{a}/v), \tag{8}$$

in which $\mathbf{A}$ is a Hermitian weighting matrix, given by

$$\mathbf{A} = (1/4\omega^2)\mathbf{L}^{\mathrm{H}}\mathbf{M}\mathbf{L}, \tag{9}$$

where $\mathbf{M}$ is a diagonal matrix containing the mass of the block together with its moment of inertia about each axis; and $\mathbf{L}$ is a linear transformation which yields the translational

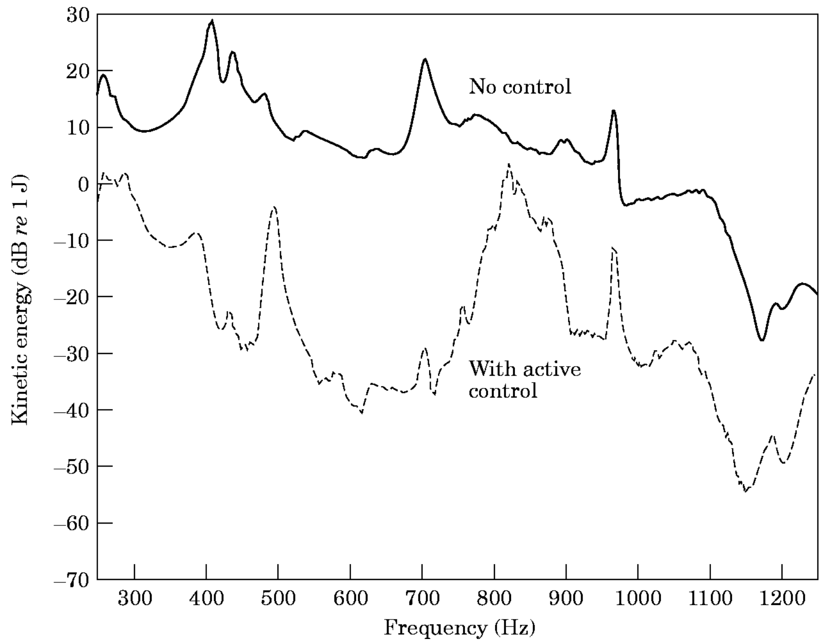

Figure 6. The calculated kinetic energy of lower endplate assembly without control (——) and with active control using three secondary actuators (– – – –).

and rotational acceleration of the block as a funtion of the measured accelerometer error signals. **L** depends purely on the locations of the accelerometers.

Combining equations (6), (7) and (8) yields a standard Hermitian quadratic expression for **w**. Minimizing J_v, the optimum controller and the minimum value J_{v0} of the cost function are given by [17]:

$$\mathbf{w}_{v0} = -(\mathbf{S}_1^H \mathbf{A} \mathbf{S}_1)^{-1} \mathbf{S}_1^H \mathbf{A} \mathbf{P}_1 \tag{10}$$

and

$$J_{v0} = \mathbf{P}_1^H \mathbf{A} \mathbf{P}_1 - \mathbf{P}_1^H \mathbf{A} \mathbf{S}_1 (\mathbf{S}_1^H \mathbf{A} \mathbf{S}_1)^{-1} \mathbf{S}_1^H \mathbf{A} \mathbf{P}_1. \tag{11}$$

6.3. LONGITUDINAL VELOCITY EXCITATION

The measured frequency responses of the strut test assembly can now be used to predict the attenuations in kinetic energy of the lower endplate which an active control system could achieve, using the method set out above.

By a simple geometric transformation, the six signals representing acceleration of the lower endplate have been used to calculate its motion in all six rigid body degrees of freedom: translation along the x-, y- and z-axes and rotation about these axes. Given the mass of the endplate assembly and its moment of inertia about each of the axes, the kinetic energy of vibration associated with each degree of freedom has been calculated. A constant amplitude velocity input of 1 m/s has been assumed for the prediction, applied in the longitudinal (z) direction with the strut load at 1 tonne, the actuator position at 25% and using steel bearings. The predicted total kinetic energy of the endplate is shown in Figure 6, without control (solid line) and with control (dashed line). The calculations, based on frequency response measurements, indicate that substantial attenuations are possible over a wide frequency range, reaching about 40 dB in the range 550–700 Hz, for example. At

500 Hz and around 800 Hz the attenuation is not as good. This feature will be discussed presently.

In Figure 7 the kinetic energy of the endplate without active control (upper trace in Figure 6) has been decomposed into four elements: (1) longitudinal (z), heavy solid line; (2) lateral-y (translation y + rotation about x-axis), dashed line; (3) lateral-x (translation x + rotation about y-axis), dash-dot line; (4) torsional (rotation about z-axis), heavy dotted line.

In Figure 7 it is shown that the kinetic energy associated with the longitudinal (z) direction dominates, as may be expected with predominantly z-directed excitation, although the other lateral components are not insignificant. The peak at 700 Hz is due to lateral motion in the x direction, which was excited because the shaker was actually angled at 30 degrees to the vertical in the x–z plane. The torsional energy is roughly 25 dB below the longitudinal energy across the whole frequency range; i.e., about 0·3% of the kinetic energy of the uncontrolled system is torsional.

In Figure 8 are shown the kinetic energy components with active control. Although the traces are to some extent superimposed, the main features are clear. The purely longitudinal (z) component of the energy is the lowest trace at most frequencies, and has been attenuated more effectively than any other motion (around 60 dB). The largest component of the kinetic energy after control at most frequencies above 400 Hz is the torsional component. The parallel configuration of the actuators is of course not designed to control torsional motion of the structure. Comparison of Figure 8 with Figure 7 shows that at some frequencies (e.g., 550–800 Hz) the torsional energy of the endplate is reduced along with other components, while at other frequencies it remains unchanged or is increased. Coupling between different wave types is inevitable on a complex engineering

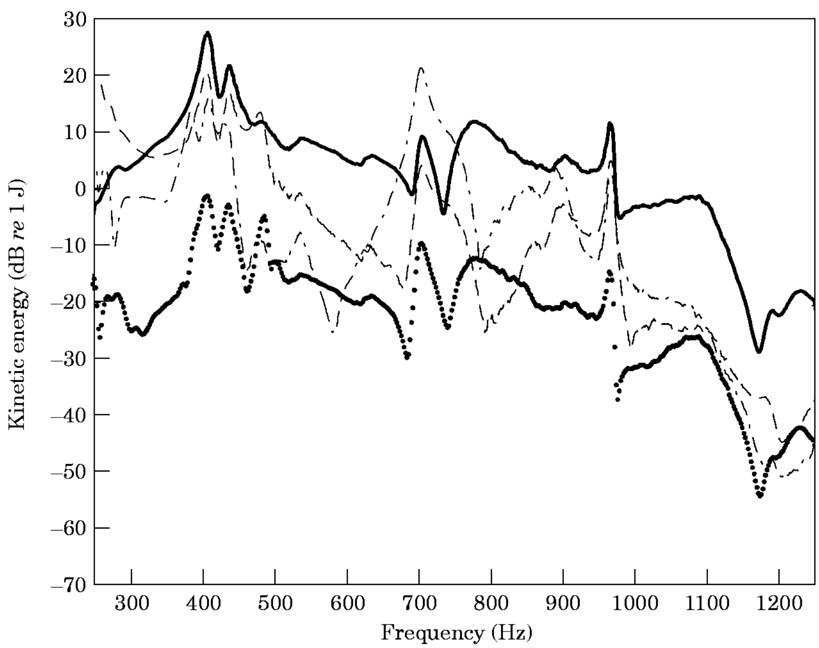

Figure 7. The calculated kinetic energy of the lower endplate without control (from Figure 6) decomposed into separate contributions. The heavy solid and heavy dotted lines correspond to longitudinal and torsional motion respectively; the light dashed and dash–dot lines correspond to lateral-y and lateral-x respectively.

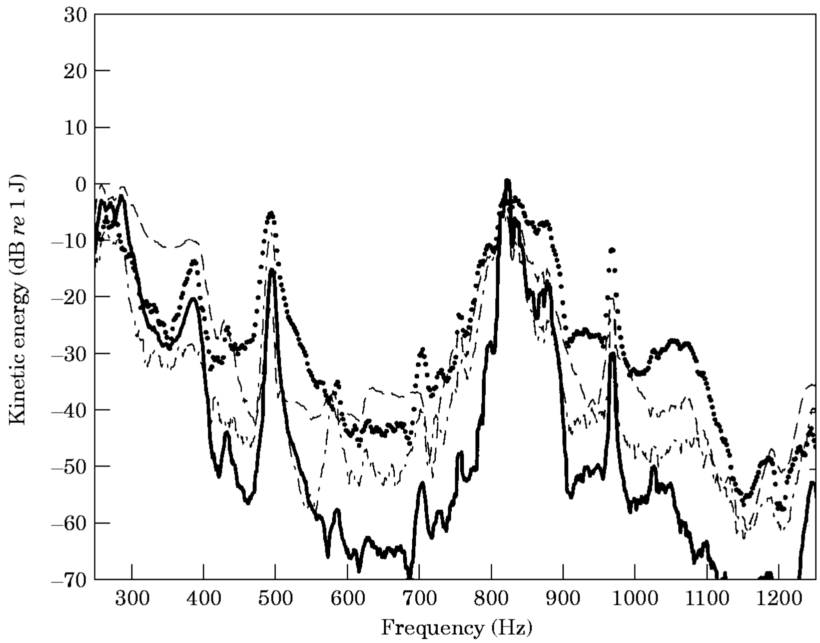

Figure 8. The calculated kinetic energy of the lower endplate with control (from Figure 6) decomposed into separate contributions due to longitudinal, lateral-x, lateral-y and torsional motion as in Figure 7.

structure such as the strut/endplate assembly and it is not surprising that torsional motion is excited. At 500 Hz and 850 Hz the torsional component of kinetic energy is apparently amplified by the secondary actuators, which are acting to control the predominantly longitudinal primary disturbance, resulting in the overall loss of attenuation at these frequencies in Figure 6. With only three independent actuators, it would not be possible to control four coupled independent wave types simultaneously.

These results are consistent with the findings of Gardonio and Elliott [10], who showed that where multiple wave types are coupled downstream of the secondary actuators (as they clearly are in this practical structure) then all the wave types must be suppressed to achieve downstream cancellation. The loss of control at 500 Hz and 850 Hz arises because there is no fourth actuator to provide independent control of torsional waves.

The root mean square of the three actuator drive voltages necessary to control the strut are shown in Figure 9. For these calculations the secondary actuators were in the lower position, giving the maximum length of strut above the actuators to be compressed. The scale in Figure 9 is volts per m s^{-1} amplitude of applied primary velocity. For a realistic helicopter vibration level, the demanded actuator voltage would exceed the rating of the actuators by about an order of magnitude at around 800 Hz. However, the lower demand level at around 600 Hz is within range, partly as a result of using an inertial design for the actuators: at this frequency the actuators are resonant and tend to behave as passive dynamic vibration absorbers. The high demand levels at 700–800 Hz may arise partly because the control of *lateral* components of the strut vibration requires high differential forces applied to the strut to create the necessary moment; also, as discussed earlier, high levels may be needed to control downstream waves in the presence of an upstream standing wave.

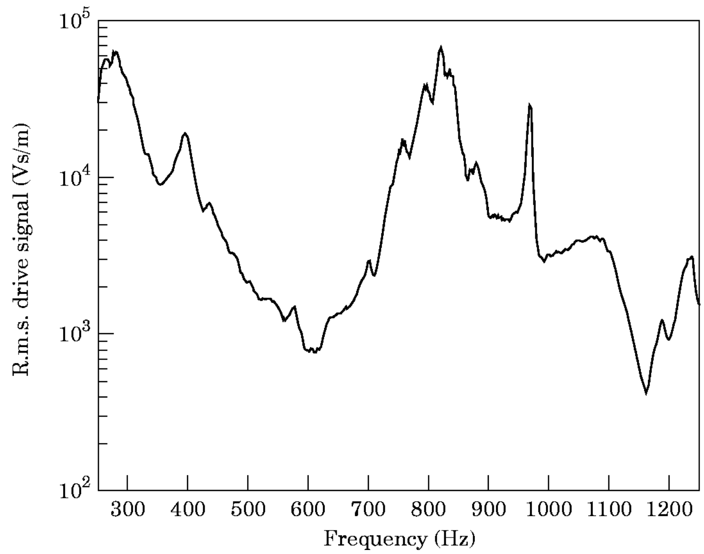

Figure 9. The root mean square of the three optimum actuator voltages required to minimize the kinetic energy of the lower endplate assembly. The strut configuration is as for Figure 6.

6.4. LATERAL EXCITATION AND THE EFFECT OF ACTUATOR POSITION

In Figure 10 is shown the predicted kinetic energy of the lower endplate assembly with and without control when the primary excitation is lateral (x-direction). The reductions of 30 dB or more are broadly similar to those achieved when the primary shaker was applied in the longitudinal direction. An analysis of the contributions to the kinetic energy

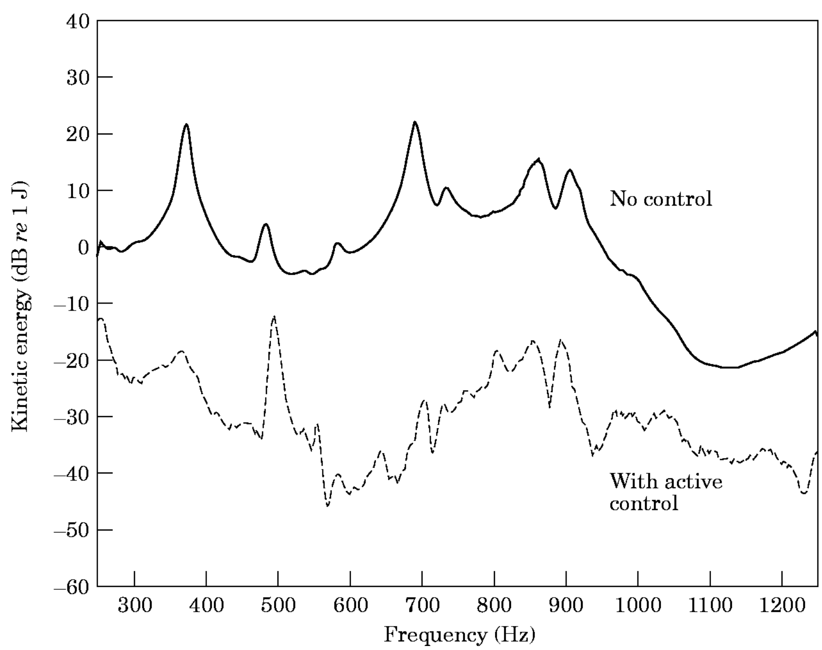

Figure 10. The calculated kinetic energy of the lower endplate assembly with lateral primary excitation. Otherwise the strut configuration is as for Figure 6. ——, Without control; – – – –, with active control using three secondary actuators.

(not shown) confirms that without control the lateral-x component dominates as expected. With control, the residual vibration is dominated by the torsional component as in the longitudinal case.

It is useful to compare the responses to lateral excitation obtained with the actuators placed in the lower and upper positions. In Figure 11 is shown the predicted kinetic energy of the lower endplate with the actuators placed in the upper position (heavy traces) and the lower position as in Figure 10 (light traces). It is clear that even with no control, simply moving the actuators affects the strut response significantly. The total mass of the assembly of secondary actuators is 3·1 kg, and if they are moved a change in the dynamic characteristics of the strut rig is to be expected. As a result, it is difficult to identify unambiguously the changes brought about purely by applying the secondary forces at a different point on the strut. However, the traces in Figure 11 show that the attenuation predicted with control using the actuators in the upper position is similar to or poorer than the results for the lower position over much of the mid-range of frequencies from 370 Hz to 1150 Hz. There is no indication that near field effects caused the control to deteriorate when the actuator assembly was in the lower position, although the wavelengths of bending waves on the strut were significant. At 400 Hz, for example, the wavelength of bending waves on the cylindrical part of the strut would be 1·4 m, so the lower bearing of the strut would be in the near field of the secondary actuators whichever position they were in. It is possible that the connecting bearing, which is not rigid, isolated the error sensors from near field effects [18].

A clear effect of raising the actuator is to increase the required drive level, as seen in Figure 12. The predictions assume a high impedance constant velocity source in the lateral direction, and so the secondary actuators are required to bend the strut to achieve control of the receiving structure. More force is required for active control when the actuators are in the upper position.

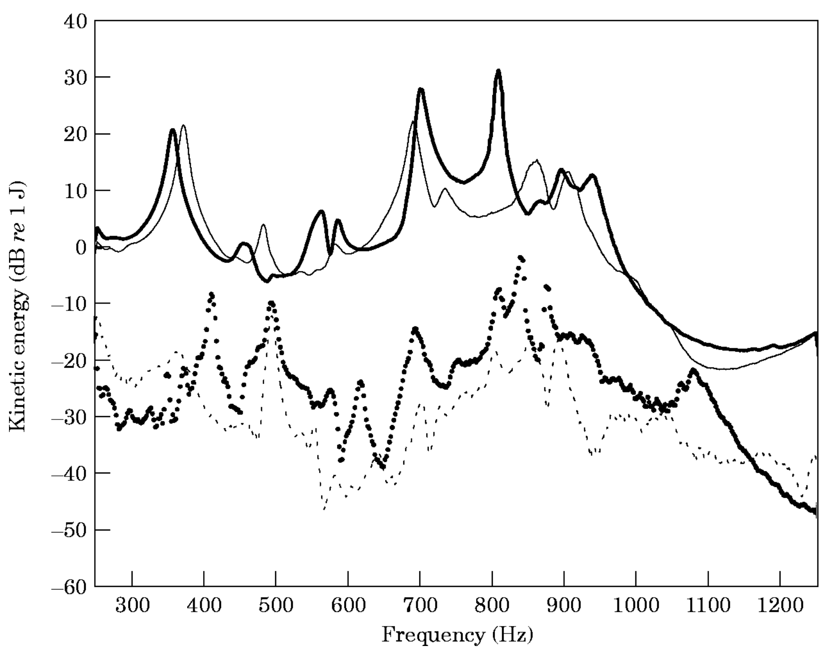

Figure 11. The calculated kinetic energy of the lower endplate assembly, comparing upper and lower positions for the secondary actuators. Light lines, lower position (as Figure 10); heavy lines, upper position; solid lines, without control; dotted lines, with active control. Otherwise strut configuration is as for Figure 10.

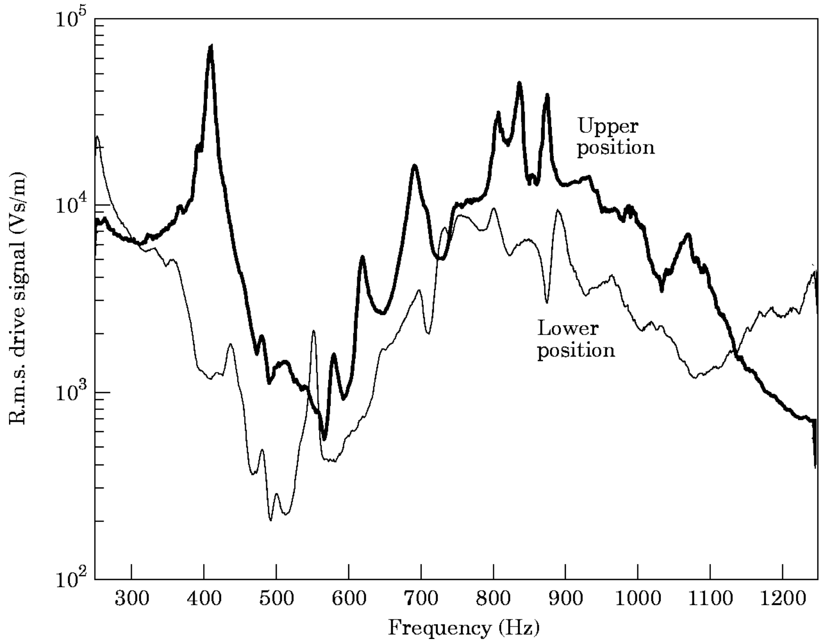

Figure 12. The root mean square of the three optimum actuator drive voltages to the actuators are compared for the upper and lower actuator positions. The strut configuration is as for Figure 11. ——, Lower position; ——, upper position.

6.5. ACTIVE CONTROL OF THE STRUT FITTED WITH ELASTOMERIC BEARINGS

During the course of the project an investigation was carried out into the use of elastomeric bearings instead of the standard steel bearings as a passive isolation measure in conjuction with active control. In Figure 13, the bold traces show the kinetic energy reduction of the lower endplate with elastomeric bearings fitted to the strut. The light traces show the corresponding result for steel bearings (previously seen in Figure 6). Apart from the bearings, the strut configuration was as for Figure 6. It is clear that the active control is much more effective with the elastomeric bearings fitted, and that the attenuation in kinetic energy is 30–40 dB over almost all of the frequency range. The control problems experienced with steel bearings at 500 Hz and 800–900 Hz have disappeared.

When the residual kinetic energy was decomposed into its different components, the torsional element was still found to be the largest contributor. The essential difference between this situation and that with the steel bearing is that there are no frequencies at which torsional motion of the endplate is amplified by the secondary actuators, and above 400 Hz significant attenuations occur. It appears likely that torsional motion of the strut is effectively isolated in this frequency range when elastomeric bearings are fitted, allowing the actuators to control all of the other forms of motion as designed.

The root mean square of the three actuator voltages required for control (Figure 14, bold trace) are generally less than in the corresponding case with steel bearings (light trace). The drive levels for longitudinal excitation with soft bearings are up to an order of magnitude below those required to control the same excitation with steel bearings fitted. This result reflects the reduced stiffness of the upper bearings, which allows the lower end of the strut to be brought to rest with a much smaller secondary force.

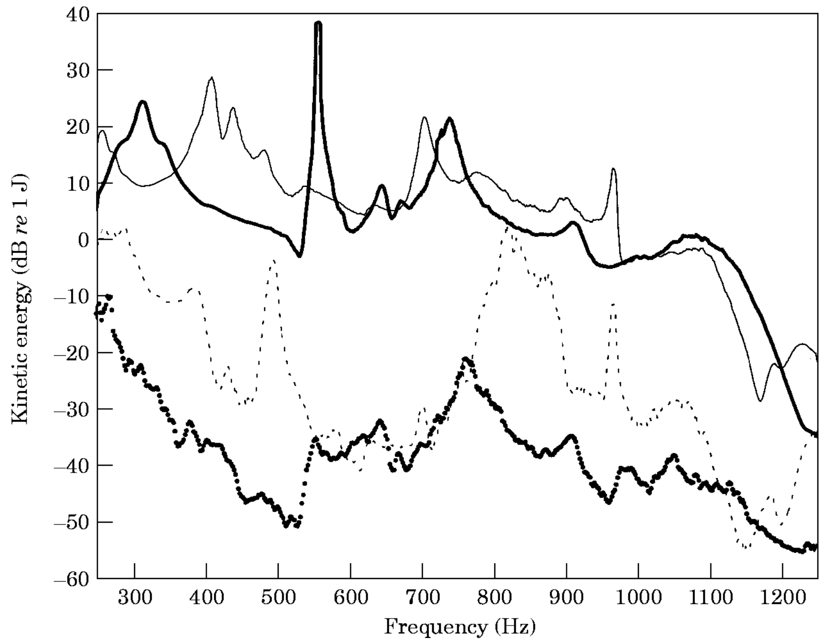

Figure 13. The calculated kinetic energy of the lower endplate assembly, comparing elastomeric and steel bearings fitted to the strut. Light lines, steel bearings (as Figure 6); bold lines, elastomeric bearings; solid lines, without control; dotted lines, with active control. The primary velocity was applied in the z direction and the actuators were in the lower position.

7. REAL-TIME CONTROL

Using a microcomputer designed for high speed digital signal processing, it has been possible to implement active vibration control of the experimental strut at a range of discrete frequencies. The control software was written at the ISVR and applies the frequency domain filtered-x LMS algorithm [19] at a frequency selected by the user. The software was modified for the tests to include a weighting matrix such that the kinetic energy of the lower endplate would be minimized by the algorithm rather than the sum of squared accelerometer signals. If $\mathbf{e}$ is the vector of six error signals, i.e., the signals from the six accelerometers attached to the lower endplate, then it is found that the cost function J to be minimized takes the form:

$$J = \mathbf{e}^{\mathrm{H}}(\omega)\mathbf{A}(\omega)\mathbf{e}(\omega), \tag{12}$$

in which $\mathbf{A}(\omega)$ is the weighting matrix defined by equation (9). For the real-time active control tests the strut rig was set up with 1 tonne static loading, the actuators at the 25% position and elastomeric bearings fitted as used for the prediction in Figure 13. Active control of the strut was attempted at a sequence of discrete frequencies between 300 Hz and 1250 Hz. (Control at 250 Hz involved an excessive drive current to the actuators, as predicted in Figure 14.) The r.m.s. input acceleration to the strut was displayed during the tests, and in each test was maintained at a constant level by adjustment of the voltage to the primary shaker. The objective of this was to represent the situation in which the primary excitation is a constant velocity source. The maximum current drawn by a typical actuator (C) was 0·26 A r.m.s. at 300 Hz.

The attenuations which were achieved with the real-time controller are shown as isolated points in Figure 15. For comparison, the attenuation in dB which was achieved at each spot frequency is shown relative to the upper prediction trace. Taken as a whole, the results

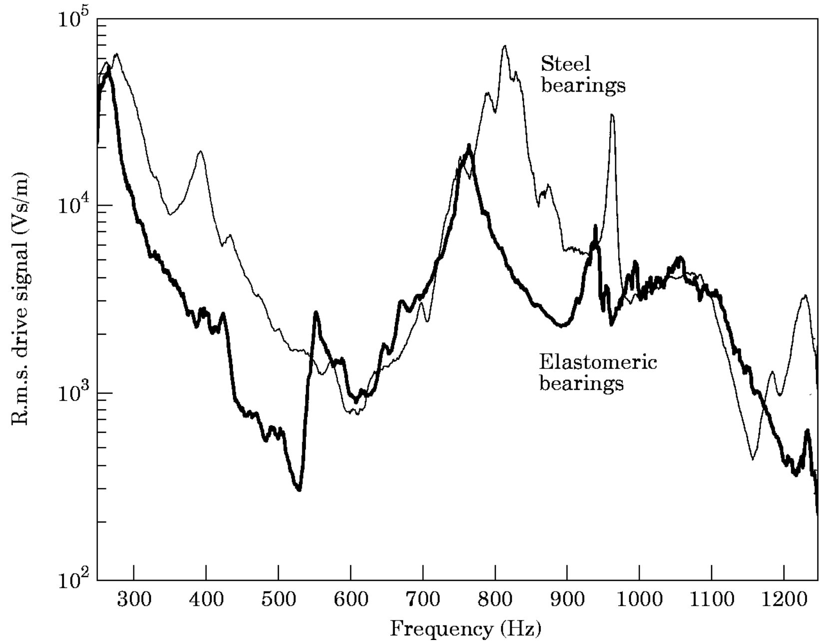

Figure 14. The root mean square of the optimum actuator drive voltages to the actuators with the elastomeric bearings fitted to the strut (——) and steel bearings (——). The strut configuration was as listed for Figure 13.

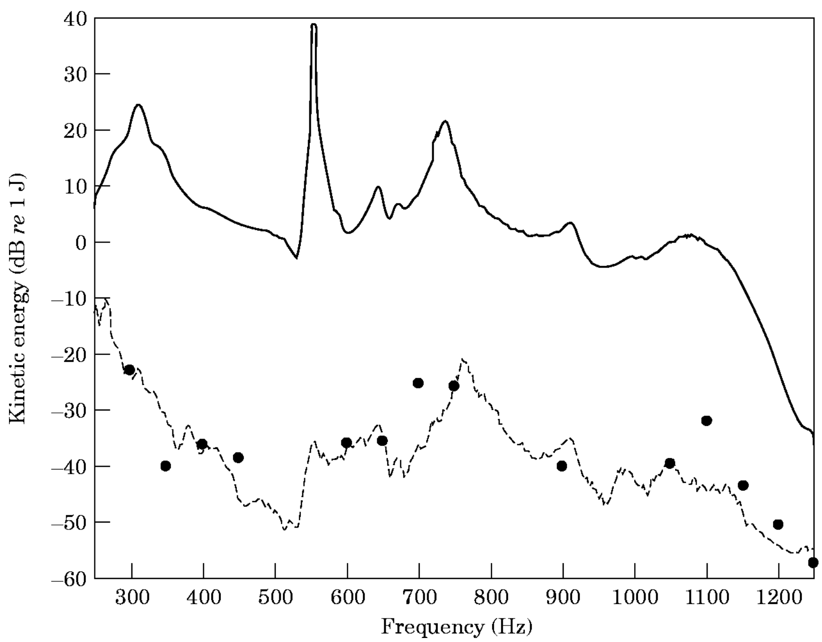

Figure 15. Real-time active control of the strut with elastomeric bearings fitted. The attenuations that were achieved using three actuators driven by the real-time controller are shown as isolated points and compared with the corresponding linear prediction from individual frequency responses. (The real-time attenuations are shown relative to the upper prediction trace.) The primary velocity was applied in the z-direction, the actuators were fitted in the lower position and the static load applied to the strut was 1 tonne.

confirm that very substantial attenuations in kinetic energy of the receiving structure are a practical possibility. Attenuations in excess of 40 dB were measured at a number of frequencies over the range. The agreement with the prediction is reasonably good, bearing in mind that with control the measured acceleration signals have been reduced to less than 1% of their initial values and so are vulnerable to noise in the measurement equipment.

Despite measurement difficulties at a few discrete frequencies, the real-time control tests have shown broad agreement with the linear predictions and confirm that active control of vibration transmission through an actual helicopter strut is practical at frequencies up to at least 1250 Hz.

8. CONCLUSIONS

A gearbox support strut from the EH101 helicopter has been set up in the laboratory under realistic static loading conditions and three magnetostrictive actuators have been clamped to it to investigate the control of longitudinal and lateral vibration transmission to a connected receiving structure. The control objective was to minimize the kinetic energy of vibration of the lower endplate assembly to which the strut was connected. The main conclusions are as follows:

(1) Using an extensive set of frequency response measurements, it has been possible to predict on a linear basis the attenuation in the kinetic energy of the receiving structure at any discrete frequency in the measurement range for a wide range of conditions. The control characteristics have been calculated for the practically realistic case of a constant velocity source as the primary input.

(2) The calculations based on frequency response measurements have shown that, with the installed steel bearings on the strut, attenuations of 30–40 dB are possible in the kinetic energy of the lower endplate structure over a broad range of frequencies between 250 and 1250 Hz. At some frequencies in the range, notably around 500 Hz and 800 Hz, the control is poorer, apparently due to torsional motion of the strut, which was amplified by the secondary actuators. Similarly, good control was obtained when the primary excitation to the strut was applied laterally rather than longitudinally.

(3) Elastomeric bearings were fitted to the strut, and it was found that they provided improved isolation of the receiving structure from torsional vibration. As a result, attenuations of around 40 dB in the kinetic energy of the receiving structure were predicted over virtually the whole frequency range of interest. In addition, the required drive signals to the actuators were significantly reduced by up to an order of magnitude. This is thought to be mainly because of the reduced stiffness of the bearing upstream of the actuators.

(4) Real-time active control has been implemented at discrete frequencies up to 1250 Hz on the test strut and has broadly confirmed the linear predictions. Attenuations in excess of 40 dB were measured in a number of cases. The tests confirmed that the active control of vibration transmission through a helicopter strut is practical at frequencies up to at least 1250 Hz.

(5) An inertial configuration was adopted for the actuators, giving a reduced secondary force requirement over part of the frequency range where the inertial actuator behaves as a tuned "dynamic absorber". Magnetostrictive actuators are considered to be suitable for the helicopter strut application, because of the high forces that they can develop.

(6) The measurements and predictions presented in the paper involved small disturbances to ensure linear behaviour of the strut. In a practical implementation, issues of non-linearity would need to be addressed. The natural frequency of the inertial actuators was found to vary by more than 200 Hz with changes in excitation level. This arose because

the effective stiffness of the magnetostrictive actuators varies with amplitude of stress variations in the Terfenol-D material. The change in natural frequency would change the secondary path characteristics of a feedforward active controller, and account would need to be taken of this in a practical system. Changes in strut load (altering the stiffness of the bearings) were also found to alter the secondary path characteristics.

(7) The strut was supported by two endplates of mass 17·6 kg, which strongly affected the dynamic response of the complete experimental assembly. The boundary conditions experienced by a gearbox support strut under flight conditions would naturally be completely different. The mechanical impedance presented to the strut by an actual helicopter fuselage would need to be measured before the work detailed here could be applied in a practical situation to reduce interior helicopter noise.

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