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THE WESTLAND ROTOR HEAD VIBRATION ABSORBER DESIGN PRINCIPLES AND OPERATIONAL EXPERIENCE

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Abstract

The design process for a rotor head mounted vibration absorber for the Lynx and Westland 30 is discussed. The objectives of low damping, no routine maintenance and ability to suppress vibratory loads arising from b-1 and b+1 per rev blade loads are achieved, and result in considerable advantages compared with centrifugal absorbers. It is shown that the distribution of the mass of the absorber is important with respect to the performance of the device.

The effectiveness of the absorber in reducing airframe vibration on the Naval Lynx, Utility Lynx and Westland 30 is presented. During an operational research exercise into the benefits of reduced vibration on Naval Lynx defect rates a significant reduction in aircraft maintenance requirements was achieved, in addition to the improvement in subjective ride quality.

Introduction

The desire to minimise helicopter vibration is shared by manufacturers, operators and passengers, but the price of satisfying this aim in terms of weight and life cycle costs must be minimised. Effective elimination of most low frequency components of vibration can be achieved at little cost by careful control of manufacturing tolerances and by introducing procedures such as periodic tail and main rotor dynamic mass balancing. Achieving acceptable levels of vibration at the main rotor blade passing frequency, bP, is an order of magnitude more difficult due to the unavoidable azimuthal variation in blade aerodynamic loading in forward flight.

During the design of a rotor system the effect of such fundamental parameters as number of blades, type of hub and blade stiffness on fuselage vibration are given careful consideration, see reference 1 for example, whilst of course having to satisfy all the other design requirements. Regardless of the attention paid to vibration by the rotor designer some residual bP forcing will be generated and to minimise its effect careful consideration must also be given to the fuselage structure to avoid the presence of major modes of the aircraft close to bP. If the magnitude of the rotor excitation is expected to be large the designer can introduce fuselage isolation systems such as DAVIs (ref. 2) or LIVE (ref.3); but it will be recognised that such systems led to considerable weight penalties and probably require significant maintenance and, possibly, sophisticated tuning procedures.

An attractive alternative to isolation systems is the use of tuned mass absorbers, many of which have been used in the industry. Examples are the S61 battery absorber, the S61 and S76 rotor head bifilars (ref.4), and blade pendulum absorbers (ref.5). In order to produce a global effect it is desirable to attach absorbers as close as possible to the source of vibration, and an ingeneous solution is to mount the absorber on the main rotor hub and to use the centrifugal force field as the 'spring', as in the bifilar. This has the added advantage of staying 'in tune' as the rotor speed varies, but it does require the use of rolling or sliding contact between the bob weight and bifilar arm. This introduces the possibility of wear and the need for routine maintenance. A novel solution to the wear problem has recently been announced by Viswanathan and McClure, ref.6, who have designed a bifilar using mercury as the bob-weight, but this in turn introduces problems with the containment of this very corrosive liquid. If the dominant rotor loads are in-plane shears or pitch and roll moments these arise from (b-1)P and (b+1)P oscillations of the rotor blade, and in general a bifilar can only deal directly with one of these components, although Mouzakis (ref.7) has managed to incorporate both (b-1)P and (b+1)P tuned masses in a single device.

As an alternative to the bifilar, Westland have designed a tuned-mass absorber which can be mounted on the main rotor head but which uses the structural deformation of beams as the spring. This absorber operates in the plane of the rotor disc and is tuned to bP. Preliminary results from the development testing of the device have been given by White, Ref.8, since when considerable in-service experience has been gained. This paper reviews the design concept, the results of the development testing and the in-service experience on both the Lynx and Westland 30 helicopters.

Design Considerations

The absorber was initially designed for the Lynx, and the first problem was to determine the force capability of the device. Although the Lynx semi-rigid rotor system has considerable advantages in terms of drag, maintenance and aircraft manoeuvrability it is well known that this type of rotor system generates large vibratory moments. For example, from the analysis of rotor strain gauge data gathered during flight trials the following values for the 4P vibratory loads at 156 and 172 knots have been deduced:

Longitudinal shear	2000N	at 156 kts.		1300N	at	172 kts.
Lateral shear	1300N	"	"	2500N	"	"
Vertical force	3100N	"	"	4000N	"	"
Pitch moment	3200Nm	"	"	5400Nm	"	"
Roll moment	4500Nm	"	"	4200Nm	"	"

From knowledge of the fuselage mode shapes it has been found that 1 Nm of pitch or roll moment produces approximately the same level of airframe vibration as 1 N of longitudinal or lateral shear. Using this ratio it is possible to compute the force requirement of the absorber, this being 5200 N at 156 knots and 6700 N at 172 knots longitudinally, and 5800 and 6700 N laterally. Based upon these considerations the required force capability of the absorber was estimated to be $\pm$ 7000 N in any in-plane direction, bearing in mind that not all the vibratory rotor loads will be in-phase. Vertical vibratory rotor forcing is the least important on the Lynx and no attempt has been made to absorb this component.

A second requirement was to reduce the damping of the absorber to an absolute minimum, since it is easy to show that an absorber placed at the source of excitation is most effective when its damping is low. To achieve this objective no sliding or rolling joints were to be used, and this would help to meet the third objective of no routine maintenance. A concept which meets these aims and has in addition polar symmetry is shown in Fig.1. The absorber is attached to the rotor hub by a central spindle which utilises existing holes in the hub (for the attachment of an aircraft lifting eye). The outer ring is effectively a rigid body and forms the mass part of the device, and is connected to the spindle by spiral springs. These springs are disposed symmetrically around the spindle, and provided there are at least three the in-plane stiffness of the absorber will be the same in any direction

Spring Design

Stiffness and stress calculations for the spiral spring were performed using a purpose-written computer programme in which the spring was modelled as a large number of short straight beams. From the results of the analysis of a single spring the overall stiffness of the outer ring relative to the spindle may be calculated. Factors which influence the overall stiffness are the usual terms such as cross sectional shape, material properties and spring length and also how much each spring wraps around the spindle, since for a long straight beam the axial stiffness is much greater than the bending stiffness. The torsional rigidity of the spring does not feature in the in-plane and yaw motion of the absorber, but it is the dominant flexibility for out-of-plane motions.

A number of materials were considered for the springs. From consideration of the required stiffness and the allowable stress the weight of a set of springs made from steel, titanium and GFRP would be 27, 11 and 3.6 kg, respectively. Since the manufacturing difficulties are also less with unidirectional GFRP this was chosen as the best material. It also introduces the possibility of adjusting the shear stiffness of the spring relative to the tensile stiffness, which could be useful with respect to the placement of the natural frequencies of the absorber in the vertical and pitching degrees of freedom.

Consideration was given to tapering the springs, but was rejected as not being cost effective. Unlike a cantilever absorber the bending moment distribution along the length of a spring is not zero at the tip and a maximum at the root. In fact the distribution is basically sinusoidal, but because the device works in any direction in its plane the distribution of peak tensile stress along the springs is as shown in Fig.2, in this case for a displacement of the outer ring of 10 mm.

Whilst it is desirable to wrap the springs around the spindle as much as possible the built-in curvature must not be too high as this can result in high direct stresses normal to the axis of the spring. Also, it is essential to maintain an adequate clearance between the springs as the absorber deflects. Thus, the more the springs wrap around the spindle the greater the overall diameter becomes, which will increase aerodynamic drag. The precise shape of the springs is not critical, and for ease of manufacture the shape used is a series of circular arcs. Again for manufacturing reasons the depth of the spring was limited to 38 mm, and therefore the final absorber is made from two layers with four springs in each layer.

The calculated stiffnesses for the complete absorber are:

In-plane	692,000	N/m
Vertical	116,000	N/m
Yaw	7,120	Nm /rad
Pitch and roll	5,650	Nm /rad

and the overall diameter is .46 m. The low vertical stiffness relative to the in-plane is a consequence of the low shear rigidity of unidirectional glass fibre. The maximum tensile stress is 130×10^6 Nm^{-2} for a deflection of 10 mm.

Mass Design

With all the mass concentrated in the outer ring the pitch mode frequency of the absorber is only just over half the in-plane frequency. Laboratory experiments with a prototype absorber showed that this could lead to subharmonic pitch excitation of the absorber which resulted in excessive pitching motion. To overcome this difficulty part of the tuning weight was moved from the outer ring to top and bottom covers, as shown in Figure 1, which reduced the pitch inertia and therefore increased the pitch mode frequency.

In addition, the modification introduced a fail-safe feature into the design. In the extremely remote event of failure of all eight springs the absorber cannot leave the rotor head and also cannot touch the hub, due to the geometry of the top cover and top spindle.

Fine tuning is achieved by adjusting the number of top tuning weights, a very simple operation. The total weight is 55 kg. of which 40 kg. forms the moving mass of the absorber. This is less than 1% of the gross all-up-weight of the Westland 30.

Laboratory Testing

Laboratory testing has consisted of endurance testing to establish the S-N curve for the springs and vibration testing to investigate the dynamic behaviour of the absorber. The natural frequencies of a typical absorber are:

In-plane	20.8 Hz.
Vertical	9.9 Hz.
Pitch and roll	13.4 Hz,
Yaw	12.2 Hz.

Blade passing frequency (4P) is 22 Hz. for both the Lynx and Westland 30. The results quoted above for the absorber natural frequencies are with the spindle attached to a rigid earth. The effect of a finite fuselage impedance is to move the in-plane mode to 4P on the aircraft. Variations in spring manufacturing tolerances can lead to slight differences in the in-plane natural frequency in the lateral and longitudinal directions, but to date the maximum difference has been 0.12 Hz., and flight trials have shown this to be acceptable.

The damping of the in-plane mode has also been measured and typically is $\frac{1}{4}$ percent of critical; i.e. an amplification at resonance (Q) of 200.

Vibration testing has been carried out with the absorber spindle fixed to a table which could be vibrated in only one direction, being effectively rigid in all other directions. The absorber was not rotating during these tests. An interesting phenomenon occurs when the table is vibrated at a frequency close to the sum of the pitch and vertical modes' natural frequencies, i.e. 23.3 Hz. For low amplitudes the in-plane response of the outer ring occurs at the forcing frequency and the out-of-plane response is very small. As the amplitude is increased a threshold is reached at which the outer ring starts to pitch and heave at the modal natural frequencies, 13.4 and 9.9 Hz. respectively, even though the excitation is at 23.3 Hz. Fig.3 shows the waveform of the vertical acceleration measured on the outer ring at amplitudes just below and just above the threshold; the change in the waveform is quite clear. As the in-plane excitation frequency is varied either side of the critical frequency the amplitude of the input required to produce the out-of-plane motion increases, as shown in Fig.4. It can be seen from Fig.4 that the region of non-linear behaviour is well clear of the normal operating rotor speed.

Non-linear behaviour similar to that obtained from the absorber has been reported by Barr and Ashworth, ref.9, but no attempt has been made to model the phenomenon in detail. Barr refers to the motion as 'autoparametric' as it arises from parametric excitation amplified by internal resonances. This type of behaviour is only rarely encountered in real engineering structures, and is probably only observable here because of the low damping. The problem found with the original configuration of the absorber, when the pitch frequency was half the in-plane frequency, was another manifestation of autoparametric behaviour.

An advantage of composite material over isotropic material becomes apparent here, since the vertical and pitching stiffnesses are dominated by the shear stiffness of the springs. This could, if it were necessary, be simply increased by the addition of glass or carbon $\pm 45^{\circ}$ wraps to the springs.

In-Flight Testing

The absorber has been extensively flight tested on both the Lynx and Westland 30, which have the same hub design and rotor speed. It is a standard fit for the Westland 30 and is offered to users of the Lynx who wish to avail themselves of its considerable benefits.

As part of the substantiation of the absorber,motions of the outer ring and stresses in the spindle have been measured in flight. The motions were measured by fitting six accelerometers to the ring. Computing displacements from the accelerometer measurements is not a trivial task, due to the effect of the steady rotation of the rotor on the accelerometer readings. A typical time history, over one rotor revolution, of the dominant in-plane displacements is shown in Fig.5, the dominant frequencies being 3P and 5P. The motion of the centre of mass of the absorber as seen by an observer rotating with the rotor is also shown in Fig.5, and this agrees well with the theoretical prediction given in ref.8. Excellent correlation was obtained between the stress measured in the spindle and the absorber force computed from the theoretical stiffness and the measured amplitudes.

The maximum force produced by the absorber on the Westland 30 is $\pm$8500 N at 1.1Vne, but within the normal flight envelope the force does not exceed 7000 N, which is very close to the original design load. Based upon these data the spring life is expected to exceed 2000 hours, and indeed at the time of writing four absorbers have each already exceeded 1000 hours with no maintenance.

Absorber motions and forces are very similar on both the Lynx and Westland 30, which is hardly surprising as the rotor systems are very similar. Thus a similar life is to be expected on the Lynx.

Lynx Experience

The results of flight trials of the absorber on a typical in-service Royal Navy variant of the Lynx is shown in Fig.6 where it can be seen that considerable benefits arise from fitting the absorber, especially at the co-pilots seat and in the fore-aft direction. The slight increase in the pilots vertical vibration is more than compensated for by the reduction in fore-aft cockpit vibration, and this is reflected by the favourable comments of the crew. Typical of the comments made by service personnel is that the aircraft at 150 knots with the absorber felt like a standard

Naval Lynx at 120 knots. Operating the aircraft in different roles and at different all-up-weights does not require re-tuning the absorber.

An operational research exercise into the benefits of reduced vibration has been carried out on the RN Lynx during which absorbers were fitted to three aircraft and the defect rates with and without the absorbers compared. None of the absorbers required maintenance or had defects, and one achieved over 700 hours of flying. The overall average defect rates for the three aircraft are compared in Fig.7 where the benefits are seen to be considerable. Analysis of the defect rate in terms of cause indicates that the defects which could be directly linked to vibration were halved by fitting the absorber; this is also shown in Fig.7. The total reduction in defect rate is three times that estimated to be due directly to vibration, which indicates that some defects which could not be linked directly with vibration were in fact caused or aggravated by vibration.

Similar improvements in airframe vibration have been obtained on the Utility variant, as is shown in Fig.8; the difference between the Utility and Naval versions being the lower level of fore-aft vibration on the Utility aircraft. This is due to a difference in airframe dynamics resulting from the tail fold joint. The results shown in Fig.8 were obtained from a typical in-service aircraft. A trial similar to that on the Naval Lynx is underway on the Army aircraft, but the flying hours to date are too low to be conclusive. The comments by the crew are however very favourable. The British Army have recently decided to fit absorbers to their entire fleet of Lynx.

Westland 30 Experience

The absorber is a standard fit for the Westland 30. Fig.9 shows the in-flight vibration levels measured on a development standard Westland 30 with and without the absorber at a number of positions in the cabin and cockpit. It shows a very large reduction in vibration is obtained by fitting the absorber, to levels which are very reasonable. The high levels of vibration present in the cockpit without the absorber are due to the presence of a mode of the fuselage close to 4P (at 24 Hz. compared with 4P = 22 Hz.). In order to attempt to further reduce vibration a structural modification was tried which increased the frequency of this mode to 28 Hz. This had the desired effect of significantly reducing the cockpit vibration without the absorber. When this modification was flown in conjunction with the head absorber it was found to be no better than the datum aircraft plus absorber.

This interesting but rather disappointing result has been examined theoretically in a very simple manner by analysis of the damped, two mass plus absorber model shown in Fig.10. The response of the mass m_1 as the forcing frequency is varied is shown in Fig.10 for the case of no absorber. The response obtained with an absorber fitted is also shown, for various forcing frequencies, and it can be seen that the minimum response obtained with the absorber is almost independent of how close the baseline system is to resonance. Also, the force the absorber is required to generate does not change significantly with the natural frequency of the baseline system; the force is related simply to the magnitude of the external force. Thus, the improvement obtained by fitting an absorber is increased the closer the basic system is to resonance, but the absolute vibration level achieved is almost independent of the dynamics of the original system. This phenomenon has been confirmed by the above mentioned flight test results from the Westland 30, because on this aircraft the response is dominated by a single mode. On an aircraft for which the response of a number of airframe modes is important the effect of structural changes, in the presence of an absorber, would be less simple than shown in Fig.10, and could therefore be beneficial.

The effect of rotor speed changes on vibration is shown in Fig.11 where the desirability of maintaining close limits on the rotor speed can clearly be seen. It is interesting that the minimum vibration level at the two positions is achieved at slightly different rotor speeds; consequently absorber tuning is always something of a compromise. In view of this result isochronous rotor speed governing has been introduced on the Westland 30, which limits the rotor speed variation in normal flight to $\pm$ 0.5%.

Fine tuning of the absorber is achieved by flying the aircraft and noting the cabin vibration at one selected standard position as the rotor speed is varied at 100 knots. This is a routine task during the production clearance flying of new aircraft. If it is necessary for an operator to change an absorber a special piece of test equipment, known as the "bonkmeter", is available which allows him to carry out the same procedure. This kit can also be used to measure the natural frequency of the absorber on the aircraft, thereby allowing a very simple check of absorber integrity to be made.

The absorber has given very little trouble in service, and as mentioned above four absorbers have already each exceeded 1000 hours of flying. The absorbers are 'on-condition' and removed only if the crew have observed an increase in airframe vibration. Subsequent investigation of rejected absorbers has shown that none has suffered any structural damage as a result of flying within the permitted envelope.

Comparison with the Bifilar

In the 1970's an inplane bifilar absorber, tuned to 3P, was flight tested on the Lynx. At its best the performance of the bifilar in terms of vibration reduction was nearly as good as the flexispring absorber, but it was not consistent from flight to flight, and, in addition it required considerable routine maintenance. The weight of the bifilar was 65 kg. Comparing the two types of absorber in terms of weight, maintenance requirement, ease of manufacture and performance the flexispring absorber is better. The only advantage of the bifilar is its self-tuning feature, which however is not found to be particularly important due to the close governing of rotor speed on modern helicopters.

Conclusions

The design process for the Westland flexispring rotor head absorber has been described, and the concept shown to be viable. It is demonstrated that such a device can be very beneficial to helicopter vibration, producing improvements in both the subjective ride qualities and the maintenance requirement. The lack of routine maintenance required by the absorber and its ability to deal with vibratory loads arising from b-1 and b+1 per rev. blades loads gives it considerable advantages over the bifilar type of absorber. It is also shown that the absorber deals effectively with both shear and moment excitation from the rotor.

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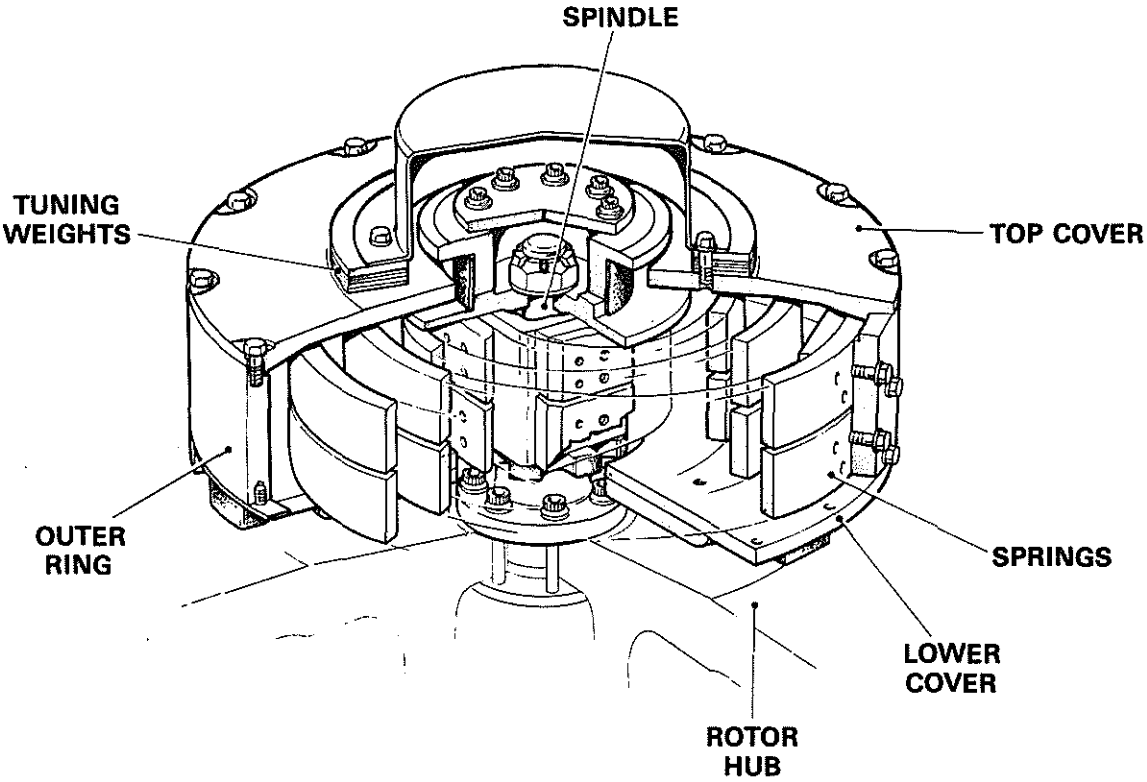

FIG 1 GENERAL ARRANGEMENT OF ABSORBER

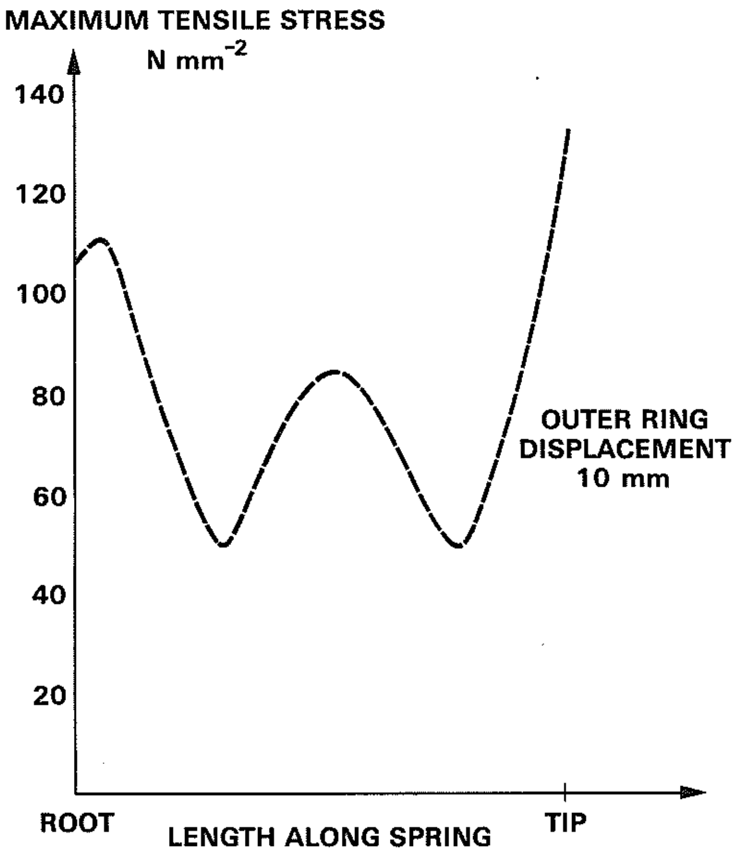

FIG 2 SPRING TENSILE STRESS DISTRIBUTION

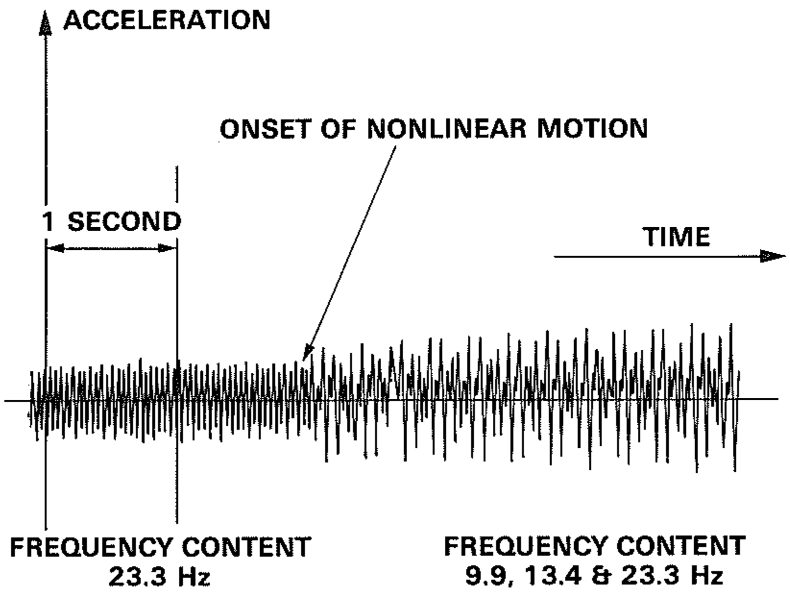

FIG 3 NON-LINEAR ABSORBER MOTION

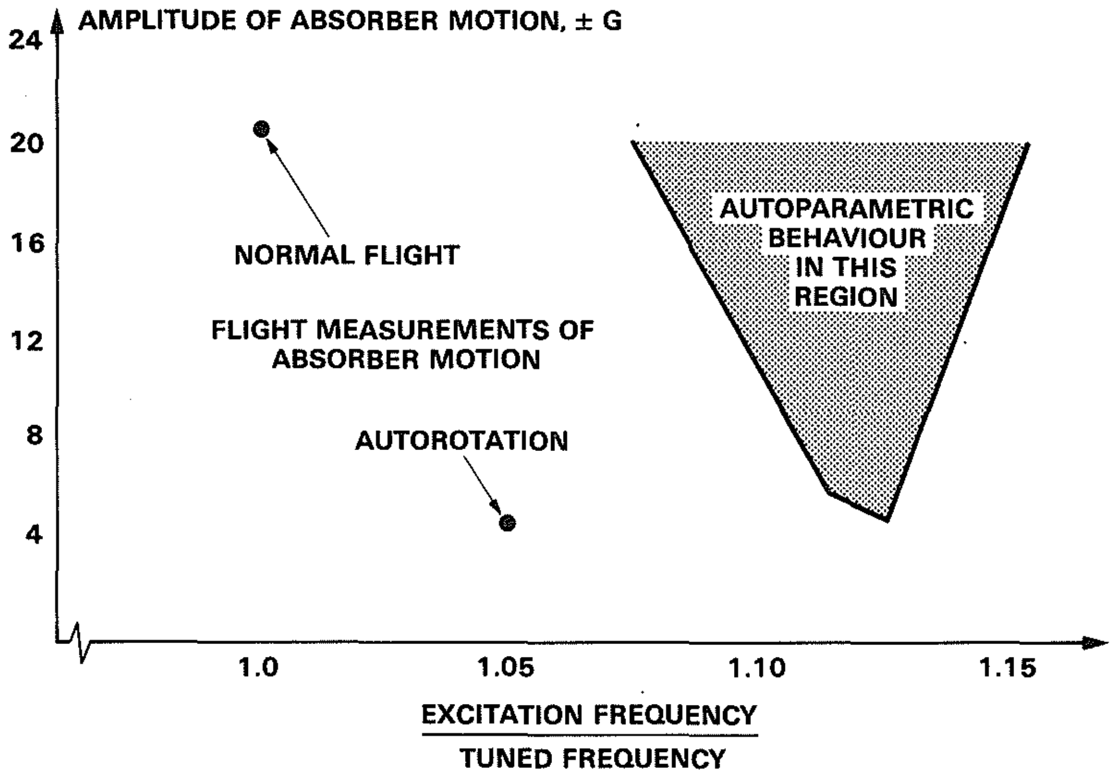

FIG 4 AUTOPARAMETRIC MOTION BOUNDARY

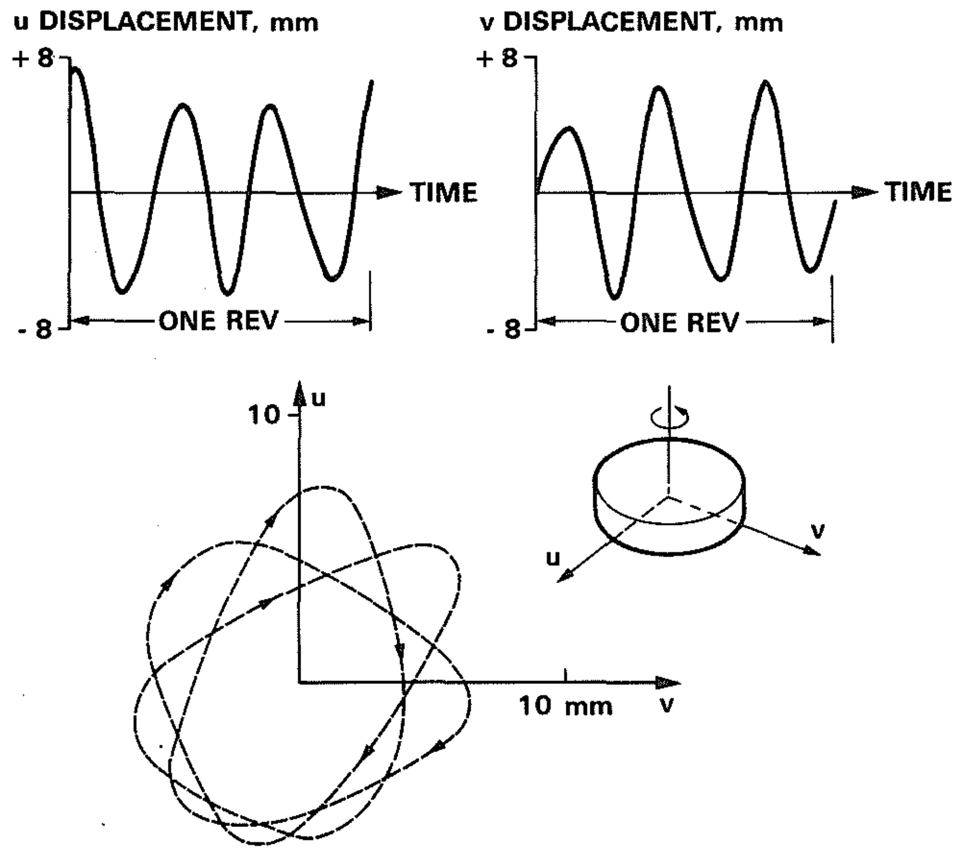

FIG 5 ABSORBER MOTION, FLIGHT MEASUREMENT

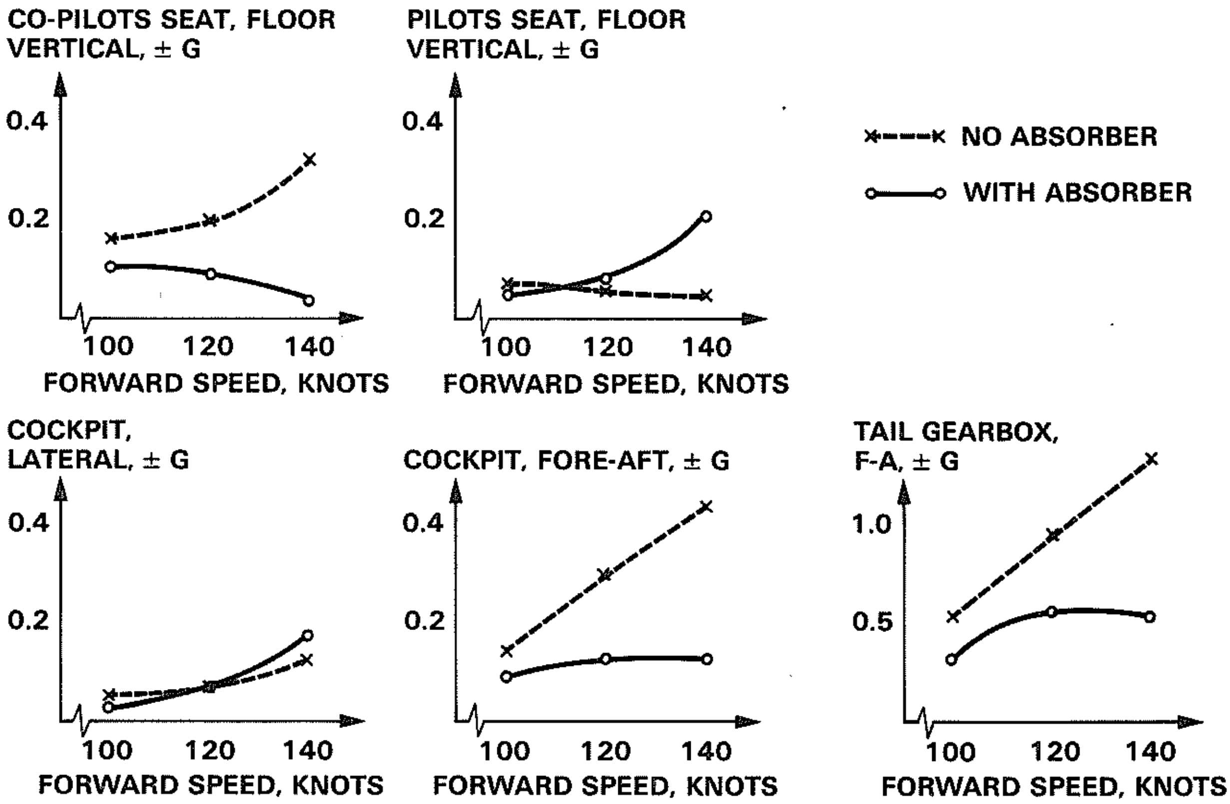

FIG 6 NAVY LYNX VIBRATION, WITH AND WITHOUT ABSORBER

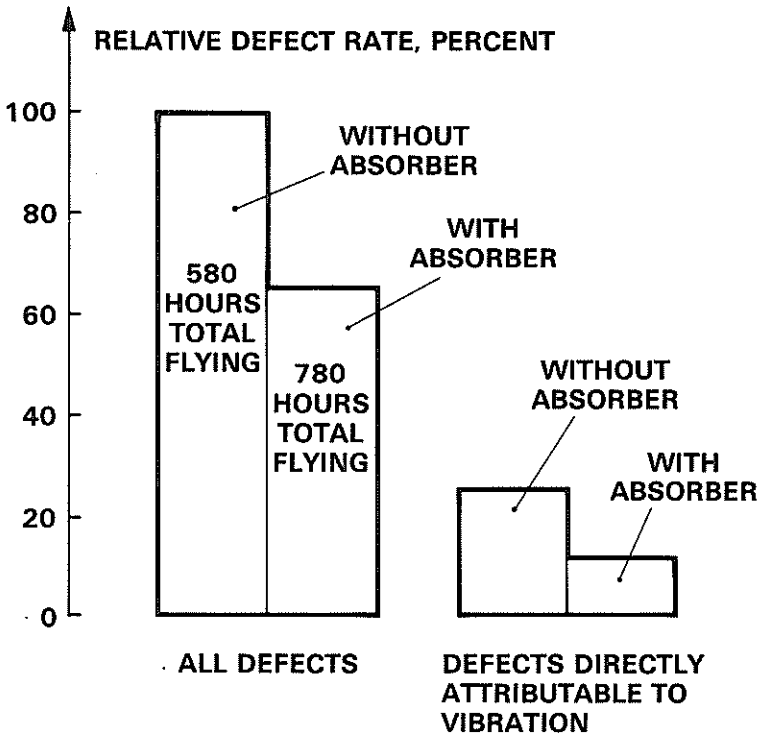

FIG 7 RESULTS OF THE OPERATIONAL RESEARCH EXERCISE ON NAVAL LYNX

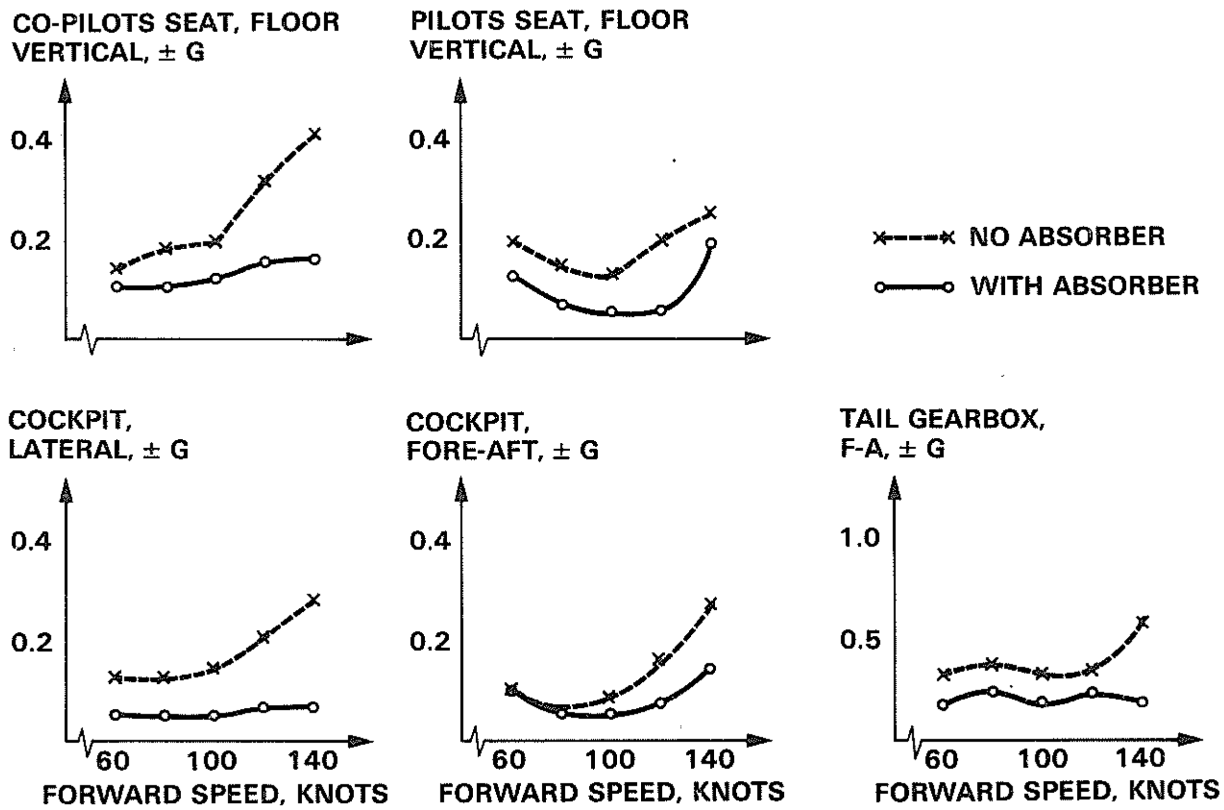

FIG 8 ARMY LYNX VIBRATION, WITH AND WITHOUT ABSORBER

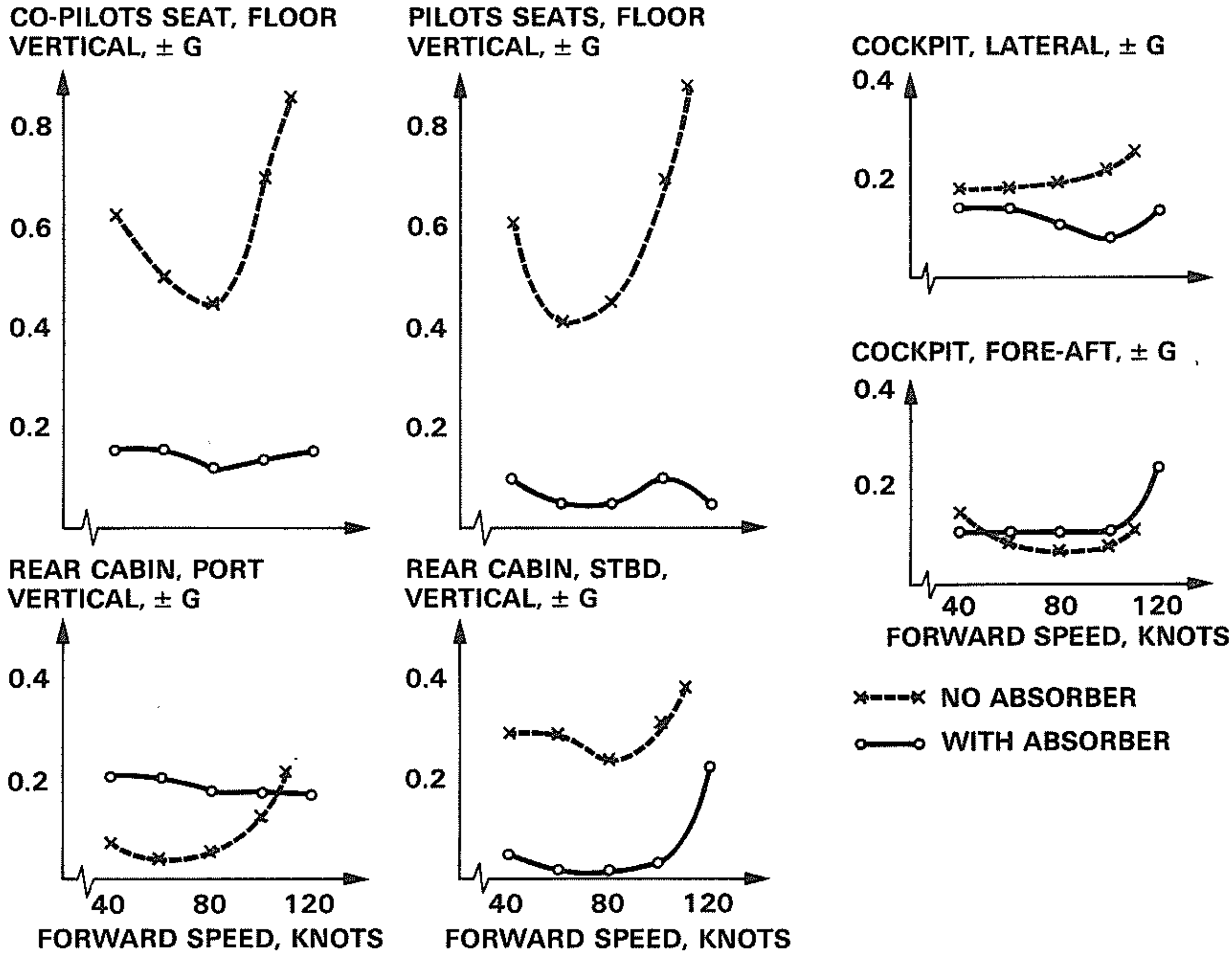

FIG 9 WESTLAND 30 VIBRATION WITH AND WITHOUT ABSORBER

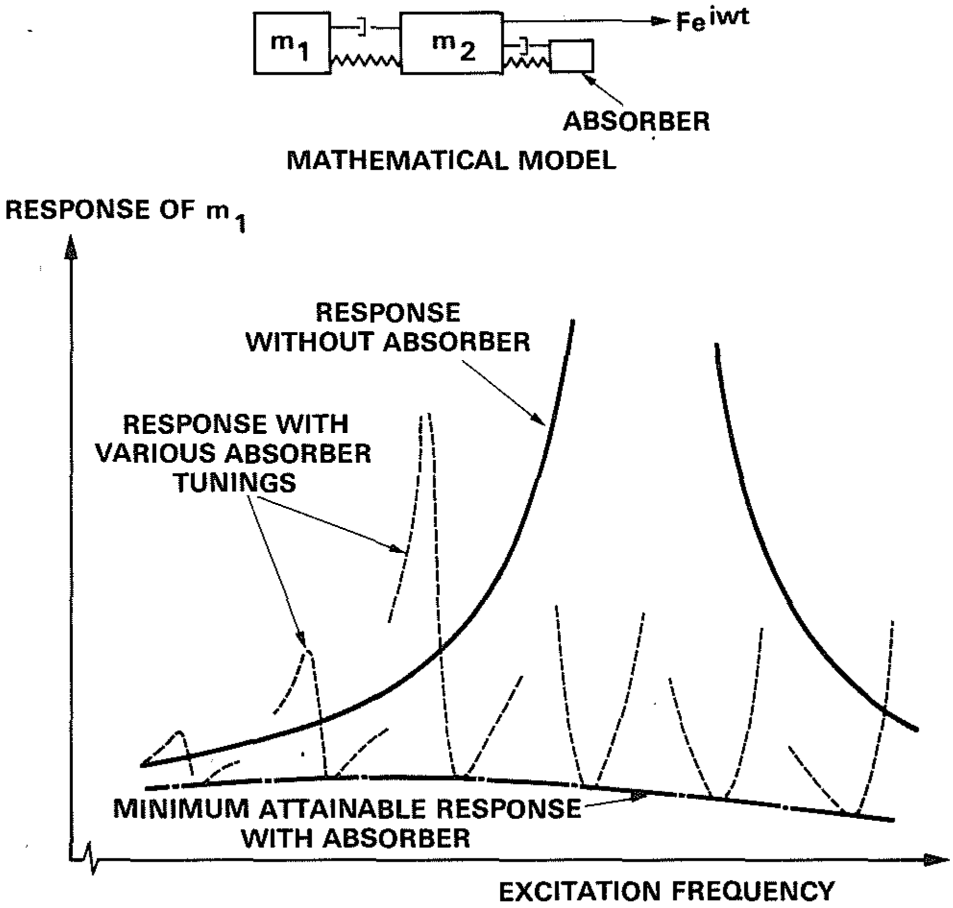

FIG 10 ABSORBER EFFECTIVENESS, 2 D.O.F. SYSTEM

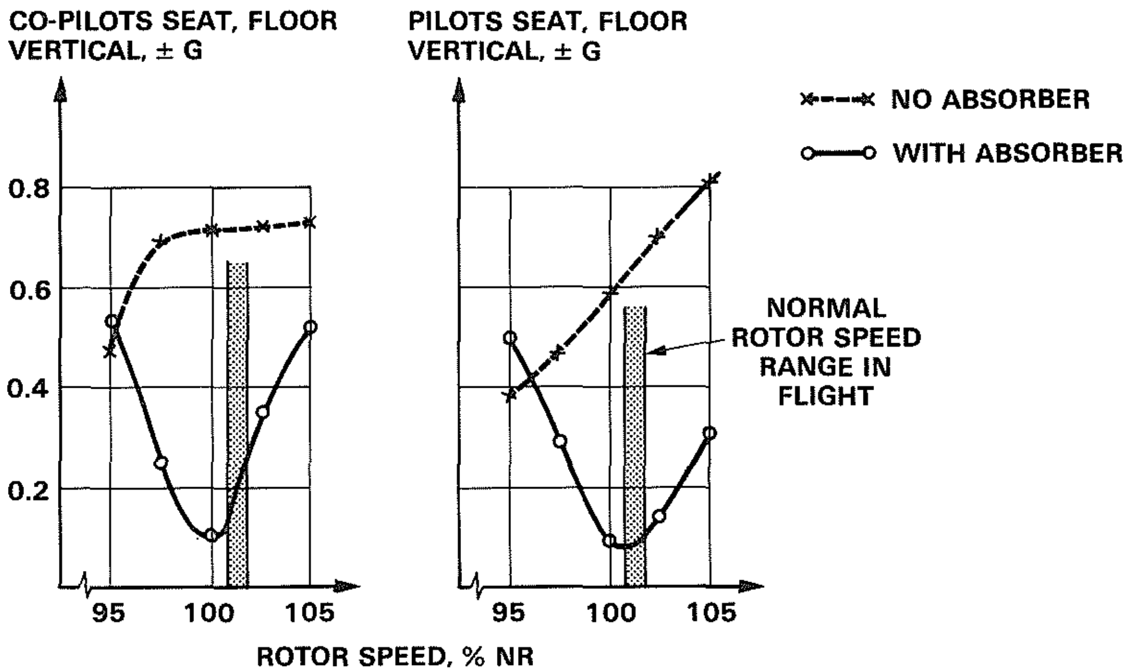

FIG 11 EFFECT OF ROTOR SPEED ON WESTLAND 30 VIBRATION